CONVEYOR BELT TECHNIQUE

DESIGN AND CALCULATION

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Foreword

Conveyor belts have been used for decades to transport bulk and unit loads. They have proved their worth everywhere because belt conveyor installations can be adapted to meet nearly all local conditions. They are work-safe and economical.

The demand for ever increasing capacities and ever longer conveying lengths has accelerated the development of the belt conveyor technique, new materials are being developed, new conveying systems are being planned and tested especially those having regard to the environmental.

The conveyor belt plays the major part in the whole system and has to overcome the many and varied stresses. In addition to this every conveying problem is different and needs careful planning and selection of the right elements in order to achieve the optimum conveying capacity in an economical way.

There are a number of practical rules, values and experiences which can be useful during the planning stage. This manual aims to be of assistance to operators, engineers and project people and provides a substantial amount of elementary data. In addition there are a number of instructions and hints offered to enable accurate calculations or checks on installation components that impinge on the running of the belt.

In future there will be more and more use of the computer for calculations and dimensioning of belt conveyors. With this often the correlation of the valuation criteria will no longer be recognizable. This manual should also help to understand the background to a calculation, the selection criteria for an optimum belt type and to recognize a special operating case.

All new standards DIN, EN, or ISO, have been taken into consideration as well as the results of individual research studies. The developments continue and we are grateful for all hints and practical experiences.

July 1994

Chronological Development

Until the mid 1970's conveyor belt development and technology was concentrated on the search for appropriate materials for the belt and the solving of drive problems. In the first instance transmission of traction played a part. As the demand grew for conveyors of larger capacity and longer length, additional requirements effecting the belt had to be considered and researched such as greater work load, elongation, slit resistance and endless splice jointing.

from 1870	Trials with plain cotton belts.
up to 1914	First rubber conveyor belts developed from drive belts.
1921	Founding of the Enerka factory. Manufacture of drive belts and later conveyor belts.
1923/1924	First use of belts underground, not a success due to drive problems.
1926	First belts with robust Balata reinforced covers.
from 1928	Use of belts with Maco cotton plies.
from 1933	Development of Rayon/cotton belts and pure rayon belts. Transition from natural rubber to synthetic rubber for protection of carcase.
from 1939	Increased use of rayon and synthetic rubber.
1941/1942	Use of PVC belts above ground.
1942	Steelcord belts used for the first time for major long haul installations in the United States.
from 1945	Further development of rayon belts. Introduction of mixed material fabrics includes synthetic weft.
1954/1955	Development of high tensile strength belts e.g. plies from rayon, polyamide and polyester. Cover rubbers with various surface designs. Steep incline belting with profiles and cleats.
from 1955	Development and use of steel cord belting in Europe.
from 1970	Use of Aramide as reinforcing material for the carcase
from 1980	Development of new conveyor systems e.g. the tube conveyor, hammock conveyor.

This brief history illustrates the more important stages in conveyor belt development. The search for new materials became necessary because of the ever increasing demands to optimise installation construction such as pulley diameters, vertical and horizontal curves etc. and the demands of new conveying systems.

Development Goals

- **Optimising conveyor belt reinforcing materials** i.e. exploit to their maximum strength limits to obtain optimum working life economy.
- **Optimum dimensioning** of installation components e.g. pulley diameters, idlers, bearings and shafts, troughing transitions, drive and take-up systems.
- **Adaptation of cover quality** to each duty i.e. cover quality to provide optimum solution at the most economical cost.
- Manufacture of conveyor belts and installation construction with the **environment in mind.**

DUNLOP-ENERKA
Belt Testing Rig

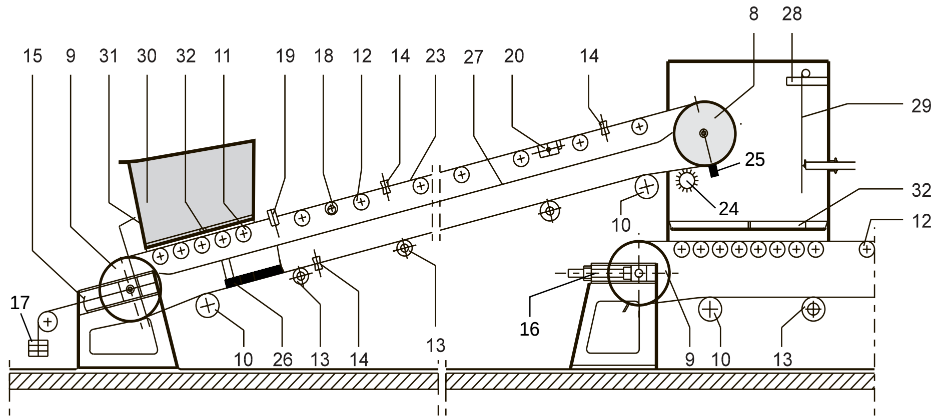
To assist in future development of new belt types and splice joining techniques, Dunlop-Enerka have developed and installed a new type of belt testing rig. The principal intention of the testing methods is to simulate actual operating conditions encountered by a belt in service.

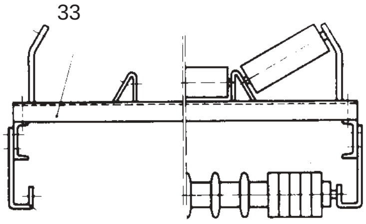
With this rig it is possible to simulate changing stresses, bending changes, apply maximum pre-tension, use various pulley diameters, observe belt slip to capacity limits etc. Analysing the test data can enable optimum use to be made of conveyor belt materials and belt conveyors thereby providing more economical operations.

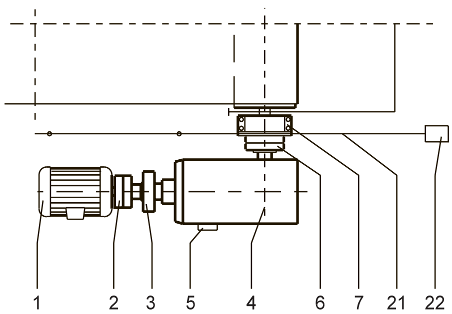
Construction

An installation consists of a drive system, a take-up system, additional components and the principal item, the conveyor belt. In addition to the conventional conveyor there are other conveyor system developments in which the belt is the conveying element.

Basic Sketch





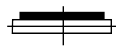


Components

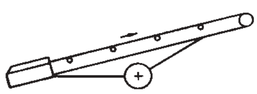
1	Motor	18	Belt Run Counter
2	Motor Coupling	19	Off Track Control
3	Brake	20	Belt Steering Idler
4	Drive Transmission	21	Pull Wire
5	Anti Runback	22	Emergency Switch
6	Drive Coupling	23	Conveyor Belt
7	Pulley Bearings	24	Brush Roller
8	Drive Pulley	25	Scraper
9	Tail Pulley	26	Plough
10	Deflection or Snub Pulley	27	Decking Plate
11	Impact Idler Garland	28	Cowl (Head Guard)
12	Carrying Side Idler	29	Baffle Bar
13	Return Side Idler	30	Delivery Chute
14	Guide Roller	31	Chute Lining
15	Counter Weight Take-up	32	Skirt Board
16	Screw Take-up	33	Upper Belt Location
17	Take-up Weight	34	Lower Belt Location

Explanation

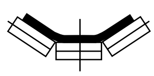
Belt conveyor-fixed position



Flat conveying



Mobile conveyor
With wheels



Troughed conveying

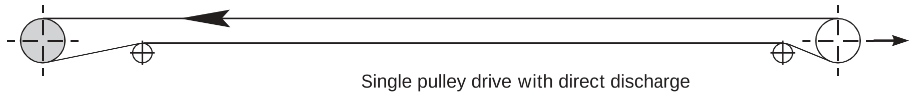
Without wheels

Drive Systems

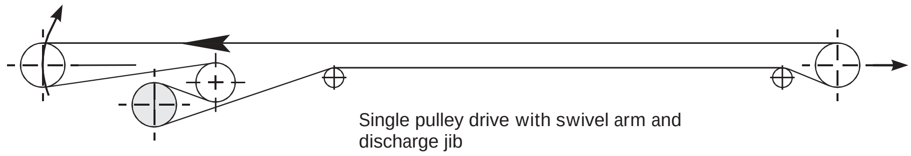
A drive system consists of all components that provide for driving, start-up and braking forces. The transmission of traction power from the drive pulley is dependent upon the following factors:

- **Angle of wrap α the belt makes on the drive pulley**
- **Friction coefficient μ between belt and drive pulley**
- **Pre Tension T_v.**

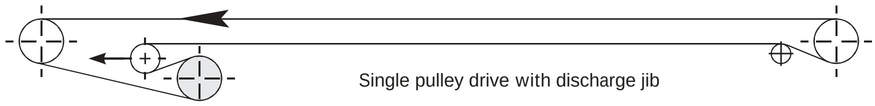
Pulley Arrangement and Belt Path



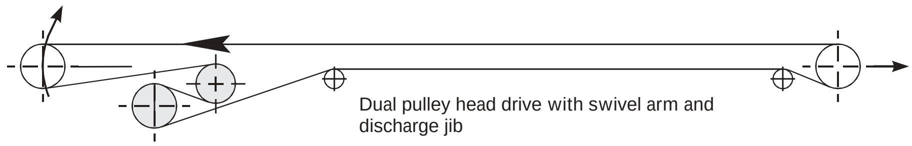
Single pulley drive with direct discharge



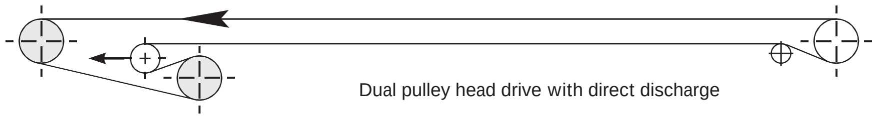
Single pulley drive with swivel arm and discharge jib



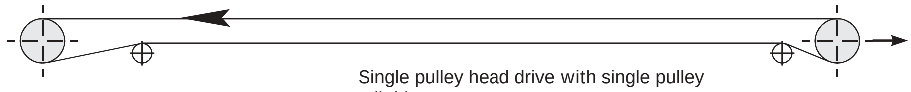
Single pulley drive with discharge jib



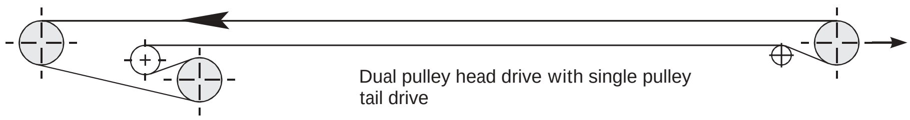
Dual pulley head drive with swivel arm and discharge jib



Dual pulley head drive with direct discharge



Single pulley head drive with single pulley tail drive

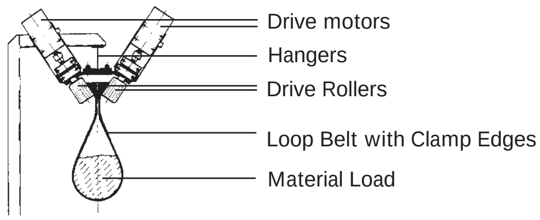


Dual pulley head drive with single pulley tail drive

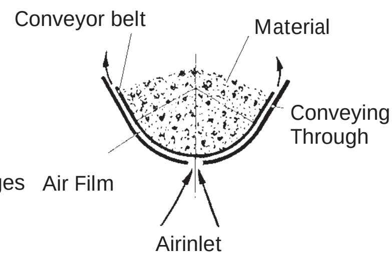
Conveyor Systems

The above sketches of drive systems and belt paths are of the classical and most frequently used types. Additionally there are a number of new proven developments or those which are still undergoing trials, for instance, tube conveyors, piggy-back conveyors, clamp conveyors, aerobelt conveyors, loop belt conveyors and others.

Here are some examples:

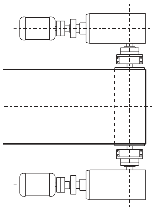


Loop Belt Conveyor
(FMW System)



Aerobelt Conveyor
System

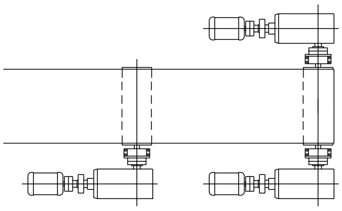
Single Pulley Head Drive



The single pulley head drive is the most common and preferred drive. For light duty applications the **Motorized Conveyor Pulley** or **V-rope Drive** is often used.

Under normal circumstances the drive unit comprises motor, coupling and gear box located at the side of the drive pulley and connected by means of a flexible coupling, flanged coupling or extension gear box. With higher duty applications the drive units can be located on both sides of the drive pulley.

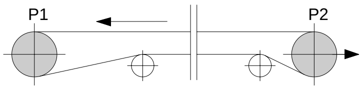
Dual Pulley Head Drive



For the higher duty applications the dual pulley drive is used which enables an increase in angle of wrap and traction transmission. 2 to 4 drives are possible which as a rule for reasons of standardization, would be of the same size.

By distributing the drive power in the ratio of 1:1 or 1/3 : 2/3, the transmission capability of the first pulley would not be fully utilized. All motors take approximately the same work load. Almost the same size work load can be achieved by selecting a fluid coupling whose slip characteristic can be modified by adjusting the volume of fluid in the working circuit. If slip ring motors are used this can be achieved by adjusting the slip resistance.

Head and Tail Drive



The head and tail drive may be used for relatively long installations, reversible drives or where high return side resistance can occur. Start-up and braking is made easier on long installations. The tail drive overcomes the resistances to motion on the return side run, the pre-tension at the head drive can be increased.

General Criteria

The choice of drive system depends on the total working duty, the belt characteristics and general operating conditions. With the higher duties the drive pulley is driven on both sides. The pulley shaft is then symmetrically used and able to take a higher loading.

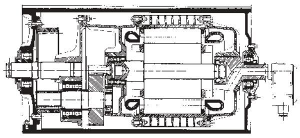
With multi-pulley drives the belt should be so reeved that **the pulley side is leading over the drive pulley.** In situations of high belt stress additional deflection of the belt should be avoided thereby preventing unnecessary **bending stresses.**

One should be particularly aware of **dirt build-up.** Prompt cleaning can avoid down time and repair costs.

Drive Components

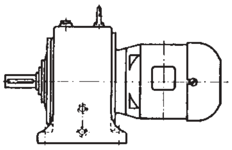
In principle there are three possibilities for conveyor belt driving systems.

Pulley Motor



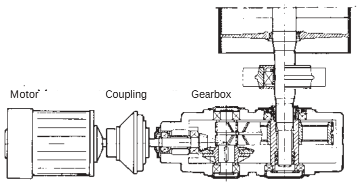
For use on small to medium duty applications. These are distinguished by their compact construction. As a rule in the range of approximately 20 kW to a maximum of 100 kW where pulley diameters up to 1400 mm are used and **belt speeds** in the range 0.02 to 5 m/s.
The heat loss is minimal particularly with lagged pulleys.

Geared Motor



Utilized up to approximately 30 kW, special applications up to approximately 120 kW. Possibilities are spur gearing, angle drive, worm gearing. Compact and frequently used for light duty applications.
The geared motor is connected to the drive drum by a flexible or fixed coupling.

Motor Gears Coupling



Drive units comprising these three components are used the most. Typical are work loads up to 600 kW with maximums up to approximately 1500 kW.

The advantage of this arrangement is the good accessibility and interchange-ability of components and favourable spares inventory etc.

In most cases, especially on large installations, coupling and brake are built into the drive unit.

Drive Motor

As a rule a belt conveyor has to operate under variable load conditions. The drive capability has to take account of these changes, also must run at con-stant revolutions and even speed.

Squirrel Cage Motor

This type of motor is simple to build, is robust and economical. The high tor-que when starting can be minimised and adjusted by means of flexible coup-lings, fluid couplings and slip couplings. Small installations up to approximate-ly 10-12 kW can run without intermediate coupling. The high stresses imposed upon the system during loaded start-up can be limited by step by step increases in resistance or by the Star Delta reduced voltage system.

Slip Ring Motor

With this type of motor the starting torque can be reduced by the increase of resistances within the electrical circuit.

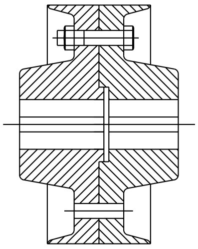
The slip ring motor is used on large installations if a quasi stationary gentle start-up is required.

Coupling Types

When starting up a belt conveyor, for a short time forces result which are higher than those which occur during normal running conditions e.g. start-up and acceleration forces. The motor in contrast creates by far the greater start-up torque. The difference between the load torque and the start-up torque of the motor, is the acceleration torque.

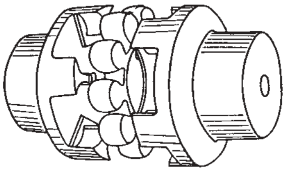
Depending on the size and type of motor, a gentle start-up is provided when a flexible or hydraulic coupling is used.

Rigid Coupling



Rigid couplings are used only on small installations up to a maximum of 30 kW and at slow speeds.

Flexible Couplings



Flexible couplings of diverse construction are already built into 16-20 kW units.

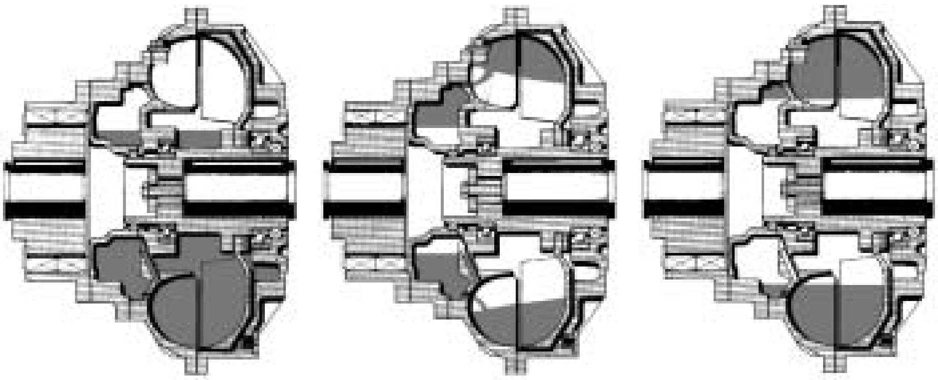
Centrifugal Couplings

Centrifugal and **magnetic type** couplings with controlled torque are hardly ever used because of their high cost and are no longer a significantly used item for belt start-up.

Hydraulic Coupling

Fluid couplings or **hydraulic couplings** are often used for larger drives in combination with squirrel cage motors. They permit a load free acceleration of the motor and consequently with increasing oil fill, provide a gentle quasi steady state start-up of the belt conveyor. The maximum torque occurring during the start-up process is restricted to lowest possible level. The conveyor belt and splice joints are relieved and conserved.

The principle of
a Turbo coupling



Empty at Rest | Full at Start-up | Full Working

(Principle of the VOITH Turbo coupling with delay chamber)

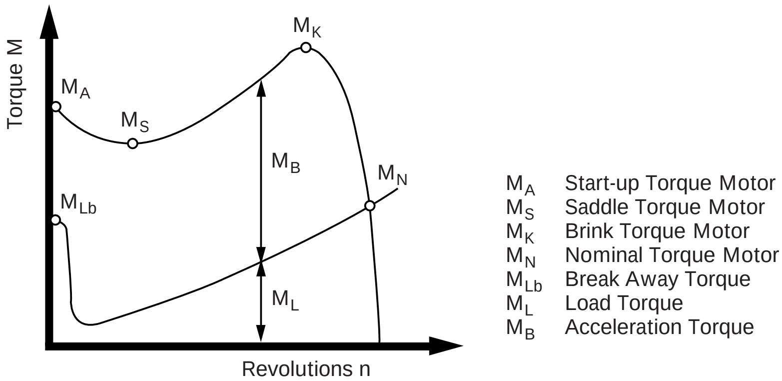
Start-up Procedure

The electric motor produces a variable torque with speed of rotation which is illustrated below by line M. The belt conveyor as a working unit opposes torque M_M at the same rotational speed set against torque M_L.

Squirrel Cage Motor with Fixed Couplings

At the moment of switch on, the electric motor delivers the start-up torque M_A. When this happens, the belt conveyor demands the break away torque M_{LB}. As rotation increases the motor torque declines to saddle torque M_S and then increases to the brink torque M_K. Thereafter the torque at the nominal speed of rotation reduces to the nominal torque of the motor M_N. The load increases steadily.

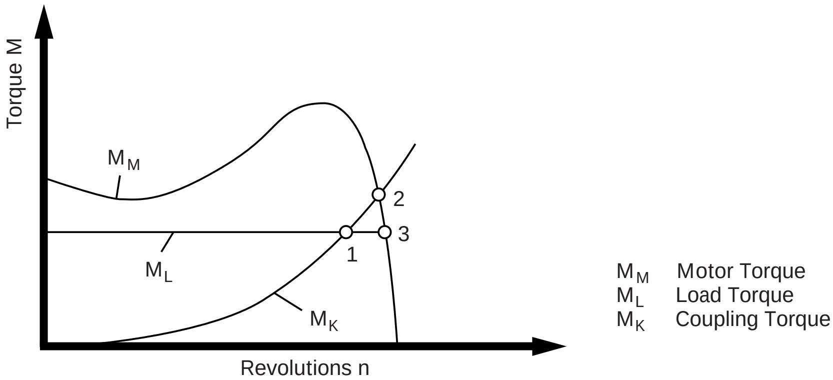
The difference between the motor torque M_M and the load torque M_L is the acceleration torque M_B up to the torque intersection of both lines.



Squirrel Cage Motor with Hydraulic Coupling

The start-up of the electric motor occurs practically load free compared with the coupling torque M_K up to the cut-off point 1 of the graph line M_K and the load graph line M_L.

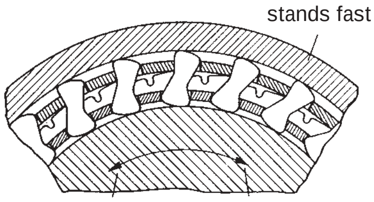
The motor remains at point 2 and accelerates the belt conveyor up to synchronous speed. Thereafter the motor, the coupling and the belt conveyor reach the drive speed of rotation at point 3. This type of coupling is only effective on a high speed shaft.



The start-up torque is reached by a continuous filling of the coupling from 0 to break away torque within a period of approximately 10 to 20 seconds. Thus the belt is slowly tensioned and longitudinal vibrations are avoided.

Holdback

A holdback device is necessary if the force due to the loaded incline of a conveyor F_{St} exceeds that of the peripheral force F_H stemming from conveying the load over the horizontal distance. This is the case with all steeply inclined conveyors and with those having a decline of approximately 6° to 10°.



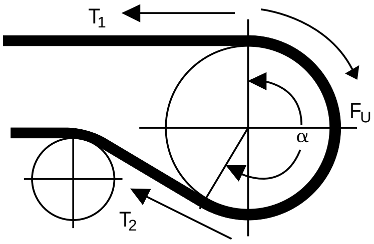
$F_{St} > F_H$	(N)
$F_{St} = H * g * m'_L$	(N)
$F_H = f * L * g (m'_L + m'_R)$	(N)

F_{St}	(N)	Incline Force
F_H	(N)	Force due to Horizontal Conveying
m'_L	(Kg/m)	Mass of Load Stream on Belt
m'_R	(Kg/m)	Mass of Rotating Carrying Idler Rollers
L	(m)	Conveyor Length
H	(m)	Height Difference
f	(-)	Artificial Friction Factor
g	(m/s^2)	Acceleration due to gravity

During installation a check should be made as to whether the direction of pulley rotation corresponds with the free wheeling direction of rotation and that reverse travel of the belt is no longer possible.

SUPERFORT Belt in Service
Conveying Overburden

Force Transmission



The belt tensions required for frictional transmission of peripheral force F_U are:

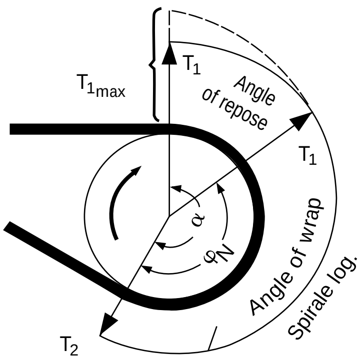
T_1 = Entry side force = Tight side tension
T_2 = Leaving side force = Slack side tension

Because of the friction coefficient between belt and pulley, T_1 is greater than T_2. The difference between both forces is peripheral force F_U.

$F_U = T_1 - T_2$ or $T_1 = F_U + T_2$

On braking the forces are reversed.

Eytelwein Limitations



In the extreme case of friction cut off that is, if the slip limit is about to be reached and the angle of wrap α is fully utilized, then at the limit of slip

$\frac{T_1}{T_2} \leq e^{\mu\alpha}$ or $T_1 \leq T_2 * e^{\mu\alpha}$

T_1 decreases along the angle of wrap α to the value of T_2 over a logarithmic spiral.

From the formulae for peripheral force F_U and T_1 the following relationships can be derived:

$$T_1 = F_U * \left(1 + \frac{1}{e^{\mu\alpha} - 1}\right) = F_U * c_1$$

$$T_2 = F_U * \left(\frac{1}{e^{\mu\alpha} - 1}\right) = F_U * c_2$$

c_1 and c_2 are the drive factors.

Slip

If the peripheral force F_U is greater than the transmission capability according to the Eytelwein theory borderline conditions, then the drive pulley over runs and slip occurs.

Creep

During the reduction of tension from T_1 to T_2 along the service arc, a decrease in the belt stretch occurs. It is not proportional to the time it occurs at a slower rate. Creep remains on the pulley circumference and often creates a whistling sound.

Entry side speed = pulley peripheral speed.
Leaving side speed < entry side speed.

Friction Coefficient μ

The factors determining the transmission of peripheral force F_u are the friction coefficient, the angle of wrap μ and the pretension T_2 e.g. T_v.

The number of drive pulleys, the pretension force as well as various other **installation components** are dependent upon the friction coefficient μ. In order to assess the friction coefficient, full knowledge of the operation is desirable. In practice the friction coefficient varies within limits which are dependent upon:

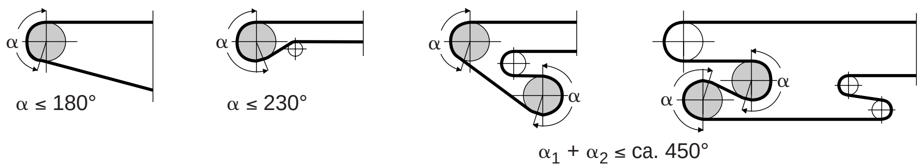
- Surface condition of the pulley
- Lubricating material such as water or load slime between belt and pulley
- Temperature
- Surface pressure
- Slip and creep speed

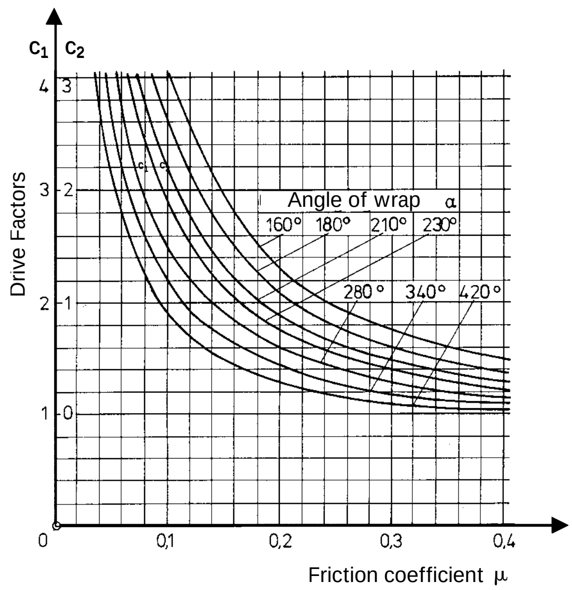
The friction coefficient declines with increasing surface pressure and increases with increasing speed of creep for instance at start-up.

Operating Condition	Pulley Surface			
	Plain Steel (smooth)	Polyurethane lagging (grooved)	Rubber lagging (grooved)	Ceramic lagging (porous)
Dry	0.35 to 0.4	0.35 to 0.4	0.4 to 0.45	0.4 to 0.45
Wet (Clean)	0.1	0.35	0.35	0.35 to 0.4
Wet (dirty mud, clay)	0.05 to 0.1	0.2	0.25 - 0.3	0.35

Angle of Wrap α

The angle of wrap can be increased by means of a snub pulley to a maximum of 230°. By using two drive pulleys an angle of up to approximately 450° can be achieved.





From the graph the following relationships can be derived.

- An increase in value over 0.35 does not result in any great advantage from c_1 and c_2.
- With friction coefficient μ value less than 0.3 an increase of the angle α is an improvement from c_1 and c_2.
- Values μ and α take into account the values of μ and α with the same value of drive factors c_1 or c_2.

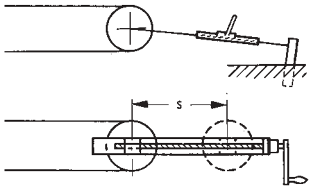
General

Belt conveyors are equipped with **take-up (tension) devices.** These are necessary to make it possible for the transmission of force on the drive pulley or to accommodate changes in the length of the belt as the load changes and to provide a smooth start-up of the installation. The choice of take-up system depends on the **general conditions** of the installation, the **elongation characteristics of the belt,** the **start up behaviour, climate** and perhaps the conveying distance and the inclination of the installation.

Depending on the application one differentiates between:

- Fixed take-up
- Moveable take-up with a constant pre-tension provided by a gravity weight or with a pre-tension regulated by motor

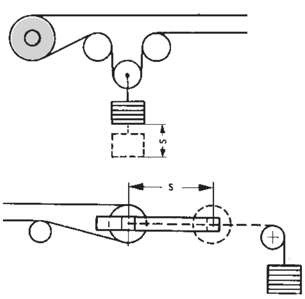
Fixed Take-up Device



This simple take-up device is used for relatively short conveyor lengths or with belts having a low elongation such as steelcord. After tensioning the belt, the length stays constant but the belt pull force changes according to the change in load through permanent or elastic stretch.

The level of pre-tension force can be determined practically by tensioning the belt until the forces react at all parts of the installation. The belt is permanently tensioned sufficiently high as is necessary for full load conditions. It is possible to ascertain and actually adjust the tension by means of a sensor.

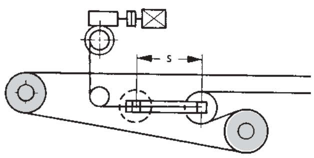
Gravity Weight



On longer installations a moveable gravity weight tension device is used. With this a constant pre-tension is achieved at all parts of the installation. The length changes of the belt are evened out in the take-up travel. It must be determined in such a way as to accommodate those changes i.e. additionally to permanent belt elongation.

A value for take-up travel with textile carcase belts is approximately 1.5% based on conveyor centre to centre distance and with steelcord belts approximately 0.3%.

Regulated Take-up Winch



With especially long installations or to reduce start-up vibrations of belt, so called regulated take-up winches are used. By means of electric switching a pre-run of the take-up winch, before the belt start-up, is achieved. During start-up the pre-tension is kept above nominal value. After reaching the steady running condition the winch is adjusted so as to provide the nominal value of pre-tension.

For the operation of complex belt conveyor installations a range of **safety devices** were developed. They serve to **prevent accidents,** to **guard the conveyor belt** and **to run and automize** the complete installation.

Belt tension detection

This device detects the belt tension changes during different working conditions such as starting, braking and load variations.

A tensiometer measures the changing belt tensions. As soon as the defined limits of the upper and lower belts tensions are exceeded a signal is transmitted to the take-up unit.

Off track detection

An off-track of the belt can be limited or corrected by means of a number of mechanical devices. Another possibility is to track the belt with light beams. If a deviation beyond the defined limits is detected a motorized steering system is set into operation. If the off-track is not adjusted the installation is brought to a halt.

Chute detection

With the help of this device the overloading of a transfer point is registered.
A contact probe is suspended in the chute and adjusted to the required material flow. With an overload, contact is made with the probe and the installation is switched off.

Belt Slip detection

This device detects the force transmission between belt and drive pulley. Too great or too long a duration of slip leads to overheating of the pulley surface which can lead to fire. Slip can occur more often at start up as well as overload conditions or when the pre-tension is too low.

Longitudinal Slitting Safety Device

Slitting open of conveyor belts is a relatively frequent cause of damage which leads to prolonged interruptions and incurs great costs. In most cases it stems from sharp edged pieces of material being trapped in the transfer chute.

To avoid such damage the following remedies are possible.

- Textile or Steel breaker ply embedded in the cover rubber
- Bunched steel cords are embedded in the carrying side cover set at fixed spacing.
- Rip detection loops at approximately 50 to 100m spacings vulcanised into the belt.
- Carcase with a high resistance to longitudinal slittings such as Dunlop FERROFLEX belt or DUNLOPLAST belt.

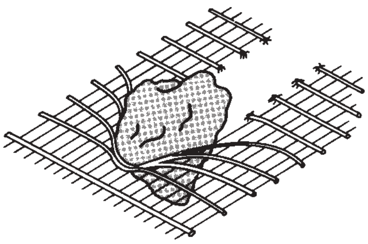


Illustration of the transverse reinforcement in a FERROFLEX Belt

General

Belt conveyors when operating need to be cleaned constantly depending on the type and characteristics of the load residues which stick to the carrying side of the belt. The return idlers get dirty which leads to further encrusting of idlers and pulleys. The consequences are, amongst others:

- **loss of load**
- **soiling of the installation**
- **damage to the belt**
- **off tracking of the belt**
- **damage to idlers**

The belt cleaning requires special measures having regard to:

- **material characteristics such as, dry, damp, sticky, granulated**
- **degree of cleaning, rough to `clinically' clean**

Basic rules

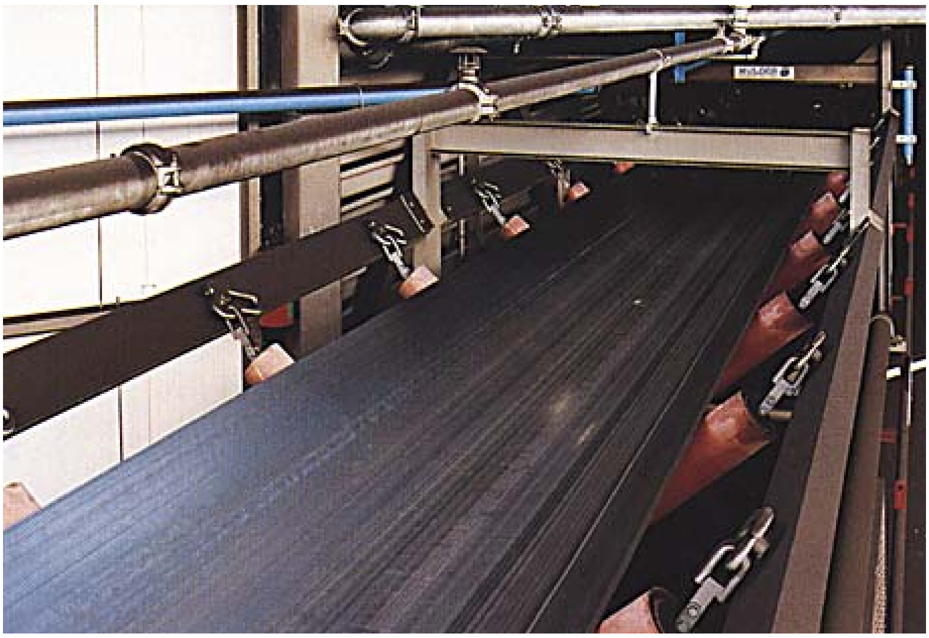
The **material from which the scraper** is made has to be in accord with the type of belt and the material to be scraped off.

The following need to be observed:

- the **hardness of the belt and scraper;**
  In general the material from which the scraper is made should not be harder than the belt surface. The scraper should be designed in such a way that it makes contact with the biggest possible area of the belt, perhaps a double or staggered scraper.

- the **contact pressure of the scraper;**
  An excessive contact pressure increases energy consumption and wear and tear of the belt. A small gap between the scraper and belt surface provides a sufficient cleaning effect with most bulk loads, causing minimum wear and tear. Sliding friction becomes rolling friction.

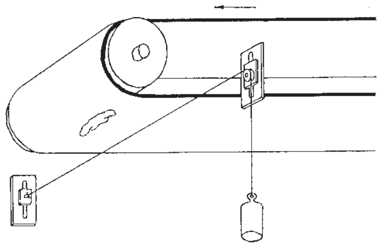
In nearly every case the most effective and economic solution has to be established, possibly by trial and error. Several well proven systems are listed as follows.



TRIOFLEX Belt in operation on Garland Idlers

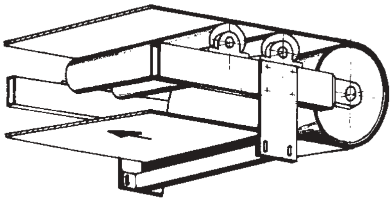
Piano Wire Scraper

Steel cords tensioned transversely over the belt carrying side behind point of discharge for coarse cleaning of sticky material such as loam, clay and the like.



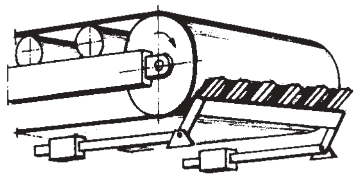
Transverse Scraper

Fixed or moveable also possible as a double bladed scraper.



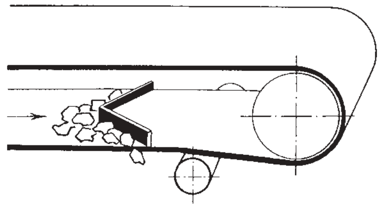
Fan Scraper

Used for moderately sticky material
Big contact area.



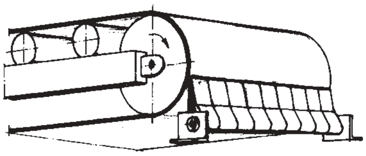
Plough Scraper

For cleaning the under side of the belt often located before tail pulleys. In special cases it can be used in double bladed form. Also prevents material from running into the tail pulley.



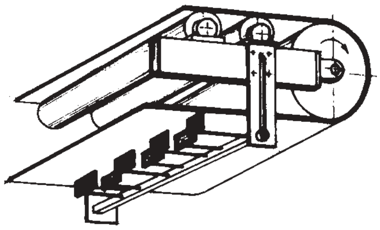
Scratch Action Scraper

Used for rough cleaning immediately at the discharge pulley.
Adjustable versions are available.



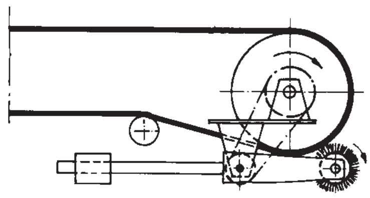
Staggered Scraper

Used for nearly all non-sticky materials. Can be adjusted depending on the load and desired degree of cleaning.



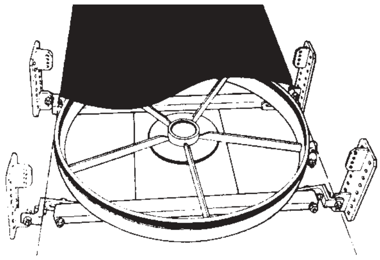
Rotary Scraper

Rotating, partly self driven brushes with rubber or nylon bristles. Not suitable for sticky materials such as clay or loam.



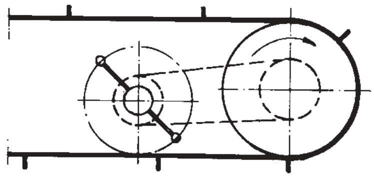
Rotary Scraper

Horizontally rotating scraper located on return run. Ring scraper in slightly tilted position - belt driven.



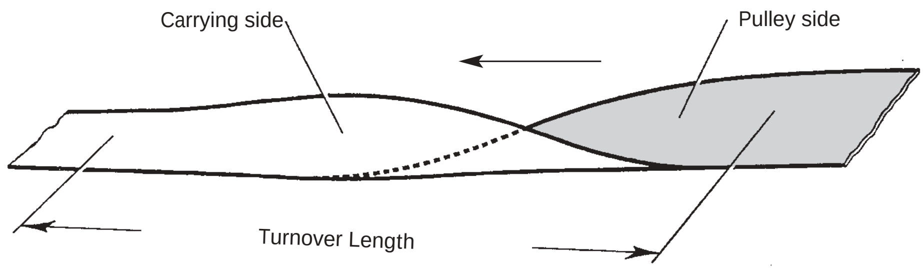
Rapping Roller

Situated on the belt running side located in the region of the discharge hopper. For belts with a profiled carrying side and steep inclined belts.



Belt turnover

Located at inaccessible areas such as bridges, tunnels or over-passes. The belt is turned over on the return side so that the dirty carrying side is uppermost.



High Pressure Water

Good cleaning method where good water supply and drainage are available.

General

It is important to know the **exact make-up** and **material component** of the load when constructing a conveyor installation. The choice of conveyor belt is determined by the **physical and chemical properties of the materials** and possibly by **safety regulations** and demands.

Bulk Loads

Dusty, granular or lumps. Loads for instance, stone, earth, sand, grain, cement. They are defined by their physical characteristics such as density, granulometry, moisture, oil and fat content, pH value, abrasiveness, angle of surcharge etc.

Unit Loads

Piece loads comprise such as boxes sacks, blocks etc., and are defined by shape, measurement and weight.

Bulk Density ρ

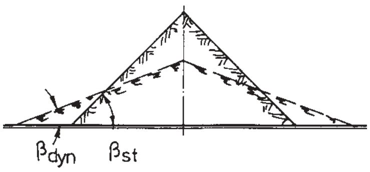
It is the quotient of the mass and volume V of the bulk load

$$\rho = \frac{m}{V} \quad (t/m^3)$$

Angle of repose Surcharge β

If a load is loosely poured a **static angle of slope β_{st}** will form. If the under layer is moved as when being transported on a conveyor belt, the angle of surcharge **β_{dyn}** is formed. The particles of the bulk load interact and the lower the internal friction of the material the lower the surcharge angle β_{dyn}.

Approximately:

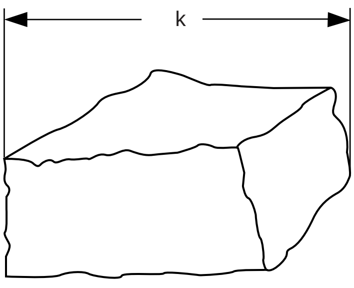


$$\beta_{dyn} = (0.5 - 0.9) * \beta_{st}$$

The surcharge angle **is only to a degree related to material size.** It depends on the friction between material and belt, how the material is loaded and the geometry of the conveyor installation. The tabulated values in the Appendix are approximate values. If exact values are necessary they have to be determined by practical trials.

Granulometry k

As a **measure** of **granulometry** or **lump size** the maximum diagonal corner to corner dimensions k are used.
Whichever is the predominant granulation or lump size decides whether the bulk load is sized or unsized.



Measurement

Sized load	$k_{max} / k_{min} \leq 2.5$
Unsized load	$k_{max} / k_{min} > 2.5$
Granulometry	$k = 0.5 (k_{max} + k_{min})$

Bulk Load Definition	Granulometry (mm)
Dust	0.5
Granular	0.5 - 10
Lumps	10 - 200
Large Lumps	> 200

Temperature

It is important to have as near accurate as possible knowledge of the load **temperature** and **ambient conditions** in order to select the optimum cover quality and in certain circumstances the layout of particular installation components.

Further influencing factors are:

- **Granulometry of the material** and hence the **contact density** made with the belt.
- **The speed of the belt** and therefore the **heating up** and **cooling off times.**
- **Whether installation is open or enclosed.**

Moisture

The moisture content of the material has an influence on the surcharge angle β_{dyn}, the friction value between material and belt and thus the maximum angle of inclination of an installation. It is difficult to give an exact assessment and if necessary practical trials need be conducted. The moisture content is measured in percentage.

Chemical Characteristics

A sound knowledge of the chemical characteristics of the load is necessary when selecting the cover quality and also the carcase reinforcement. Important are the proportions of oils and fats (mineral and vegetable) and acidity.

The temperature of the chemical material as well as the proportion of acidity influences the level of attack made on the belt cover.

pH Value

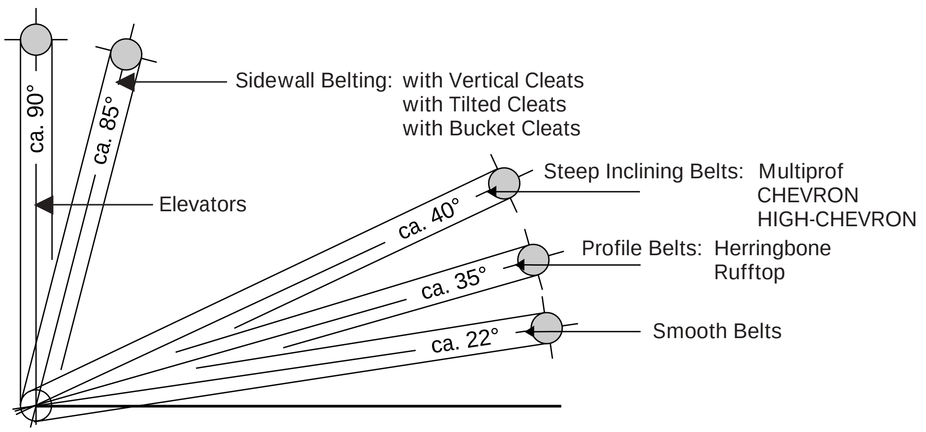
The pH value is the concentration of hydrogen ions which are present in a solution e.g. the negative exponent to base 10 of the concentration (hydrogen exponent).

pH > 7 ▲ alkalis
pH = 7 ┼ neutral
pH < 7 ▼ acid

The neutral point is pH = 7
Under 7 indicates acidity
Over 7 indicates alkalinity (bases)
The pH value indicates the degree of acidity or alkalinity and can be important in selecting the cover quality.

Angle of Inclination

The maximum angle of inclination of a belt conveyor depends on the **friction value** between material and belt and **the form of material.** Large lump and moist material decrease the angle of inclination. The method of loading such as direction and rate of feeding are also important criteria. For most bulk loads and belts with a smooth carrying surface, **the limit angle** lies between 18° and 20°. For steeper inclinations up to 90°, profiled belts, belts with cleats or elevator belts are used.



General

The conveyor belt is the most important element of a belt conveyor installation. It has to be capable of doing numerous tasks:

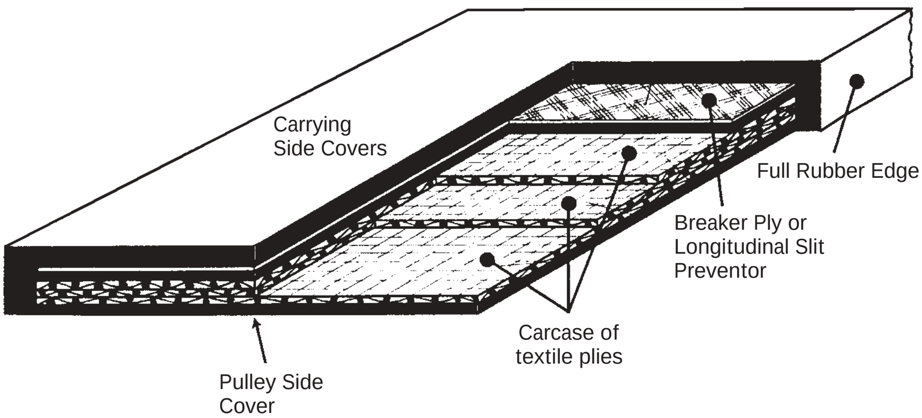
- Absorb the stresses developed at drive start-up.
- Transport the load.
- Absorb the impact energy at the loading point.
- Withstand temperature and chemical effects (heat, oil and fat containing materials, acidity etc.).
- Meet safety requirements (flame resistant, antistatic etc.).

Belt construction

The conveyor belt consists of the following components:

- **Carcase** consisting of textile plies, steel weave or steel cord.
- **Covers** in different qualities of rubber or PVC.
- **Additional components** (as required) such as edge protection, impact protection, longitudinal slitting prevention etc.
- **Special construction elements** like profiles on steep incline belts, cleats or corrugated edges etc.

All items mentioned above should be considered carefully. The selection of the belt specification depends on the application.

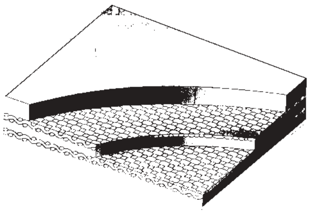


Schematic Construction of a Plied Belt

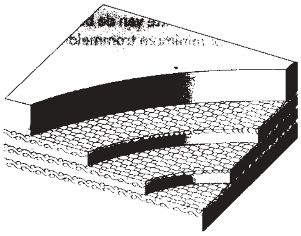
Carcase

The carcase can be made from various materials and in different constructions. The most frequently used are textile ply carcases and Steel Cord.

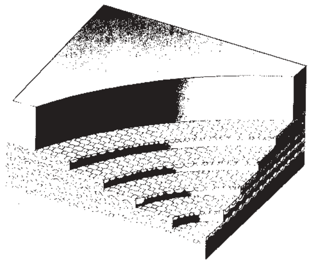
Carcase with one or more textile plies (up to a maximum of 6 ply)



DUNLOFLEX
2 Ply



TRIOFLEX
3 Ply



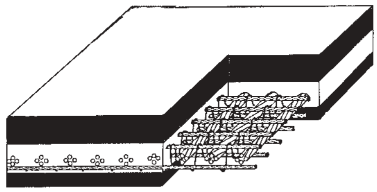
SUPERFORT
Multi Ply

Carcase of the Solid woven type. Monoply belts

DUNLOPLAST

Dunloplast Belts have a PVC impregnated textile carcase of monoply construction. Depending on tensile strength and duty the carcase fibres are in polyester, polyamide or aramid.

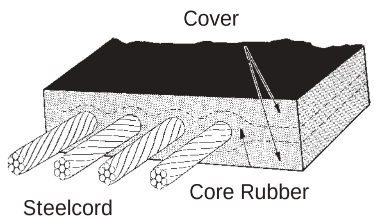
Carcase of the Steel weave type



FERROFLEX

The transmission of force is by means of the longitudinal steelcords laid next to one another in the same plain. Above this carcase is a transverse layer also of steel which is held in place by a polyamide binder cord.

Carcase with Steel Cords. ST-belts



SILVERCORD

With this belt force is transmitted via steel cords of the appropriate strength. The cords are transversely bound together by an intermediate layer of rubber only.
Transverse Elements serve to prevent impact damage or longitudinal slitting.

Material

For textile, solid woven or steel reinforced types the principal materials used are as follows:

Symbol	Ply Materials	
B	Cotton	(Natural fibre)
P	Polyamide	(Synthetic fibre)
E	Polyester	(Synthetic fibre)
EP	Polyester-Polyamide	(Synthetic fibre)
D	Aramid	(Synthetic fibre)
F	Steel Weave	(Ferroflex)
ST	Steel Cord	(ST belt)

Of the textile ply types the **wholly synthetic plies** have proved to be the best over the years e.g. **polyester (E)** in the **warp (longitudinal direction)** and **polyamide (P) in the weft (transverse direction).** The abbreviation of this ply construction is called EP.

Carcases from **Aramid (D)** are in development and are high tensile and low elongation carcases.
Steel Cord belts have very low elongation and are used predominantly on long haul installations.

Evaluation of various Material Properties

Characteristics	Key and evaluation of the plies						
	B	P	E	EP	D	F	ST
Tensile Strength	–	++	++	++	+++	++	+++
Adhesion	–	++	++	++	++	++	+++
Elongation	–	–	++	++	+++	+++	+++
Moisture Resistance	– –	+	++	++	++	+	++
Impact Resistance	–	+	+	+	+	++	+++

– – = bad – = medium + = good ++ = very good +++ = excellent

Covers

The carcase is protected against outside influences by the covers which are normally made out of either rubber or PVC. The carrying side cover should not be more than 3 times thicker than the running side cover.

Cover Thickness Ratio

Carrying side : Running Side ≈ 3:1

Belt Edges

With wholly synthetic and rot resistant plies it is not necessary to have full rubber edge protection, it is sufficient to have heat sealed cut edges. Exceptions are belts requiring special qualities such as oil and fat resistance.

Grades

The basic grades and principal properties are in accordance with ISO and DIN.

DIN 22102 (April 1991)					
Cover grade		W	X	Y	Z
Tensile strength	N/mm^2 min.	18	25	20	15
Breaking elongation	%	400	450	400	350
Abrasion loss	mm^3 max.	90	120	150	250
ISO 10247 (November 1990)					
Cover grade		H	D	L	
Tensile strength	N/mm^2 min.	24	28	15	
Breaking elongation	%	450	400	350	
Abrasion loss	mm^3	120	100	200	

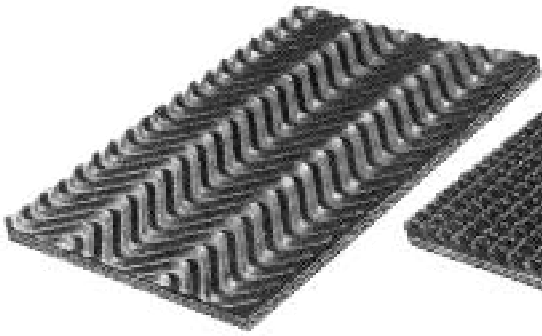
Various other special grades exclude laid down mechanical or other values.

Special Qualities

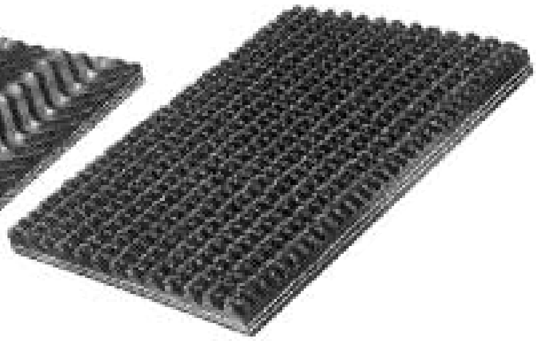
Characteristics	Type
Flame Resistant	F
Anti Static	E
Flame Resistant, Anti Static	S or K
Heat Resistant	T
Low Temperature Resistant	R
Oil and Fat Resistant	G
Food Stuff	A
Chemical Products	C

Cover Surface

The carrying side surface depends on the load, the inclination of the installation or depending on the use of the belt, smooth, profiled, cleated and with corrugated edges.

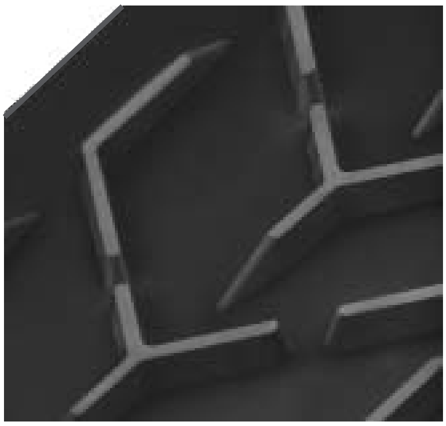


Herringbone Profile



Rufftop Profile

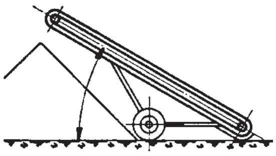
Multiprof Profile

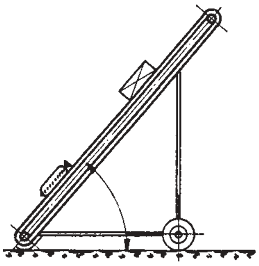


HIGH-CHEVRON Profile

Gradient values

Cover Surface	Belt Type	Max. Gradient	Application
Smooth	Normal	18° - 20°	Unit and bulk loads all types
Profiled	Fishbone Rufftop	up to 35°	Piece and bulk loads
Steep conveyor profile	Steep conveyor CHEVRON HIGH-CHEVRON Multiprof	up to 40°	Bulk loads (non-sticky) Piece loads (sacks)
T-cleats with or without corrugated edges	Belts with corrugated side walls, with or without T-Cleats	up to 90°	Piece and bulk loads
With Steel or rubber buckets attached	Elevator belts	80° - 90°	Bulk loads all types





Load	Gradient angle for conveyor belt with	
	Rufftop profile	Herringbone profile
Wooden crates	40°	30°
Bricks	40°	30°
Paper Sacks	35°	30°
Jute Sacks	35°	35°
Plastic Boxes-dry	40°	30°
Plastic Boxes-wet	25°	25°

Carrying side Cover thickness

The thickness of the carrying side cover depends upon the nature of the load and loading conditions (type of load, gradient, height of fall etc.).

Cover Thickness Gauges

Conveyor Load/Duty	Cover Thickness (mm)	
	Carrying Side	Pulley Side
Light package Conveying	2	2
Gravel, Earth, Potash etc.	2 - 4	2 - 3
Ore, Ballast, Coal	4 - 8	2 - 3
Slag	4 - 8	2 - 3
Coarse ballast, coarse Ore	8 - 12	3 - 5
large lump Coal	8 - 12	3 - 5

Dunlop-Enerka Cover Qualities

Dunlop-Enerka Quality	Quality		Temperature (°C)			Basic base	Characteristics Application
	DIN	ISO.	min.	duration	max.		
RA	Y (N)		-30°	80°	100°	SBR	Abrasion resistant, for normal service conditions encountered in carrying bulk and aggregate materials.
RE	X (M)	H	-40°	80°	90°	NR	Extra abrasion resistant and cut resistant for heavy duty service conditions (sharp materials and adverse loading conditions).
RS	W	D	-30°	80°	90°	NR/SBR	Super abrasion resistant, for heaviest service conditions, abrasive materials with a large proportion of fines.
BETA-HETE	T		-20°	150°	170°	SBR	Heat resistant, for materials at moderate temperatures.
STAR-HETE	T		-20°	180°	220°	IIR	Very heat resistant, for materials with controlled high temperatures.
DELTA-HETE	T		-20°	200°	400°	EPDM	Very heat resisant, for heavy duty service conditions including abrasive materials, at temperatures up to 400°C (or more) at times, e.g. some isolated burning materials or red-hot cores, such as embers, sinter, coke etc.
ROS	G		-20°	80°	120°	NBR	Oil and grease resistant, for oily materials on mineral oil base.
ROM	G		-30°	80°	90°	SBR/NBR	Oil and grease resistant, for vegetable oils and animal greases.
MORS	G		-20°	80°	90°	SBR/NBR	Oil and grease resistant, for vegetable oils and animal greases, and for heavy service conditions of the cover.
BV	S/K		-30°	80°	90°	CR/SBR	Fire resistant for conveyance of materials with fire and explosion danger, such as fertilizer
BVO	S/G		-20°	80°	90°	CR/NBR	Fire and oil resistant for conveyance of oily materials (vegetable oils and animal greases), e.g. fertilizer, cereals, derivates etc.

The values apply to the materials temperatures
Other qualities for special applications are available on request.

Basic materials

Code	Rubber type
NR	Natural Rubber
SBR	Styrene-Butadiene Rubber
NBR	Nitrile Rubber
IIR	Butyl Rubber
EPDM	Ethylene-Propylene-Diene Rubber
CR	Chloroprene Rubber

Basic Materials	Characteristics and areas of application
Natural Rubber NR	Natural rubber, because of its special properties is a good basic material for belt cover rubbers. **Technical Characteristics** • Very good tensile strength and elongation • High heat resistance and elasticity • High tearing and shear strength • Good abrasion resistance characteristics **Temperature Stability** Generally stable within the temperature range of -30° to + 80°C. With special rubber compounding a widening of this range may be achieved from -40°C to + 100°C. **Chemical Stability** Resistant to water, alcohol, acetone, dilute acids and alkalis - limited resistance to concentrated acids and alkalis where compounding and service temperatures are major consideration. **Special characteristics** With special compounding natural rubber based mixes can be made antistatic and flame resistant. By adding antiozonants a substantial protection against harsh temperature effects, sunlight and ambient weather conditions can be achieved. **Scope** Everywhere where high physical properties are called for and the chemical and temperature demands are not excessive.
Synthetic Rubber SBR	SBR is a synthetic polymerisation product consisting of styrene and butadiene whose characteristics are similar to natural rubber. Tensile strength and cut resistance are good. Abrasion, heat and ozone resistances are better than natural rubber.
Nitrile Rubber NBR	NBR is a copolymer of butadiene and acrylonitrile. It is not resistant to Ketones, esters aromatics and hydrocarbons. The physical property values are slightly lower than those of natural rubber. The temperature operating range can be controlled between -40°C to +120°C. NBR is relatively abrasion resistant, resistant to ageing and is used for oil and fat resistant belt covers.
Butyl Rubber IIR	Butyl rubber is a polymerisation product of Isobutylene and Isoprene. It has a very good ozone and temperature resistance. In addition to a very good resistance to ageing, depending on compounding, it is able to withstand temperatures of -30°C to + 150°C. It has a limited resistance to acids and alkalis, animal and vegetable fats. Butyl rubber is used mainly for heat resistant conveyor belting.
Ethylene Propylene Rubber EPDM	EPDM is temperature resistant similar to Butyl but with a considerably higher resistance to wear and tear. Additionally EPDM has a better ozone resistance than all other basic polymers.

Chloroprene Rubber CR

CR is a Synthetic polymerisation product of Chlorobutadiene. The mechanical properties are similar to natural rubber but significantly better in respect of ozone and oil resistance. The chlorine in Chloroprene gives the product a high degree of flame resistance.

The working temperature range is -30°C to +80°C.

The resistance to animal and vegetable oils and fats is superior to natural rubber as is the ageing resistance.

Nitrile Chloroprene Rubber NCR

The use of Nitrile in Chloroprene rubber enhances the dynamic properties. For cover rubber which requires a high oil and fat resistance whether it be animal, vegetable or mineral and also requiring superior mechanical properties. NCR is better than CR.

Note

The foregoing **basic materials** are main ingredients of cover rubber components. They provide the principal characteristics of each quality but can be affected by the addition of other **compounding ingredients** to achieve requirements of standards and regulations.

Belt Description

Conveyor belting is described according to laid down International standards. Additionally special types and qualities may be described by the manufacturer or in accordance with Customer wishes.

Ordering Example: 100 m 800 mm EP 400/3 4+2 mm X

- 100 m — Belt Length
- 800 mm — Belt Width
- EP — Material (carcase)
- 400 — Nominal Tensile Strength
- 3 — Number of Plies
- 4 — Carrying Side Cover Thickness
- 2 mm — Pulley Side Cover Thickness
- X — Cover Quality

Belt Length

The belt length is generally given in metres either open or endless.
Open Length is the length around the pulleys plus an allowance for making endless.
Endless Length is the inside circumference of the endless belt.

DUNLOP types

D	DUNLOFLEX	2 Ply Belt	F	FERROFLEX	Steelweave Belt
T	TRIOFLEX	3 Ply Belt	DLP	DUNLOPLAST	Monoply Belt
S	SUPERFORT	Multiply Belt	ST	SILVERCORD	Steelcord Belt

Nominal Tensile Strength the Carcase

This number gives the nominal or breaking strength of the carcase in **N/mm of belt** width. The values are Internationally Standardized.

125	160	200	250	315	400	500
630	800	1000	1250	1600	2000	2500

Number of Plies

The nominal strength of the carcase is made up by a number of plies. The monoply belt has a solid woven carcase.

Nominal Tensile Strength of Plies

63	80	100	125	160	200	250	315	400

The nominal tensile strength of the full thickness carcase is the sum of the ply strengths rounded up to the next nominal tensile strength. The number of plies is not indicated in specially described types such as DUNLOFLEX, TRIOFLEX or DUNLOPLAST.

Main Data Values

After establishing the **duty of the operation** and the **type of belt conveyor,** the main data may be determined.

Belt Speed	v (m/s)
Belt Width	B (m) or (mm)
Carrying Idler Arrangement	
Cross Sectional area of Load Stream	A (m^2)
Conveyor Capacity	Q (t/h)

Speed V

The belt or Conveying Speed V (m/s) must be appropriate for the **material composition** and **operation conditions.**

High Speed - Narrower belt widths
Lower belt tension
Greater wear and tear

Low Speed - Greater belt widths
Higher belt tension
Less wear and tear

The most economical installation is that having the highest belt speed consistent with the type of material and operating conditions.

Standard Values

Speeds V (m/s)
0.42 - 0.52 - 0.66 - 0.84 - 1.05 - 1.31 - 1.68
2.09 - 2.62 - 3.35 - 4.19 - 5.20 - 6.60 - 8.40

Recommended Velocity (m/s)

Duty	v (m/s)
Unit Loads, Assembly Lines	≤ 1.68
Mobile Conveyors	0.52 - 1.68
Very dusty loads such as Flour, Cement	≤ 1.31
Ash and Refuse	≤ 1.68
Grain, Crushed Limestone Gravel, Sand Readymix	1.05 - 2.09
Ores, Bituminous Coal, Sinter Storage and transhipment, Power Stations	1.31 - 3.35
Long distance conveying, overburden Brown coal	2.62 - 6.60
Thrower belts	≥ 8.40
Steep gradient belts Type CHEVRON and HIGH CHEVRON	0.84 - 2.62

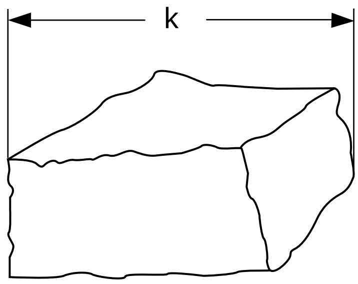
Belt Width B

Wherever possible a standard belt width should be selected. Type and granulometry of the materials determine **the minimum belt width.** After determining the belt type a check on **troughability** may be necessary under some circumstances.

Standard Widths (mm)

300	400	500	650	800	1000
1200	1400	1600	1800	2000	2200

Minimum Belt Widths

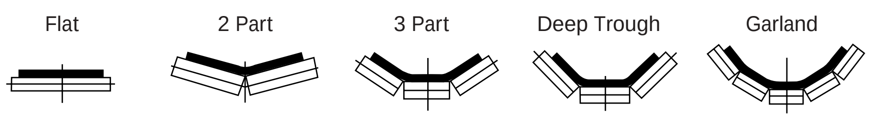


Min. Width (mm)	Lump Size K	
	Sized	Unsized
400	50	100
500	80	150
650	130	200
800	200	300
1000	250	400
1200	350	500
1400	400	600
1600	450	650
1800	550	700
2000	600	800

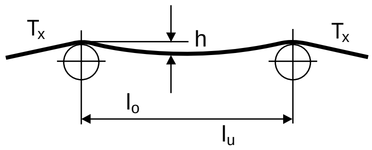
With unsized material the large lumps are embedded in the smaller granulated material.

Carrying Idler Disposition

The disposition of the carrying idler varies from application to application. The number of idler rolls in a carrying idler set and troughing angle determine the **cross sectional area** of the load stream and thus the **conveying capacity.**



Distance between Carrying Idlers



In deciding on the carrying idler spacing one has to take account of the load limitation of the carrying idler (note: Refer to manufacturer's specification). After establishing the belt tension the **belt sag** between idler rollers has to be checked.
The roller spacing has to be selected in such a way that the sag of a loaded belt is no more than 0.5%-1.5% of the centre to centre distance. With return side idlers one may allow approximately 2-3% sag.

Carrying Side

$$l_o = \frac{T_x * 8 * h_{rel}}{(m'_L + m'_G) * g} \quad (m)$$

Return Side

$$l_u = \frac{T_x * 8 * h_{rel}}{m'_G * g} \quad (m)$$

T_x	(N)	Belt tension at point X
m'_L	(kg/m)	Weight of load per metre
m'_G	(kg/m)	Weight of belt per metre
g	(m/s^2)	Gravitational acceleration (9.81 m/s^2)
h_{rel}	(-)	Relative belt sag
		Carrying run : h_{rel} = 0.005-0.015
		Return run : h_{rel} = 0.020-0.030

Values for Idler Spacing

Carrying Side
l_o = 0.5 à 1.0 m   Small installation or high impact
l_o = app. 1.2 m   Normal installation
l_o = 1.4 à 4.0 m   High tension installation

Return Side
$l_u = (2\text{-}3) * l_o$ Maximum approx 6 m

Idler Rotation

$$n_R = \frac{60 * v}{\pi \quad D_R}$$ (r.p.m.)

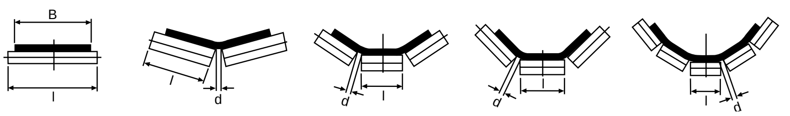
D_R (m) Roll diameter
v (m/s) Belt speed

Idler rotation should not be greater than approximately 650 r.p.m.

Standard Idler Diameter (mm)

Carrying Idlers	51	63.5	88.9	108	133	159	193.7	219
Impact Idlers				156	180	215	250	290
Return Run Support Discs		120	138	150	180	215	250	290

Standard length L (mm) of rollers

Belt Width B (mm)	Troughing Type Flat	2 roll	3 roll	Deeptrough	Garland
					
300	380	200	-	-	-
400	500	250	160	-	-
500	600	315	200	-	-
600	700	340	250	-	-
650	750	380	250	-	-
800	950	465	315	200	165
1000	1150	600	380	250	205
1200	1400	700	465	315	250
1400	1600	800	530	380	290
1600	1800	900	600	465	340
1800	2000	1000	670	530	380
2000	2200	1100	750	600	420
2200	2500	1250	800	640	460

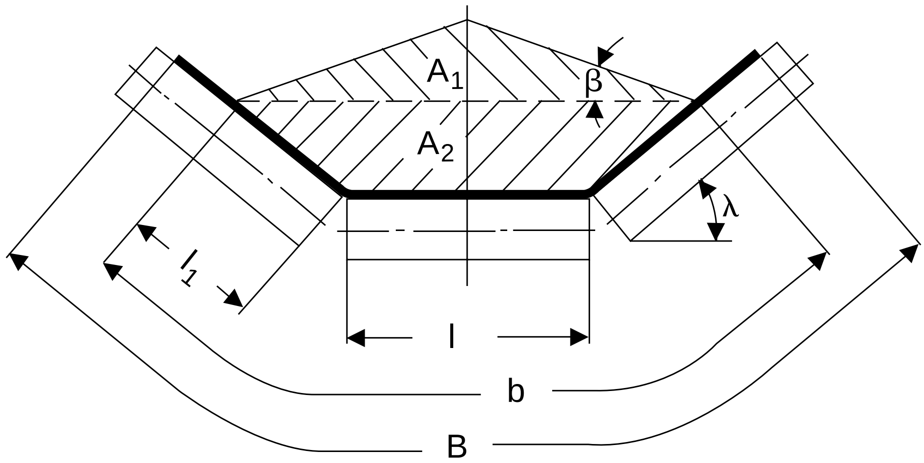
The length of the middle idler roll determines the cross sectional area of the load and thus the conveying capacity.

The **gap d** between 2 adjacent rolls should not be greater than 10mm, with belt widths B > 2000mm d = 15mm refer to DIN22107, carrying idler arrangements.

FERROFLEX Conveying Broken Stone

Cross Sectional Area of Load Stream

To determine the cross-sectional area of the load stream A, one may use as a basis the geometric relationship which can be constructed from the troughing angle λ, the usable belt width b and the angle of surcharge ß.



Cross-Sectional Area of Load Stream

$$A = A_1 + A_2 \qquad (m^2)$$

For 1, 2 and 3 roll carrying idler sets, the part cross-sectional area can be calculated as follows:

Part Cross-Sectional Area

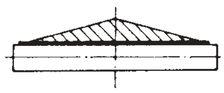
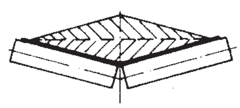
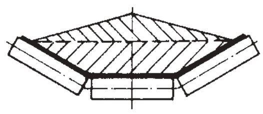
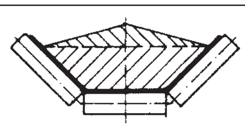
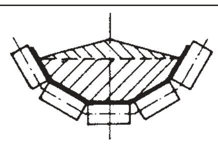
$$A_1 = 0.25 * \tan\beta * [\, l + (b - l) * \cos\lambda \,]^2 \qquad (m^2)$$

$$A_2 = l_1 * \sin\lambda * [\, l + l_1 * \cos\lambda \,] \qquad (m^2)$$

l (m) Length of middle carrying roll
l_1 (m) Loading width of outer rolls
$l_1 = 0.5 (b - l)$ for 3 roll idler sets
$l_1 = 0.5 (b - 3 * l)$ for 5 roll Garland sets

b (m) Usable belt width (loadstream width)
$b = 0.9 * B - 50$ mm for belts $B \leq 2000$ mm
$b = B - 250$ mm for belts $B > 2200$ mm
λ (°) Troughing Angle
β (°) Surcharge Angle

Cross Sectional Area Comparison for Various Forms of Troughing.

Troughing Form	Troughing Angle	Load Cross Section Area A(M^2)	Comparison
	Flat	0.0483	44%
	20° 30°	0.1007 0.1145	91% 104%
	20° 30° 45°	0.0935 0.1100 0.1247	85% 100% 113%
 Deep Trough	20° 30° 45°	0.0989 0.1161 0.1284	90% 106% 117%
 Garland	30°/60°	0.1329	121%

Note:
The values for the cross sectional area and the comparison are for a belt width B = 1000 mm and for an angle of surcharge β = 15°.

Conveying Capacity

The conveying capacity is determined from the **cross-sectional area A** and the **belt speed v (m/s).**

Load Stream Volume

$$Q_V = A * v * 3600 * \varphi \qquad (m^3/h)$$

The **effective or nominal load stream volume** is determined **from the effective degree of filling.** This takes account of working conditions and the gradient of the installation.

Effective Degree of Filling

$$\varphi = \varphi_1 * \varphi_2 \qquad (-)$$

Degree of Filling φ_1

The degree of filling is dependent upon the characteristics of the load, e.g., the lump size, the surcharge angle and the working conditions, e.g, method of loading, tracking or the reserve capacity etc.

Values for φ_1

$\varphi_1 = 1$ for normal working conditions;
$\varphi_1 = 0.8 - 0.95$ for adverse condition.

Reduction Factor φ_2

The reduction factor φ_2 takes into consideration the reduction in part cross-sectional area A_1 as a result of the conveying gradient.

for Smooth Belts

Gradient	2°	4°	6°	8°	10°	12°	14°	16°	18°	20°	22°
φ_2	1.0	0.99	0.98	0.97	0.95	0.93	0.91	0.89	0.85	0.81	0.76

for Steep Incline Belts

Angle of inclination	15°	20°	25°	30°	35°	40°
Spharical rolling and coarse Material	0.89	0.81	0.70	0.56	–	–
Sticky material	1.00	0.93	0.85	0.68	0.58	0.47

For Bulk Loads

The load stream Qm (t/h) is calculated thus:

Load stream mass

$$Q_m = Q_V * \rho \qquad (t/h) \quad \text{Theoretical value}$$

$$Q_m = Q_V * \rho * \varphi \qquad (t/h) \quad \text{Effective value}$$

For Unit Loads

For the calculation of the conveying capacity for unit loads the following formula applies.

Quantity conveyed

$$Q_{st} = \frac{3600 * v}{l_{st} + a_{st}} \qquad (St/h - pieces\ per\ hour)$$

Load Stream

$$Q_m = \frac{m_{st} * 3.6 * v}{l_{st} + a_{st}} \qquad (t/h)$$

Q_V	(m^3/h)	Volume (values see Appendix) with v = 1 m/s
v	(m/s)	Belt Speed
ρ	(t/m^3)	Bulk density (see Appendix)
m_{st}	(kg)	Piece Weight
l_{st}	(m)	Piece Length in direction of travel
a_{st}	(m)	Spacing of pieces

This section deals with **estimated values** and **relevant matters** primarily to enable the Project Engineer to provide a **speedy assessment** of requirements from the given service data. To make an optimum selection of the conveyor belt and conveyor components it is recommended that detailed calculations be done in accordance with subsequent sections.

Power Requirements

With the aid of the following formulae, **power requirements** can be **roughly assessed.** The accuracy is sufficient for normal installations with simple straight-forward running conditions. From the power calculations, the belt type can be closely determined. The actual nominal belt weight can be used in the more precise calculation of the belt.

Power at Drive Pullley

$$P_T = P_1 + P_2 + P_3 \quad (kW)$$

Power for empty Conveyor and Load over the Horizontal Distance

$$P_1 = \frac{C_B * v + Q_m}{C_L * k_f} \quad (kW)$$

Power for Lift (or Fall)

$$P_2 = \frac{H * Q_m}{367} \quad (kW)$$

Additional Power

P3 is the sum of additional power for trippers, skirtboard friction, ploughs.

Required motor power $P_M = P_T / \eta$ (kW)

Installed Motor

P_N selected motor from standard list.

Q_m	(t/h)	mass of load stream
v	(m/s)	belt speed
C_B	(kg/m)	width factor (see table)
C_L	(m^{-1})	length factor (see table)
H	(m)	conveyor elevation $H = \sin \delta * L$
L	(m)	conveying length
δ	(°)	angle of inclination
k_f	(-)	Service factor (see table)
η	(-)	efficiency of the drive $\eta = 0.9$ for drives with fluid couplings

Standard electric motors (kW)

1.5	2.2	3	4	5.5	7.5	11
15	18.5	22	30	37	45	55
75	90	110	132	160	200	250
315	400	500	630			

Width factor C_B

Duty	Bulk Density ρ (t/m3)	Belt Width B (mm) 300	400	500	650	800	1000	1200	1400	1600	1800	2000	2200
Light	Up to Ca.1.0	31	54	67	81	108	133	194	227	291			
Medium	1.0 to 2.0	36	59	76	92	126	187	277	320	468	554	691	745
Heavy	Over 2.0		65	86	103	144	241	360	414	644	727	957	1033

Length Factor C_L

L (m)	3	4	5	6	8	10	12.5	16	20
C_L	667	625	555	526	454	417	370	323	286
L (m)	25	32	40	50	63	80	90	100	150
C_L	250	222	192	167	145	119	109	103	77
L (m)	200	250	300	350	400	450	500	550	600
C_L	63	53	47	41	37	33	31	28	26
L (m)	700	800	900	1000	1500	2000			
C_L	23	20	18	17	12	9			

L (m) Conveying Length

Working Conditions Factor k_f

Working Conditions	k_f
Favourable, good alignment, slow speed	1.17
Normal (Standard Conditions)	1
Unfavourable, dusty, low temperature, overloading, high speed	0.87 - 0.74
Extremely low temperature	0.57

Additional Power Values

Trippers (throw-off carraiges)	Belt Width B (mm)	P (kW)
	≤ 500 ≤ 1000 > 1000	$0.8 * v$ $1.5 * v$ $2.3 * v$
Scrapers (for installations L ≤ 80 m)	**Scraper Type**	
	simple, normal contact	$0.3 * B * v$
	heavy contact	$1.5 * B * v$
	multifunctional fac scraper	$1.8 * B * v$
Material - skirtboard	beyond loading point	$0.16 * v * l_f$
Discharge plough	Bulk density $\rho \leq 1.2$ Angle $\alpha = 30° - 45°$	$1.5 * B * v$

B (m) Belt width
v (m/s) Belt speed
l_f (m) Length of material between skirtboard

Belt Type

This formula is to enable calculation of **belt breaking strength** and applies to installations with a single pulley head drive, an angle of wrap α = 200°, a safety factor S = 8 to 10. By using a dual pulley head drive or a head and tail drive a lower strength belt type can result.

Breaking Strength

$$k = \frac{c_R}{c_V} * \frac{P_T}{v} \quad \text{(N/mm)}$$

Nominal Breaking Strength

The nominal Breaking Strength k_N is obtained by rounding up to the next highest belt type.

Number of Plies

The number of plies or thickness of carcase depends mainly on the belt width, material characteristics (bulk density, lump size) and conditions such as transitions, height of material fall, installation path, troughing etc.

Friction Value Factor c_R

Drive Pulley Surface	Friction Value μ	Belt Width B (mm) 300	400	500	650	800	1000	1200	1400	1600	1800	2000	2200
bare, wet	0.15	98	74	59	45	37	30	25	21	18	16	15	14
rubber lagged, wet	0.25	69	52	41	32	26	21	17	15	13	12	10	9
and dirty	0.30	62	46	37	28	23	18	15	13	12	10	9	8
bare, dry, lagged, wet	0.35	57	43	34	26	21	17	14	12	11	9	8	8
rubber lagged, dry	0.40	53	40	32	25	20	16	13	11	10	9	8	7

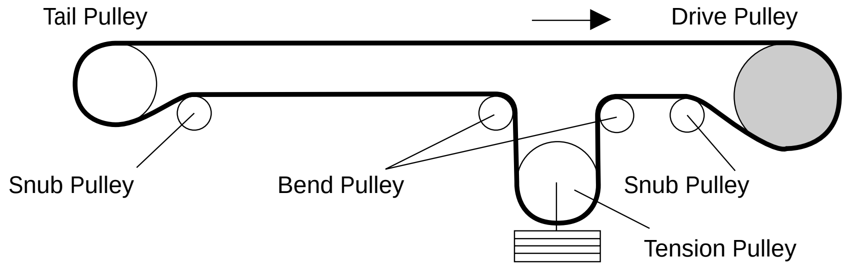
Breaking Strength Loss at Joint Factor C_v

DUNLOP Belt Type	Splice Type Ply Rating	Factor c_v
DUNLOFLEX	2 ply overlap 100% 1 ply overlap 50%	1.00 0.50
TRIOFLEX	3 ply overlap 100% 2 ply overlap 67%	1.00 0.67
SUPERFORT	Number of plies 1 2 3 4 5 6	0.70 0.50 0.67 0.75 0.80 0.83
FERROFLEX	Zig-Zag Splice Joint	0.90
DUNLOPLAST	Finger Splice Joint	0.90
Steel Cord Belts	Splice 1 and 2 step 3 step 4 step	1.00 0.95 0.90
Various	Mechanical Joints: Refer to Manufacturer	

Pulley Diameters

The pulley diameter for a conveyor belt depends upon the **belt construction** (belt type, carcase material and thickness of carcase), the **duty** and the **method** of **splicing.** One can differentiate between 3 pulley groups depending on pulley location and angle of wrap α.

GROUP	APPLICATION	
A	Pulleys in the areas of high belt stress.	Drive Pulleys
B	Pulleys in areas of low belt stress.	Tail Pulleys
C	Pulleys with an angle of wrap $\alpha \leq 90°$,	Deflection or snub Pulleys



To determine the pulley diameter first ascertain the diameter of the **drive pulley** with maximum tension.

$$D_{Tr} = C_{Tr} * d \quad (mm)$$

d	(mm)	Thickness of carcase (see Appendix C)
C_{Tr}	(-)	Value for warp material of carcase i.e. Belt type.

Value C_{Tr}

C_{Tr}	Material of Carcase in Warp or Belt Type	
90	Polyamide (P)	
80	DUNLOFLEX	2 ply Belt
95	TRIOFLEX	3 ply Belt
108	SUPERFORT	Multiply Belt (EP)
138	FERROFLEX	Steel Weave Type
145	SILVERCORD	Steel Cord Belt
100	DUNLOPLAST	Monoply Belt

The calculated pulley diameter is rounded up to the next higher standard diameter or if working conditions are favourable may be rounded down.

Standard Pulley Diameter (mm)

100	125	160	200	250	315	400	500
630	800	1000	1250	1400	1600	1800	2000

Once the diameter of the largest pulley has been determined to a standard size the diameter for pulley groups A, B, C can be obtained from the following table.

Pulley Diameter D_{Tr} (mm)	Diameter Of Pulley Groups (mm) A	B	C
100	100	-	-
125	125	100	-
160	160	125	100
200	200	160	125
250	250	200	160
315	315	250	200
400	400	315	250
500	500	400	315
630	630	500	400
800	800	630	500
1000	1000	800	630
1250	1250	1000	800
1400	1400	1250	1000
1600	1600	1250	1000
1800	1800	1400	1250
2000	2000	1600	1250

The diameters shown in the above table apply to belts operating at 60% - 100% of allowable tension.

Utilisation Percentage

$$k_A = \frac{T_{max} * S}{B * k_N} * 100 \qquad (\%)$$

T_{max}	(N)	Max. belt tension
S	(-)	Belt safety factor when running
B	(mm)	Belt Width
k_N	(N/mm)	Nominal breaking strength of belt

When the rated tension of the belt is not fully utilized, it is permissible to reduce pulley diameter to 1 to 2 sizes smaller.

Degree of Utilization	Pulley Diameter	
k_A > 0.6 à 1	Diameters as table	
k_A > 0.3 à 0.6	Group A, B and C	one size smaller
k_A < 0.3	Group A and B	2 sizes smaller
	Group C	one size smaller

For pulley diameters of the DUNLOP belt types: see Appendix K.

Pulley Revolution

$$n_T = \frac{v * 60}{\pi * D} \qquad (\text{T/min})$$

Pulley Loading

$$F_T = \frac{T_{A1} + T_{A2}}{9.81} \qquad (\text{kg})$$

Average Surface Pressure

$$p_T = \frac{T_{A1} + T_{A2}}{D * B} \qquad (\ N/cm^2\ )$$

With rubber lagged drive pulleys the average surface pressure limit is approximately 70 N/cm^2. This limit is only reached on heavy drives and relatively small pulley diameters.

Transmission Capability

$$p_{ü} = \frac{360 * F_U}{\pi * D * \alpha * B} \qquad (\ N/mm^2\ )$$

The transmission capability $p_{ü}$ is not to be confused with surface pressure! The surface pressure is summary calculated, the belt bending stresses being ignored. With high belt stresses the pulley diameter becomes too large because of the low values for $p_{ü}$.

$p_{ü}$ = 1600 - 2000 N/mm^2 above ground
$p_{ü}$ = 3000 - 3500 N/mm^2 under ground

This formula is no longer used to determine the pulley diameter.

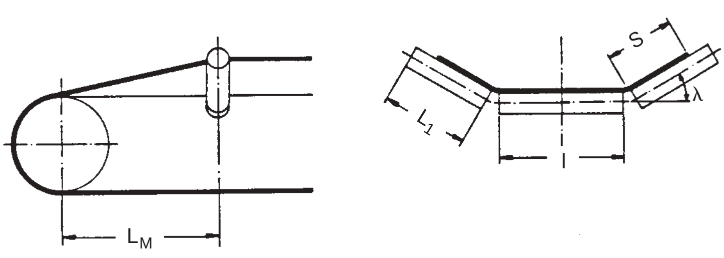
Torque at Start-Up

$$M_A = \frac{F_A * D}{2 * 1000} \qquad (\ Nm\ )$$

D	(mm)	Pulley diameter
B	(mm)	Belt width
F_A	(N)	Peripheral force at start-up
α	(°)	Angle of wrap
v	(m/s)	Speed
F_U	(N)	Peripheral force when running

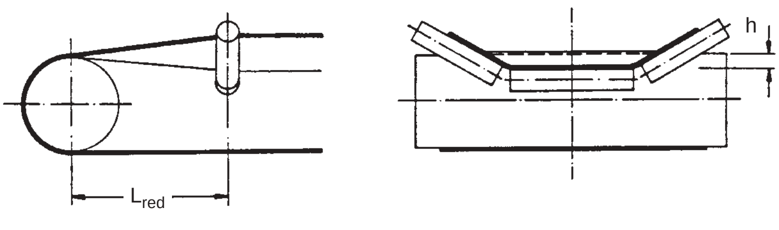
Troughing Transition

The distance between a terminal pulley and the adjacent fully troughed idler set at either the head or tail end of a conveyor is know as the transition distance. In this distance the belt changes from a fully troughed to a flat profile or vice versa respectively. The belt stretches additionally at its edge zone and buckling in the middle of the belt is also possible.



Elevation of Pulley

To relieve the belt and edge stresses, the pulley may be raised slightly to the value h (mm) (see Page 11.13).



The necessary transition length for tension distribution with or without pulley elevation when designing can be estimated or taken from the table.

Troughing Transition

$$L_M = x * s * \sin \lambda \quad (mm)$$

Pulley Rise

$$h = \frac{s^2}{B} * \sin \lambda \quad (mm)$$

Reduced Transition Distance

$$L_{red} = x * (s * \sin \lambda - h) \quad (mm)$$

L_M	(mm)	Normal transition length	
L_{red}	(mm)	Reduced transition length with raised pulley	
l	(mm)	Length of centre carrying roller (see page 11.3)	
s	(mm)	Portion of belt in contact with side idler roller, s = 0.5 * (B-l)	
x	(-)	Factor for carcase	x = 8 for Textile belts x = 16 for ST. belts
λ	(°)	Troughing angle	
h	(mm)	Pulley lift	

The pulley lift should not exceed the value h (mm) in order not to get a “ski jump” effect. Lift-off from the centre roller is not desirable as this would have an adverse effect on belt tracking.

Trough Transition Values

Troughing Angle	Textile belts L_M (mm)		Textile belts L_{red} 1) (mm)			ST Belts L_{red} 1) (mm)	
Troughing λ	20°	30°	30°	40°	45°	30°	45°
Belt Width (mm)							
500	410	600					
650	550	800					
800	665	970	680	870	950	1350	1900
1000	850	1240	860	1100	1210	1710	2420
1200	1000	1470	1020	1310	1440	2040	2870
1400	1190	1740	1200	1540	1700	2400	3380
1600	1370	2000	1380	1770	1950	2750	3890
1800	1550	2260	1550	1990	2200	3100	4390
2000	1710	2500	1720	2200	2430	3432	4860
2200	1920	2800	1900	2450	2700	3800	5390

Values apply to 3 roll idler sets.

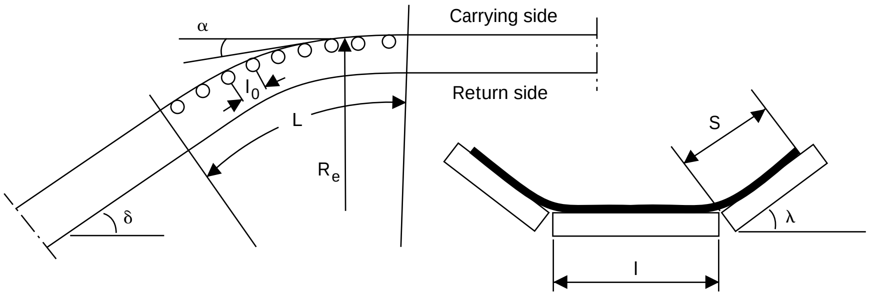
1) Reduced lengths L_{red} apply to the raised pulley with the value h (mm) according to the table.

Pulley Lift h (mm)

Belt Width (mm)	Troughing Angle λ 30°	40°	45°
800	37	47	52
1000	48	62	68
1200	56	72	80
1400	68	87	96
1600	78	100	110
1800	89	114	125
2000	98	126	138
2200	112	143	158

Convex Vertical Curve

In the transition from inclined to horizontal, the belt edge zone is subjected to **additional stretch.** In order not to exceed certain limits, as a rule 0.8% additional stretch, the transition radius R_e has to be dimensioned accordingly.



Convex Radius

$$R_e = x * s * \sin\lambda \quad (m)$$

s	(mm)	Portion of belt in contact with side idler roller
x	(-)	Factor for carcase x = 125 for textile belts x = 400 for steel cord belts

Curve Length

$$L = \pi * \delta * R_e / 180 \quad (m)$$

Number of Idlers in Curve

$$z = \delta / \alpha \quad \text{(Pieces)}$$

Idler Pitch

$$l_o = L / z \quad (m)$$

α	(°)	Deviation per idler α = approx 2° for 30° troughing α = approx 3° for 20° troughing
δ	(°)	Gradient of installation

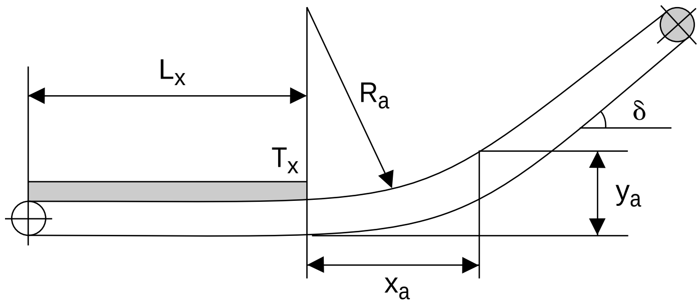
Values of Radius R_e (m)

Belt Width (mm)	Textile belts			ST Belts	
	20°	30°	45°	30°	45°
500	6.5	9.3	13.5	30.0	-
650	8.5	12.5	17.5	40.0	-
800	10.5	15.0	21.0	48.5	68.5
1000	13.0	19.5	27.0	62.0	88.0
1200	16.0	23.0	32.0	74.5	104.0
1400	18.5	27.0	38.0	87.0	123.0
1600	21.0	31.0	44.0	100.0	141.0
1800	24.0	35.0	50.0	113.0	160.0
2000	26.5	39.0	55.0	125.0	177.0
2200	30.0	44.0	62.0	140.0	198.0

Minimum radii for 3 roll carrying idlers.

Concave Vertical Curve

With a concave vertical curve, the belt path goes from horizontal to incline. At start-up or load change there may be a risk of the belt lifting off the carrying idlers in this region. This can lead to a reduction in pre-tension. If due to special circumstances a brief lift-off can be tolerated then the lifting belt can be restrained by rollers mounted above the troughing idlers. In any event action must be taken to ensure that the belt does not lose load material.



Concave Radius

$$R_a = \frac{T_x}{m'_G * \cos\delta * g} \quad (\text{ m })$$

T_x	(N)	Tension at start of curve when fully loaded
m'_G	(kg/m)	Belt weight
δ	(°)	Gradient angle, up to 18° use 18°, cos δ ≃1

Co-ordinates of Curve

$x_a = R_a * \tan\delta$	horizontal distance
$y_a = 0.5 * R_a * \tan^2\delta$	vertical distance

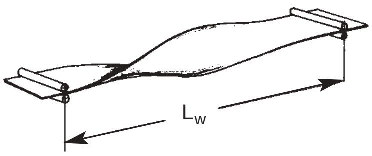
Additional Stretch-buckling

With a concave vertical curve, **buckling** of the belt edges and **overstress** of the middle of the belt can occur. (To check calculation see Appendix I.5.)

Belt Turnover

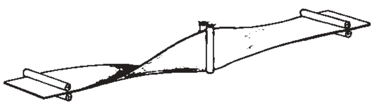
One method of solving **belt cleaning problems** on the **return side run** is the use of belt turnover. This method is employed in inaccessible areas such as conveyor bridges and tunnels. To avoid excessive strain in the **edge region** of the belt, the turnover must be a certain minimum length, depending on the method of turnover, belt type and belt width.

Un-guided Turnover



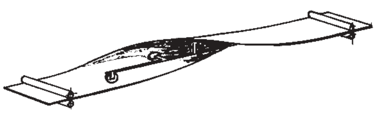
With textile belts up to a width of 1200 mm and a thickness of 10 mm, the belt may be turned over unguided and without support. At the entry and exit points of the turnover lengths, the belt is fed through a pair of rollers.

Guided Turnover



With textile and steel cord belts up to 1600 mm wide to support the belt in the middle of the turnover length, a pair of vertical rollers is used. At the entry and exit points the belt is fed through a pair of horizontal rollers.

Supported Turnover



With textile and steel cord belts up to 2200 mm wide. With this turnover the belt is fed over support rollers which run on a lengthwise axis.

Turnover Lengths

for EP belts

$$L_W = 1.36 * B * \sqrt{71} = 11.5 * B \quad (m)$$

for ST belts

$$L_W = 1.55 * B * \sqrt{245} = 24 * B \quad (m)$$

Values for Turnover Lengths L_W (m)

Belt Type	Method of guiding belt in Turnover Lengths: free or with support rollers	Method of guiding belt in Turnover Lengths: guided with a middle roller
EP belts	12 * B	13 * B
DUNLOPLAST	16 * B	19 * B
FERROFLEX	17 * B	20 * B
ST. BELTS	18 * B	22 * B

For smooth running of the belt through the turnover stretch, a **minimum belt tension** is necessary.

Min. Belt Tension at Turnover

$$\text{min. } T_x = \frac{L_W * m'_G * g}{8 * h} \quad (N)$$

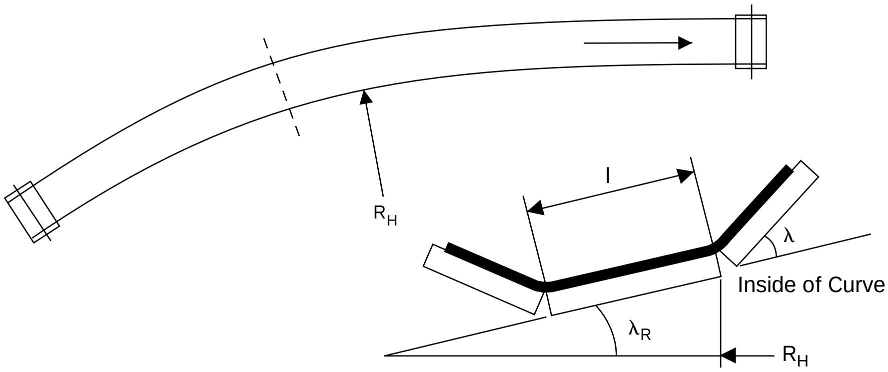
L_W	(m)	Length of Turnover Stretch
h	(%)	Degree of sag 1.5% is recommended (h = 0.015)
m'_G	(kg/m)	Load due to conveyor belt
B	(m)	Belt Width

If T_x is > than T_2 or T_3 then **belt tensions have to be corrected.**

Horizontal Curve

Within limits, belts are able to negotiate horizontal curves. To do this the carrying idlers on the inside of the curve are slightly raised by the angle λ_R according to radius and belt tension by approximately 5°-15°.

The exact position has to be determined by adjustment.



Value of Curve Radius

$$R_H = k * [\, l + B * (1 - \frac{l}{B}) * \cos\lambda \,] \qquad (m)$$

l	(mm)	Length of middle carrying roller (see Page 11.3)
B	(mm)	Belt width
λ	(°)	Troughing angle
λ_R	(°)	Lifting of the carrying rollers (ca. 5° to 15°)
k	(-)	Factor (consider belt type and duty)
		for EP belts k = 71
		DUNLOPLAST k = 150
		FERROFLEX k = 225
		ST belts k = 245

For design purposes the following radii may be selected being the **smaller permissible radii R_H (m).**

Belt width (mm)	300	400	500	650	800	1000	1200	1400
EP belts	20	26	33	42	52	65	72	91
DUNLOPLAST	42	56	69	89	110	137	165	192
FERROFLEX	-	-	-	134	165	206	248	289
ST belts	-	-	-	146	180	224	270	314
Belt width (mm)	**1600**	**1800**	**2000**	**2200**	**2400**	**2600**	**2800**	
EP belts	104	117	130	142	156	169	182	
DUNLOPLAST	220	247	275	-	-	-	-	
FERROFLEX	329	370	412	452	-	-	-	
ST belts	359	404	449	493	-	-	-	

The values R_H apply to a troughing angle of 30°

After the design of the installation and the main data has been determined **an exact calculation** can be undertaken. Established installation components can be checked and dimensioned. The belt selection can be undertaken according **to forces that have been established** and **other relevant criteria.**

Resistance to Motion F_H

The resistances to motion within a belt installation may be categorised thus:
- **Main resistance**
- **Secondary resistance**
- **Slope resistance**
- **Special resistances**

Main Resistance F_H

$$F_H = f * L * g * [m'_R + (2 * m'_G + m'_L) * \cos \delta] \qquad (N)$$

Resistance due to moving the mass of idlers, belt and material on the carrying and return runs. Running **resistance of idlers** (bearing and seal friction). **Resistance due to impressions** made in the belt by idlers and the flexing of the belt.

With gradients $\delta \leq 18°$ $\cos \delta = 1$.

Secondary Resistance F_N

$$F_N = (C - 1) * F_H \qquad (N)$$

Resistance occurring mainly in the **loading area** such as the acceleration of the material at the loading point. Resistance due to the friction on the side walls of the chute, resistance due to belt flexing on pulleys, pulley bearing resistance.

Note

- With normal installations with one loading point, the secondary resistance can be calculated by using the **factor C** as part of the main resistance. Factor C depends on the conveying length and can be taken from the table on Page 12.3.
- If the secondary resistance relative to the total resistance is high as for instance with conveyors less than approximately 80 m long with several loading points, **a separate calculation is necessary.**

Slope Resistance F_{St}

$$F_{St} = H * g * m'_L \qquad (N)$$

Resistance from **load** and **elevation.**

Special Resistances F_S

The special resistances are due to installation components such as:

- the forward tilt of outer idler rollers to improve tracking
- material deflection ploughs and belt cleaners, continuous skirtboards
- trippers (throw-off carriages)
- bunker drag out belts

The resistances to motion are relatively simple to calculate. Information not available such as pulley diameters etc. can be estimated for initial calculation formulae or taken from the appropriate table.

Formulae for calculation: see Appendix E.

Peripheral Force F_U

Peripheral force steady state working

Working Conditions

The sum of the resistances to motion is equal to the peripheral force FU at the drive pulley (or shared over several drive pulleys).

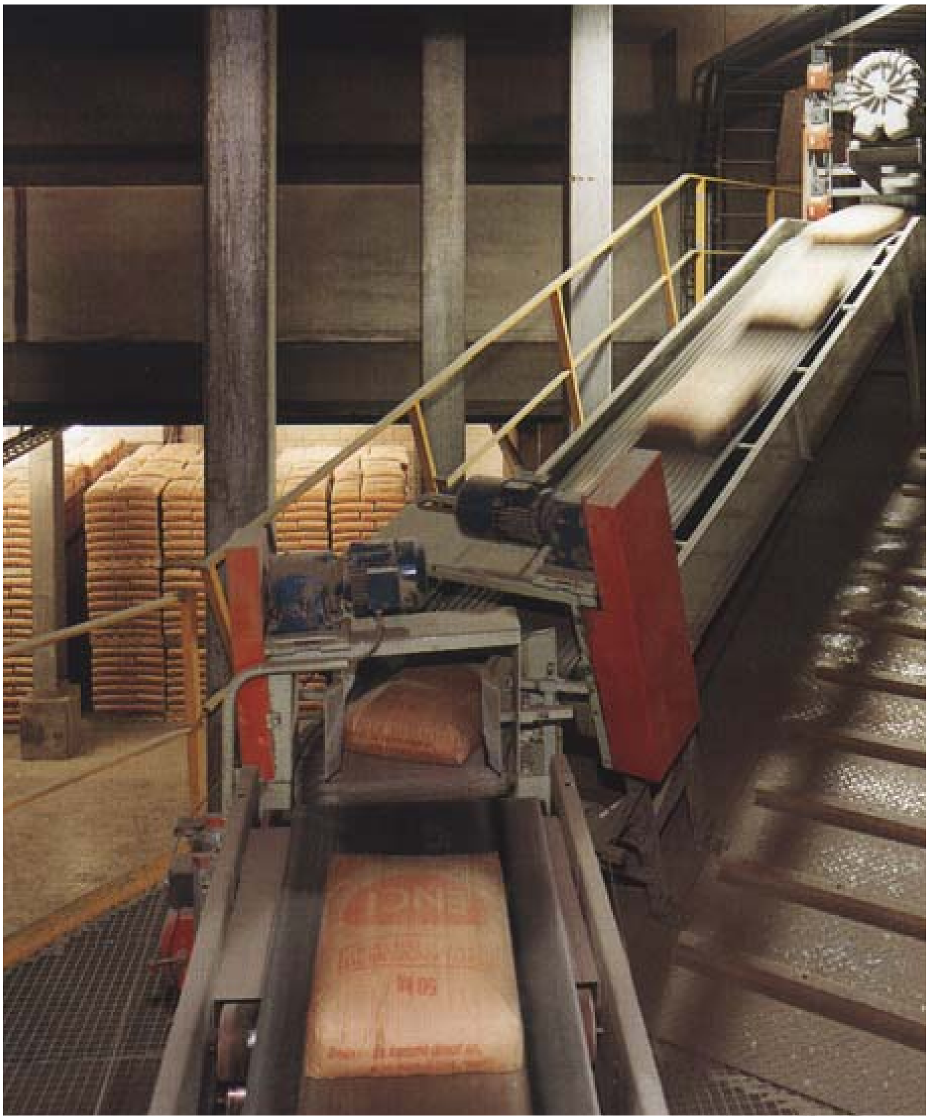
$$F_U = F_H + F_N + F_{St} + F_S \quad (N)$$

For installations with one loading point, as a rule the following summation applies:

Summation

$$F_U = C * f * L * g * [m'_R + (2 * m'_G + m'_L) * \cos\delta] + H * g * m'_L + F_S \quad (N)$$

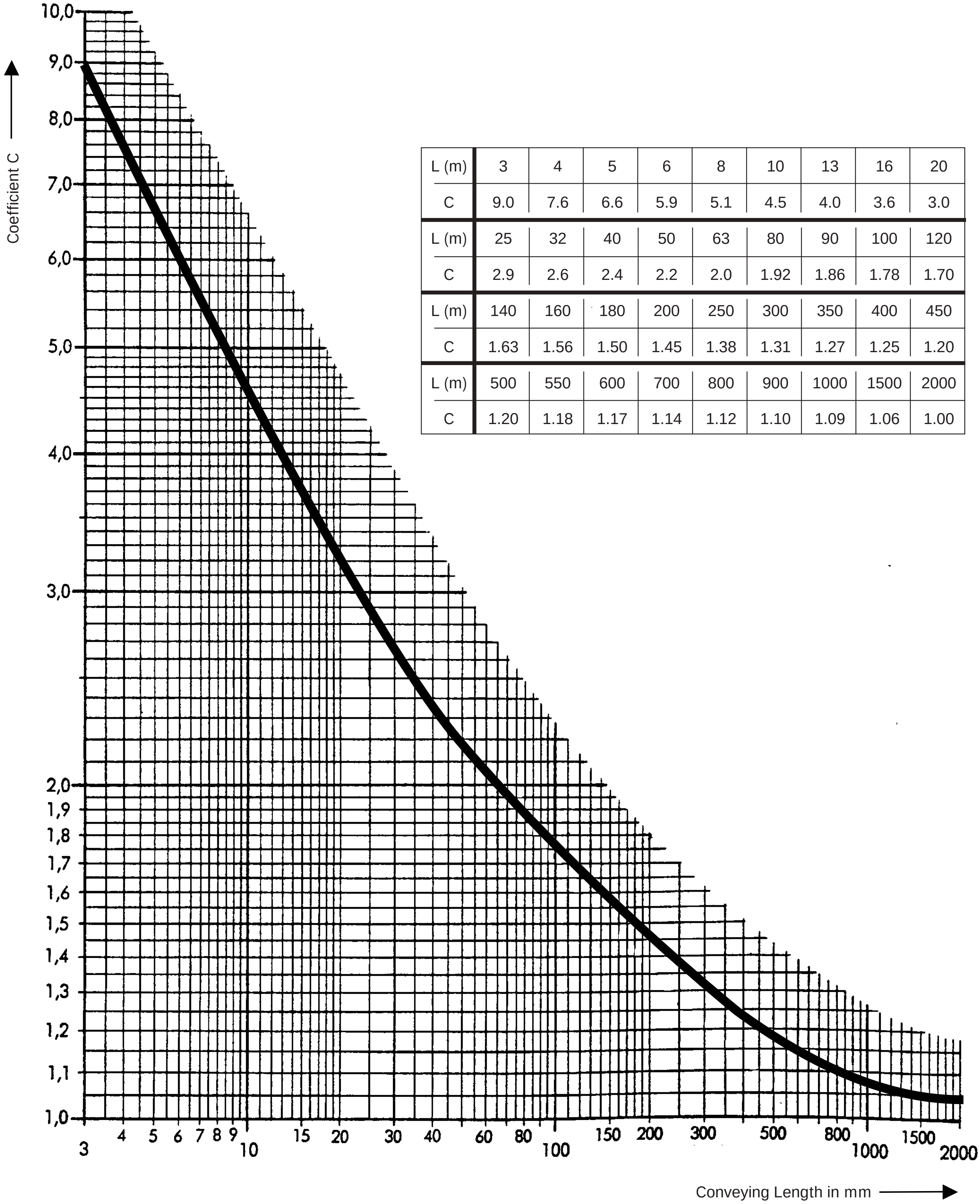
C	(-)	Length factor
f	(-)	Artificial friction factor
L	(m)	Conveyor length
g	(m/s^2)	Acceleration due to gravity
m'_R	(kg/m)	Mass of the rotating carrying and return idlers
m'_G	(kg/m)	Mass of the belt
m'_L	(kg/m)	Mass of the load
δ	(°)	Gradient of the installation with $\delta \leq 18°$ $\cos\delta = 1$
H	(m)	Conveying height
		$H > 0$ inclined conveying
		$H < 0$ declined conveying
F_S	(N)	Sum of the special resistances. For separate calculations see Appendix.



Herringbone Belt in Operation

Factor C

This adjustment factor makes allowance for **secondary resistances** (See page 12.1). The influence of Factor C will decrease the greater the conveyor length. For installations less than a conveying length of approximately 80m or having several loading points see appendix D.



L (m)	3	4	5	6	8	10	13	16	20
C	9.0	7.6	6.6	5.9	5.1	4.5	4.0	3.6	3.0
L (m)	25	32	40	50	63	80	90	100	120
C	2.9	2.6	2.4	2.2	2.0	1.92	1.86	1.78	1.70
L (m)	140	160	180	200	250	300	350	400	450
C	1.63	1.56	1.50	1.45	1.38	1.31	1.27	1.25	1.20
L (m)	500	550	600	700	800	900	1000	1500	2000
C	1.20	1.18	1.17	1.14	1.12	1.10	1.09	1.06	1.00

Friction Factor f

The friction factor f is used for the calculations of the resistances to motion. It provides an estimate of the resistance to rotation of the idlers, the belt resistance (flexing and idler impressions) and material impression resistance. Values for the factor f are dependent upon the **working conditions** and **construction characteristics** of the installation.

Horizontal, inclined or slightly declined installations - Motor driven	
Favourable working conditions, easily rotating idlers, material with low internal friction and good tracking, good maintenance	0.017
Normal installation, normal material	0.020
Unfavourable conditions, low temperature, material with high internal friction, subject to overload, poor maintenance	0.023 - 0.027
Installations with steep declines creating regenerative conditions	0.012 - 0.016

The above table values are relevant for v = 5 m/s and a temperature of + 20°C.

Adjustments

Under certain circumstances the following adjustments are possible or necessary:

a) With other **speeds** the following applies:

With v <> 5 m/s $f = c * f_{5m/s}$

v (m/s)	2	3	4	5	6
Factor c	0.80	0.85	0.90	1.00	1.10

b) With other **temperatures** the following applies:

With T < 20°C $f = c * f_{20°C}$

T (°C)	+20°	0°	-10°	-20°	-30°
Factor c	1.00	1.07	1.17	1.28	1.47

For central European regions and normal working conditions the value of f = 0.020 - 0.021 would be used.

In extreme temperatures such as those in tropical regions a value for f = 0.017 would apply.

In extremely low temperatures a value for f up to 0.035 would be used.

Mass of Rotating Parts m'_R
Weight of rollers per m

The mass of rotating parts m'_R (kg/m) is calculated from the weight of the rotating idler rollers on the carrying and return runs.

$$m'_R = \frac{m_{Ro}}{l_o} + \frac{m_{Ru}}{l_u} \qquad (\text{kg/m})$$

m_{Ro} (kg) Mass of one set of carrying idler rollers
m_{Ru} (kg) Mass of one set of return idler rollers
l_o (m) Pitch of carrying idlers
l_u (m) Pitch of return idlers

Mass of rollers
m_{Ro} et m_{Ru}

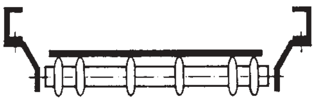
For the masses m_{RO} and m_{Ru} (kg). Applicable to standard rollers diameters and belt widths 300 mm to 2200 mm. See table in the Appendix.

Roller Pitch

If this information is not available roller pitch values as per Page 11.3 may be used.

Idler Roller Diameter

Standard **idler roller diameters (mm)** of flat or troughed rollers according to DIN 15107 are:



(51)	63.5	88.9	108	133	159	193.7	219.1

Tube dia. mm.	51	57	63.5	88.9	108	133
Disc rollers dia. mm.	120	133	150	180	180 215	215 250

The idlers and discs may vary from manufacturer to manufacturer. For location of disc rollers see Appendix M.1.

Mass of Belt m'_G
Belt weight per m

The mass of the belt m'_G is determined from the weights of the various belt types.

$$m'_G = m''_G * B \qquad (\text{kg/m})$$

m''_G (kg/m^2) Belt weight per m^2
B (m) Belt width

For the belt weights m''_G (kg/m2) of the DUNLOP-belt types see Appendix C.

Mass of the Load m'_L
Material weight per m

The load m'_L is derived from the cross sectional area of the bulk load stream Q_m alternatively with piece loads, from the weight of one piece.

For bulk loads

$$m'_L = \frac{Q_m}{3.6 * v} \quad \text{(kg/m)}$$

For unit loads

$$m'_L = \frac{m_{st}}{l_{st} + a_{st}} \quad \text{(kg/m)}$$

Q_m	(t/h)	Loadstream mass
v	(m/s)	Belt speed
m_{st}	(kg)	Weight of each piece
l_{st}	(m)	Length of each piece
a_{st}	(m)	Spacing of pieces

Peripheral Force F_A at Start-Up

Peripheral force F_A in a non-steady state running condition

At the **breakaway and start-up** of a loaded installation, the **inertial resistances** to motion of the masses to be moved, have to be overcome. The belt stresses during acceleration have to be kept to a minimum. The initial pulley peripheral force at start-up must not exceed a certain value.

Recommendations

- The maximum peripheral force F_A should not be greater than approx (1.3 to 1.5) $*$ F_u, the steady state running peripheral force.
- In order to accelerate the masses on the conveying length, a force equivalent to a minimum of 20% of the motional resistances should be available.
- The peripheral force F_A should be applied to the belt over such a period of time that the installation is maintained at almost a steady state and so accelerates with minimum **additional dynamic forces.**
- At start-up with acceleration aA and peripheral for F_A one should seek to ascertain the **friction cut-off** value between material and belt (perhaps also when braking)

Friction cut-off
Material Belt

$$a_A \leq (\mu_1 * \cos \delta_{max} - \sin \delta_{max}) * g \quad \text{(m/s}^2\text{)}$$

δ	(°)	angle of slope of the installation $\delta > 0$ inclined conveyors $\delta < 0$ declined conveyors
μ_1	(-)	friction value belt/material μ_1 = 0.5 à 0.7
a_A	(m/s^2)	acceleration (calculationl, see Page 12.7)

Start-Up Factor k_A

The peripheral force F_U increases at start-up. To take account of the start-up resistances a factor k_A is applied resulting in peripheral force F_A. The value of k_A is dependent on the drive unit motor/coupling used.

Rigid Couplings

Rigid couplings are used on small installations up to approx. 30kw motor power and squirrel cage type. The peripheral force F_A is derived from the installed motor power.

$$F_A = k_A * \frac{P_N * \eta * 1000}{v} \qquad (N)$$

P_N	(kW)	installed motor power
η	(-)	degree of drive efficiency
v	(m/s)	speed
k_A	(-)	start-up factor K_A = 2.0 to 2.2 if $F_A > F_U * 2.5$ then F_A should = $2.5 * F_U$, (cut-off torque)

Flexible Coupling

With this type of coupling the start-up torque of the motor is reduced.

$$F_A = k_A * F_U * \frac{P_N}{P_M} \qquad (N)$$

P_M	(kW)	required motor power
k_A	(-)	start-up factor, k_A = 1.2 to 1.6

The installed motor power may be substantially higher than is necessary therefore F_A should be less than or equal to $F_U * 2.5$.

Hydraulic Coupling

With hydraulic couplings the start-up torque can be regulated and limited to the desired start-up factor and by doing this, depending on the size of the installation, a near steady state running condition can be achieved.

$$F_A = k_A * F_U \qquad (N)$$

k_A	(-)	start-up factor, k_A = 1.2 to 1.5 regulated by the volume of oil in the working circuit.

At start-up and braking the acceleration and deceleration forces of the mass being moved have to be considered. If the drive elements (motor, coupling, gears) and the non-driven pulley resistances are higher than the remaining resistances, this has to be taken into account.

Acceleration a_A

$$a_A = \frac{F_A - F_U}{L * (m'_{Red} + 2 * m'_G + m'_L) + \Sigma m'_{Ared} + \Sigma m'_{red}} \qquad (m/s^2)$$

m'_{Red}	(kg/m)	Reduced mass of idlers $m'_{Red} \approx 0.9 * m'_R$
m'_{Ared}	(kg/m)	Reduced mass of drive elements
m'_{red}	(kg/m)	Reduced mass of non-driven elements (bend pulleys etc)

Acceleration time $\quad t_A = v / a_A \quad$ (s)

Acceleration Distance $\quad s_A = v * t_A / 2 \quad$ (m)

If the acceleration time t_A is greater than 10 s, in the case of squirrel cage motors a start-up coupling is necessary because of thermal effects.

Peripheral Force F_B on Braking

The **peripheral force F_B on braking** has to be determined for the most adverse circumstances. The **braking time t_B** has to take care of safety requirements (emergency stop) or the **braking distance s_B** because of the possibility of continuing flow due to over-run of the conveyor.

Delay $a_B = v / t_B$ (m/s^2)

Overun Distance $s_B = v * t_B / 2$ (m)

Peripheral Force F_B

$$F_B = a_B * L * (m'_{Red} + 2 * m'_G + m'_L) + F_U + F_T \quad (N)$$

F_T (N) If a great proportion arises from the non-braked elements these forces must be added.

$$F_T = a_B * \Sigma m_{red} \quad (N)$$

m_{red} (kg) Reduced masses of the non-braked elements or pulleys

Braking torque of the Motor

$$M_M = 975 * g * \frac{P_N}{n} \quad (Nm)$$

The brake has to take account of the braking duty and delay.

Delay Moment

$$M_V = \frac{F_B * \mu_B}{i} \quad (Nm)$$

Braking Torque of the Motor

$$M_{Br} = M_M + M_V \quad (Nm)$$

n	(T/min)	Motor revolutions
μ_B	(-)	Friction value of brake disc
F_B	(N)	Peripheral force on braking
i	(-)	Transmission ratio

Drive Motor at Pulley

$$P_T = \frac{F_U * v}{1000} \quad (kW)$$

Required Motor Power

$$P_M = P_T / \eta \quad (kW) \quad F_U > 0$$

Motor Power on Braked Installation

$$P_M = P_T * \eta \quad (kW) \quad F_U < 0$$

Installed Motor Power

P_N (kW) selected motor power

Degree of Efficiency η (Values)

Types of Drive	η
Wormgear drive Toothed chain drive V-Rope drive Pulley motor Normal coupled drive Geared and hydraulic coupling Hydraulic motor	0.7 - 0.8 0.9 - 0.95 0.95 0.96 0.94 0.90 0.86
Braked installations	0.95 - 1.0

Nominal Power P_N (kW)

Selected according to DIN 42973

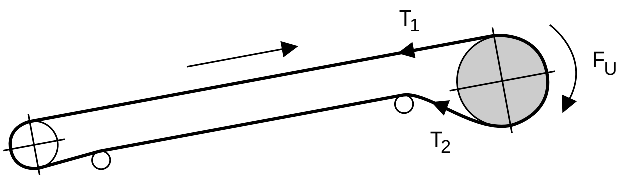
1.5	2.2	3.0	4.0	5.5	7.5	11	15	18.5
22	30	37	45	55	75	90	110	132
160	200	250	315	400	500	630		

Drive System

After the peripheral force F_U has been ascertained and hence the drive power P_T requirement, **the drive system,** position and number of motors can be established.

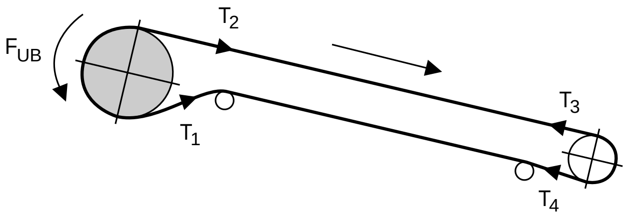
Single Pulley Head Drive

The single pulley head drive is the most common all drive systems. With horizontal and inclined conveyors these give the most favourable belt stresses and with declined conveyors with gradients of 1° to 2° providing the peripheral force remains positive.



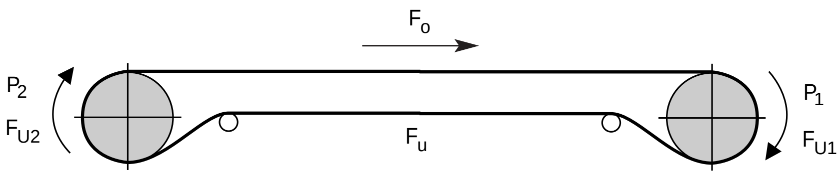
Tail Drive

The preferred location of the drive on a **declined conveyor** is at the tail end. The drive becomes a generator and acts as a brake.



Head and Tail Drive

When a relatively high proportion of motional resistance is due to the return run, this drive system provides the most favourable belt stress condition.



Head and Tail Drive

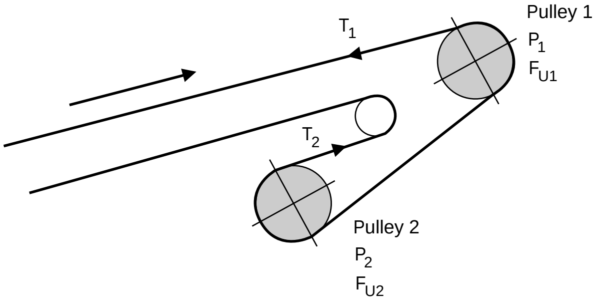
The most favourable distribution of the total power is as follows:

Total Power	$P_T = P_1 + P_2$	(kW)
Distribution	$\frac{P_1}{P_2} \approx \frac{F_o \text{ (Primary force)}}{F_u \text{ (Secondary force)}} = x$	
Motor 2	$P_2 = P_T / (x + 1)$	(kW)
Peripheral force Pulley 2 Peripheral force Pulley 1	$F_{U2} = F_U / (x + 1)$ $F_{U1} = F_U - F_{U2}$	(N) (N)

With identical drives i.e. $P_1 = P_2$, T_4 may be less than T_{min} and the pretension has to be increased accordingly.

Dual Pulley Head Drive

With high drive powers it is often necessary to distribute the available peripheral force over two pulleys. The angle of wrap can thus be increased to approx 420°.



The most favourable drive distribution is as follows:

Total Power	$P_T = P_1 + P_2$	(kW)
Distribution	$\frac{P_1}{P_2} \approx \frac{F_{U1}}{F_{U2}} \approx \frac{e^{\mu\alpha_1} - 1}{e^{\mu\alpha_2} - 1} \; e^{\mu\alpha_2} = x$	Value for factor x see Table
Motor 2	$P_2 = P_T / (x + 1)$	(kW)
Peripheral force Pulley 2 Peripheral force Pulley 1	$F_{U2} = F_U / (x + 1)$ $F_{U1} = F_U - F_{U2}$	(N)

As a rule **the friction coefficient μ** is taken to be the same for both pulleys.

Factor x for various drive conditions

Pulley 2		**Angle of Wrap α_1**			Pulley 1		
μ = 0.25		**160°**	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.00	2.20	2.40	2.60	2.80	3.00
	170°	1.90	2.10	2.30	2.48	2.67	2.86
	180°	1.83	2.02	2.20	2.38	2.57	2.75
	190°	1.77	1.95	2.12	2.30	2.48	2.65
	200°	1.71	1.89	2.06	2.23	2.40	2.57
	210°	1.67	1.83	2.00	2.17	2.33	2.50
μ = 0.3		**160°**	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.31	2.53	2.76	3.00	3.26	3.53
	170°	2.22	2.43	2.66	2.89	3.14	3.40
	180°	2.15	2.35	2.57	2.79	3.03	3.28
	190°	2.08	2.28	2.48	2.70	2.93	3.18
	200°	2.02	2.21	2.41	2.62	2.85	3.08
	210°	1.97	2.15	2.35	2.55	2.77	3.00
μ = 0.35		**160°**	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.66	2.92	3.20	3.51	3.84	4.18
	170°	2.56	2.82	3.10	3.39	3.70	4.03
	180°	2.48	2.74	3.00	3.29	3.59	3.91
	190°	2.41	2.66	2.92	3.19	3.48	3.80
	200°	2.35	2.58	2.84	3.11	3.39	3.70
	210°	2.29	2.52	2.77	3.03	3.31	3.61

In practice the most favourable distribution of power does not often occur because the norm is to use equal drive units. As a result of this the drive pulleys are not equally utilized. **With unequal distribution of the total power,** the drive powers for individual pulleys are thus:

Total Power	$P_T = P_1 + P_2$	(kW)
Distribution	$\frac{P_1}{P_2} \approx \frac{F_{U1}}{F_{U2}} = x$	
Peripheral force Pulley 2	$F_{U2} = F_U / (x + 1)$	(N)
Peripheral force Pulley 1	$F_{U1} = F_U - F_{U2}$	(N)

Before the belt tensions are determined it necessary to check whether the peripheral force of Pulley 1 or peripheral force of Pulley 2 are fully utilized.

Distribution proportion	Utilization of peripheral force	Belt stresses
$P_1/P_2 > x$	Pulley 1 full utilization	$T_2 = F_{U1} * c_{12} - F_{U2}$ $T_1 = T_2 + F_U$
$P_1/P_2 \leq x$	Pulley 2 full utilization	$T_2 = F_{U2} * c_{22}$ $T_1 = T_2 + F_U$

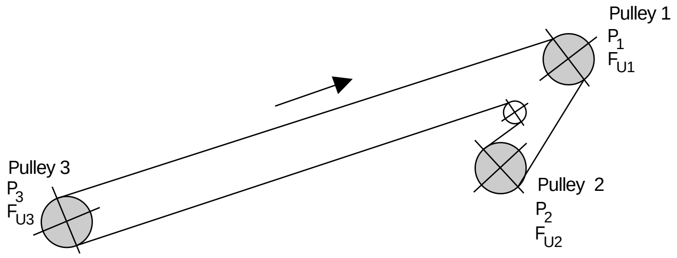
$$c_{12} = \frac{1}{e^{\mu\alpha_1} - 1} \qquad c_{22} = \frac{1}{e^{\mu\alpha_2} - 1}$$

Drive Factor Pulley 1 — Drive Factor Pulley 2

The distribution of peripheral forces at start-up and determination of belt stresses follows analogously.

Dual Pulley Head and Tail Drive

This drive system is treated in the same way as a dual pulley head drive system.



The peripheral forces can be ascertained as follows:

Total Power	$P_T = P_1 + P_2 + P_3$	(kW)
Peripheral Forces	$F_U = F_{U1} + F_{U2} + F_{U3}$	(N)
Distribution Head and Tail	$\frac{P_1 + P_2}{P_3} \approx \frac{F_{U1} + F_{U2}}{F_{U3}} \approx x$	
Peripheral for Pulley 3	$F_{U3} = F_U / (x + 1)$	(N)
Peripheral force at Head	$F_{U1} + F_{U2} = F_U - F_{U3} = F_k$	(N)
Distribution Pulley 1 and 2	$\frac{P_1}{P_2} \approx \frac{F_{U1}}{F_{U2}} = w$	
Peripheral Force Pulley 2	$F_{U2} = F_k / (w + 1)$	(N)
Peripheral Force Pulley 1	$F_{U1} = F_k - F_{U2}$	(N)

The calculation of belt stresses is as described under Dual Pulley Head Drive.

Other Drive Systems

Besides the drive systems described further drive distributions and arrangements are possible. These are treated in a similar fashion.

Belt Tension

From the calculated peripheral forces F_U and F_A maximum belt tensions T_1 and T_2, T_{1A} and T_{2A} can be ascertained for the steady state and non-steady state running conditions respectively.

$$T_1 = F_U * c_1 \quad (N)$$
$$T_2 = F_U * c_2 \quad (N)$$

Steady State Running

$$T_{1A} = F_A * c_{1A} \quad (N)$$
$$T_{2A} = F_A * c_{2A} \quad (N)$$

Non-Steady State Running

$$c_1 = 1 + \frac{1}{e^{\mu\alpha} - 1}$$

Drive Factor tight side

$$c_2 = \frac{1}{e^{\mu\alpha} - 1}$$

Drive Factor slack side

At start-up of the installation the friction value increases slightly for a short period of time, therefore calculate thus.

Friction value at start-up $\mu_A \approx \mu + 0.05$

$$c_{1A} = 1 + \frac{1}{e^{\mu_A\alpha} - 1}$$

Drive Factor tight side at start-up

$$c_{2A} = \frac{1}{e^{\mu_A\alpha} - 1}$$

Drive Factor slack side at start-up

Belt Tension Correction

Belt tensions thus calculated are not yet the belt tensions that will actually occur when working. They may be used for estimating purposes on small installations. For exact and ongoing calculations the following corrections are necessary.

Depending on the type of take-up system, a correction of the belt tension is necessary.

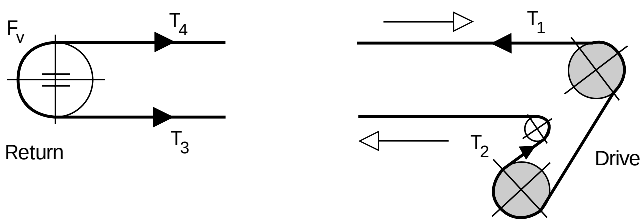
1. Correction with fixed take-up

With a fixed take-up the tension pulley is at a fixed location after tensioning. The centre to centre distance does not change. Depending on changing loads, the belt stretches and contracts within its elastic limits. The total elongation is constant. For all working condition, **the sum of the belt tensions has to be constant.**

$$\Sigma T_{Working} = \Sigma T_{Start\text{-}Up} = \text{Constant}$$

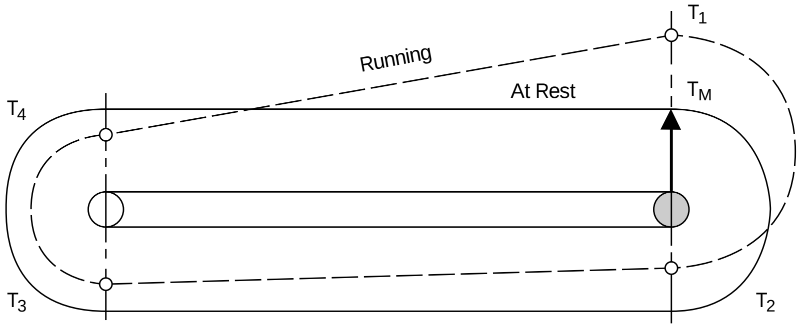
Location of take-up Pulley

The pre-tension has to be calculated and set so as to take account of the non-steady state running conditions such as start-up and braking. The location of the take-up pulley does not matter. It can be at the discharge end, the tail end or the middle of the conveyor installation.



Stress Distribution

The belt tensions on the bend pulleys are higher at rest than they are when running.



Belt tension at rest $T_M = \Sigma T_B / 4$ (N)

Average belt tension at start-up $T = \Sigma T_A / 4$ (N)

Because the length of the belt does not alter, the **take-up is constant**.

Take-up

$s_B = s_A$ (mm)

S_B (mm) Take-up when running
S_A (mm) Take-up at start-up

The pre-tension has to cater for start-up therefore all belt tensions increase by the difference between running and start-up conditions.

Correction value
for fixed take-up

$$\Delta T = \frac{\Sigma T_A - \Sigma T_B}{4} \quad (N)$$

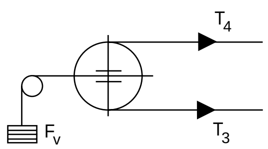
With a fixed take-up, all belt tensions when running are increased by the correction value T.

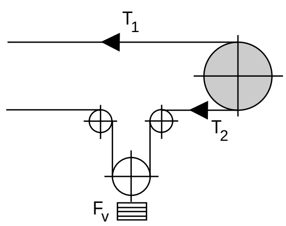
1. Correction with moveable take-up

With the movable take-up the belt length is not constant. With changes in belt stress the take-up weight adjusts to the various belt elongations/length changes. The belt tensions at the take-up location are always the same. The take-up weight has to be calculated to cater for the non-steady state running conditions such as start-up and braking. The belt tensions at the take-up location are always higher than those necessary for the steady state running condition.

Location of take-up weight

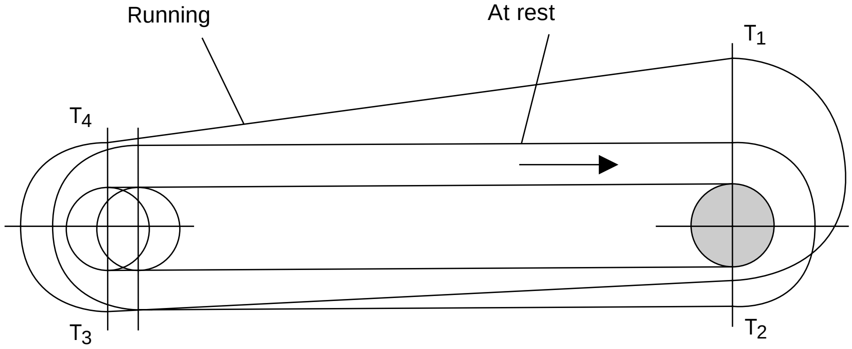
The take-up weight can be installed at the drive, the tail or at any location.





Stress Distribution

The size of the take-up weight depends on the location of the take-up system.



Take-up tension $F_V = T_3 + T_4$ or $F_V = 2 * T_2$ (N)

Take-up weight $G_V = F_V / g$ (kg)

Because the tension weight is effective both at rest and during running, all belt tensions increase by the correction value ΔT.

Correction Value $\Delta T = T_{A2} - T_2$ (N)

2. Correction

The control of minimum belt tension. T_{min} is necessary in order that the belt sag between carrying idlers at the loading point does not exceed a certain value.

Minimum belt tension

$$T_{min} = \frac{(m'_L + m'_G) * l_o * g}{8 * h_{rel}} \quad (N)$$

$h_{rel} =$	h / l_o	Sag ratio
	$h_{rel} =$	0.005 to 0.015 for carrying side
	$h_{rel} =$	0.020 to 0.030 for return side
l_o	(mm)	Distance between the carrying idlers
h	(mm)	Belt sag (see Page 11.2)

If the loading $T_4 < T_{min}$ the belt would sag between the idlers. To avoid this, all belt tensions have to be increased by the correction value ΔT.

Correction Value $\Delta T = T_{min} - T_4$ (N)

Sequential Calculation

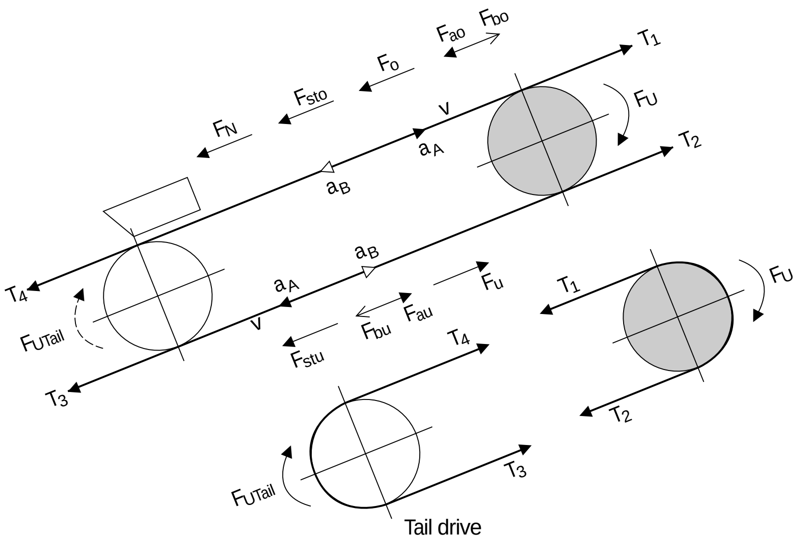
The tensions T_1 to T_4 are best determined according to the sequential calculation principle. This method enables the tension to be ascertained at any given point on the installation as well as under certain working conditions such as start-up and braking.

Force Distribution

The forces that affect the belt are determined according to their size and direction in which they operate. Those are the secondary resistances which arise at the loading point, friction resistances, the slope resistance and the inertial resistances at start-up and braking.

The force diagram can be applied to the carrying side, return side as well as to drive and return pulleys.

The point tensions T_1 to T_4 can be calculated from the condition $\Sigma F = 0$.



Carrying side	$0 = T_1 - T_4 - F_N - F_o - F_{sto} - F_{ao}$
Return side	$0 = T_3 - T_2 - F_u + F_{stu} - F_{au}$
with Head Drive	$0 = T_1 - F_U - T_2$
with Tail Drive	$0 = T_4 + F_{U\ Heck} - T_3$

Individual Resistances

The individual resistances are calculated as follows:

Main resistance	$F_H = f * L * g * (m'_R + 2 * m'_G * m'_L)$
Secondary resistance	$F_N = (C - 1) * F_H$
Friction resistance	
Carrying side	$F_o = f * L * g * (m'_{Ro} + m'_G + m'_L)$
Return side	$F_u = f * L * g * (m'_{Ru} + m'_G)$
Slope resistance	
Carrying side	$F_{sto} = H * g * (m'_G + m'_L)$
Return side	$F_{stu} = H * g * m'_G$
Inertial resistance	
Carrying side	$F_{ao} = L * a * (m'_{Redo} + m'_G + m'_L)$
Return side	$F_{au} = L * a * (m'_{Redu} + m'_G)$

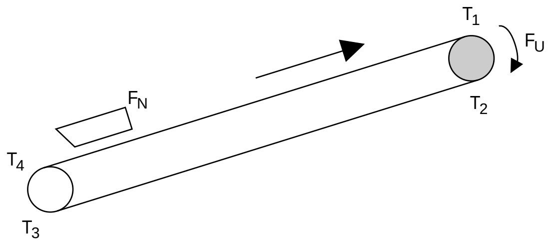
a	(m/s^2)	Acceleration a_A or deceleration a_B
m'_{Redo}	(kg/m)	Reduced mass of carrying idlers
m'_{Redu}	(kg/m)	Reduced mass of return idlers

Reduced mass $m'_{Red} \approx 0.9 * m'_R$

For other formulae symbols see Page 12.2.

Single Pulley Head Drive

Determination of the point tensions (T_1 to T_4 and T_{A1} to T_{A4} with the help of individual resistances (for peripheral force $F_U \geq 0$. positive).



$$
\begin{aligned}
T_2 &= F_U * c_2 \\
T_3 &= T_2 + F_u - F_{stu} \\
T_4 &= T_3 \\
T_1 &= T_4 + F_N + F_o + F_{sto}
\end{aligned}
$$

Belt Tension (start-up)

$$
\begin{aligned}
T_{A2} &= F_A * c_{2A} \\
T_{A3} &= T_{A2} + F_u - F_{stu} + F_{au} \\
T_{A4} &= T_{A3} \\
T_{A1} &= T_{A4} + F_N + F_o + F_{sto} + F_{ao}
\end{aligned}
$$

Non-steady state working

Control

$$
\begin{aligned}
T_2 &= T_1 - F_U \\
T_{A2} &= T_{A1} - F_A
\end{aligned}
$$

Correction of belt tensions

The tensions T_1 to T_4 and T_{A1} to T_{A4} must be increased by ΔT, if the following conditions apply:

1. With fixed take-up if $\Sigma T_A > \Sigma T$
2. With movable take-up if $T_{A2} > T_2$
3. With minimum belt tension if $T_{min} > T_4$

For correction formulae see Pages 12.13 -12.15.

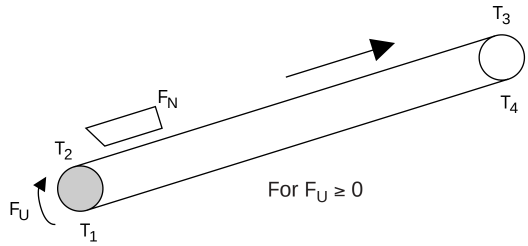
Braking

To calculate braking forces the peripheral force F_B on braking has to be determined. The sequential calculation for non-steady state working then begins with

$$T_{B2} = F_B * c_{2A}$$

The inertial mass resistance F_{bo} and F_bu is determined with the deceleration a_B. The belt tension T_{B2} lies on the tight side of the drive pulley.
Otherwise the sequential calculation can be done along the same lines.

Tail Drive



$T_2 = F_U * c_2$
$T_3 = T_2 + F_u + F_o + F_{sto}$
$T_4 = T_3$
$T_1 = T_4 + F_u - F_{stu}$

Belt tensions steady state working

$T_{A2} = F_A * c_{2A}$
$T_{A3} = T_{A2} + F_N + F_o + F_{sto} + F_{ao}$
$T_{A4} = T_{A3}$
$T_{A1} = T_{A4} + F_u - F_{stu} + F_{au}$

Belt tensions non-steady state working (start-up)

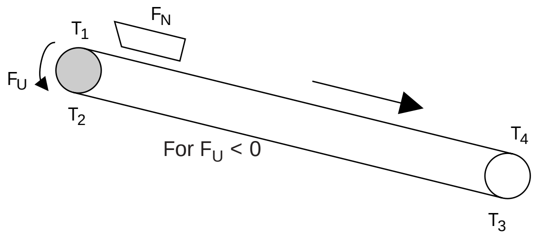
Control

$T_2 = T_1 - F_U$
$T_{A2} = T_{A1} - F_A$

Correction of belt tensions

Tensions T_1 to T_4 and T_{A1} to T_{A4} must be corrected as described under single pulley head drive.

Tail Drive (Braking)



$T_2 = F_U * c_2$
$T_3 = T_2 - F_u - F_{stu}$
$T_4 = T_3$
$T_1 = T_4 - F_N - F_o + F_{sto}$

Belt tensions steady state working

$T_{A2} = F_A * c_{2A}$
$T_{A3} = T_{A2} - F_u - F_{stu} + F_{au}$
$T_{A4} = T_{A3}$
$T_{A1} = T_{A4} - F_N - F_o + F_{sto} + F_{ao}$

Belt tensions non-steady state working (start-up)

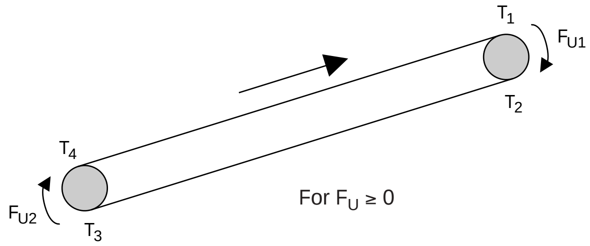
Control

$T_2 = T_1 - F_U$
$T_{A2} = T_{A1} - F_A$

Correction of belt tensions

The tensions T_1 to T_4 and $T_{A1}$to T_{A4} must be corrected as described under single pulley head drive.

Head and Tail Drive



$$
\begin{aligned}
T_2 &= F_{U1} * c_2 \\
T_3 &= T_2 + F_u - F_{stu} \\
T_4 &= T_3 - F_{U2} \\
T_1 &= T_4 + F_N + F_o + F_{sto}
\end{aligned}
$$

Belt tensions steady state working

$$
\begin{aligned}
T_{A2} &= F_{A1} * c_{2A} \\
T_{A3} &= T_{A2} + F_u - F_{stu} + F_{au} \\
T_{A4} &= T_{A3} - F_{A2} \\
T_{A1} &= T_{A4} + F_N + F_o + F_{sto} + F_{ao}
\end{aligned}
$$

Belt tensions non-steady state working (start-up)

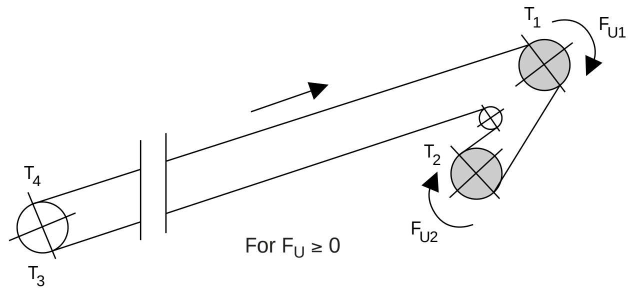
Control

$$
\begin{aligned}
T_2 &= T_1 - F_U \\
T_{A2} &= T_{A1} - F_A
\end{aligned}
$$

Correction

The tensions T_1 to T_4 and T_{A1} to T_{A4} must be corrected as described under single pulley head drive.

Dual Pulley Head Drive



$$
\begin{aligned}
T_2 &= \text{refer to page 12.13} \\
T_3 &= T_2 + F_u - F_{stu} \\
T_4 &= T_3 - F_{U2} \\
T_1 &= T_4 + F_N + F_o + F_{sto}
\end{aligned}
$$

Belt tensions steady state working

$$
\begin{aligned}
T_{A2} &= \text{refer to page 12.13} \\
T_{A3} &= T_{A2} + F_u - F_{stu} + F_{au} \\
T_{A4} &= T_{A3} - F_{A2} \\
T_{A1} &= T_{A4} + F_N + F_o + F_{sto} + F_a
\end{aligned}
$$

Belt tensions non-steady state working (start-up)

Control

$$
\begin{aligned}
T_2 &= T_1 - F_U \\
T_{A2} &= T_{A1} - F_A
\end{aligned}
$$

Belt Tension T_2

The belt tension T_2 is dependent on the destribution of the start-up force i.e. of the utilization of the peripherical force by drive pulley 1 or 2.
Refer to page 12.12.

Correction of Belt Tension

The tensions T_1 to T_4 and T_{A1} to T_{A4} must be corrected as described under single pulley head drive.

Section Calculation

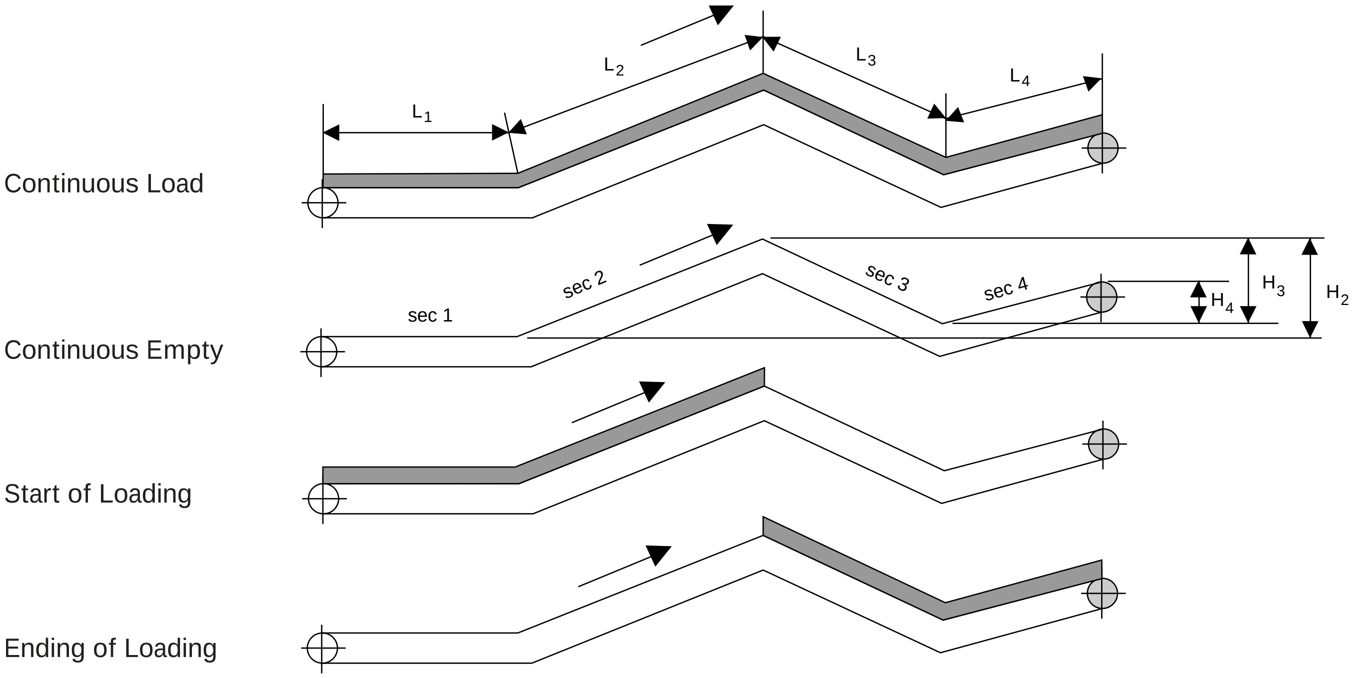
A belt conveyor can be divided into sections according to certain working and topographical conditions with differing lengths and heights. The resistance to motion for these sections can be calculated as previously with loading point and discharge point or other special resistances being apportioned to the section where the resistance occurs.

Total Resistance
Peripheral Force

$$F_{tot} = F_U = F_{sec1} + F_{sec2} + ... \qquad (N)$$

With varying conditions and changing loads on certain sections, greater or lesser resistance to motion can occur, e.g. belt force with continuous loading. To determine these points, the resistances have to be ascertained under various working conditions such as load run-on and run-off conditions and running unloaded.

At the same time it can be established whether the installation has to be driven by or braked by the motor.



To determine the individual resistances the values of L and H have to be substituted in the formulae by L_x and H_x of the respective sections.

Under certain circumstances the following working conditions may have to be examined:

- Inclined loaded sections.
- Declined loaded sections.

Nominal breaking strength

After the belt width B (mm) has been established, the maximum belt tension for the steady state and non-steady state working conditions is calculated, the **nominal breaking strength k_N** of the belt can be ascertained and the **belt type chosen.**

Nominal belt strength steady state working

$$k_N = \frac{T_{max} * S_B}{B * c_v} \quad \text{(N/mm)}$$

Nominal belt strength non-steady state running

$$k_N = \frac{T_{Amax} * S_A}{B * c_v} \quad \text{(N/mm)}$$

T_{max}	(N)	maximum belt tension steady state working
T_{Amax}	(N)	maximum belt tension non-steady state working
c_v	(-)	factor for loss of strength at the splice joint. Values see Page 11.9
B	(mm)	belt width

The nominal belt strength is equivalent to the minimum breaking strength rounded up and observing other selection criteria results in the belt type.

Safety Factor

The safety factor of the selected belt is determined and verified. For certain running conditions there are laid down minimum safety values.

$$S_B = \frac{k_N * B * c_v}{T_{max}} \quad \text{(-)}$$

Steady state

$$S_A = \frac{k_N * B * c_v}{T_{Amax}} \quad \text{(-)}$$

Non-steady state

For **textile and steel cord belts** the following **safety factor values** can be used:

Working Conditions	Drive Conditions Steady state S_B	 Non-steady state S_A
Favourable: few loading changes, large pulleys, few bending stress changes	6.7	4.8
Normal	8.0	5.4
Unfavourable: many loading changes, small pulleys, frequent bending stress changes	9.5	6.0

For mechanical joints and cold splices other factors apply (see manufacturers recommendations).

Selection of Belt Type

For the selection and correct dimensioning of the belt type a number of **criteria** and **principal influences** need to be taken into account. Firstly is the maximum belt tension decisive in the dimensioning of the carcase. It is essential to observe the factors of safety for both the running and start-up conditions. As well as these calculations, further conditions which could influence the strength, number of plies and quality of the belt have to be considered.

Belt Nominal Strength

The belt nominal strength K_N is rounded up to the **next standard belt type.** When rounding down, the safety factor needs to be checked. Further Criteria have to be observed.

Belt Safety

With the **safety factor S** the influence of the following are considered:
- Additional stresses at curves
- Overload at start-up
- Tensile strength loss due to disproportionate contribution of the plies
- Strength loss due to impact
- Deleterious effects from penetration, ageing etc.

The values according to the table on page 12.21 should be observed. For the start-up condition the value T_{Amax} is used when calculating because at start-up the friction values of the drive pulley increase for a brief period of time. With smaller installations it is not always necessary to verify the calculated value of S_A.

Load support

Certain characteristics of the material load may in turn affect the belt width dimensioning. The wider the belt and the heavier the load the higher the strength and the thicker the belt construction irrespective of the calculated strength requirement.

With the help of the **Belt Selection Table** (see page 12.25) the appropriate belt type can be determined depending on the **belt width, lump size** and **bulk density.** With a belt of a lesser type than that recommended there is no guarantee of stable running.

Number of Plies

In the case of multiply belts such as the SUPERFORT range, the strength of one ply is lost at the joint. With highly stressed and impacted belts this has to be taken into account additionally. In general the carcases of wider belts should contain more plies. This does not apply to special constructions such as DUNLOFLEX and TRIOFLEX.

Impact energy

The impact energy due to the **free fall height** of material, can be reduced at the loading point by means of deflection plates, chute and screen bars etc. If this is not possible or only so to a limited extent more or higher strength plies should be considered. In special cases a breaker ply is embedded in the carrying side cover to project the carcase.

High Temperature

When **hot materials** are to be conveyed the appropriate heat resistant quality rubber covers have to be selected. In so doing it is necessary to differentiate between the temperature of the load and the permissible temperature of the belt. The **carrying side cover** should be thicker than normal and stronger plies specified. In this way the heat absorption capability is increased and the elongation due to the influence of heat is reduced.

Further influences are:
The **belt speed** and hence the heating up and cooling down periods.
The **material lump size** and the contact density on the carrying side cover.
Installation location (inside or outside, open or closed) and hence perhaps the effect of a heat concentration.

Low Temperature

With extreme low temperature conditions account must be taken of the higher flexing resistance of the belt. Select the fiction factor f accordingly. The cover quality should be one containing a high percentage of natural rubber. Depending upon the compounding of the rubber, belts can be used down to -30°C (maximum to -40°C). PVC belts down to -15°C.

Troughability

With thicker and stronger belts it is necessary to verify whether the belt will conform to the troughing angle of the idlers. For good troughing conditions approx 40% of the belt width when running empty should be making tangential contact. Approximately 10% should make contact with the centre carrying idler roller (Recommendation see Page 12.29).

Cover Thickness

When selecting belt covers, not only the quality appropriate for the load material has to be considered, but it is also necessary to ensure there is sufficient thickness of cover on both the carrying and pulley sides of the belt. If the carrying side cover is much thicker than 3 times the pulley side cover there is the risk of the belt curling at its edges. (Recommendations see Page 12.30).

Steep Angle Conveying

With steep inclined conveying using profiled belts from the Chevron or High Chevron ranges, a carcase should be selected that is 1 or 2 plies stronger than calculated or in accordance with the belt selection table. By doing this the running stability is improved especially under wet and frosty conditions. This can also be an advantage with smooth surface belts which convey steeper than 12°.

Oil & Fat Effects

Normal cover qualities tend to swell even if the load to be conveyed has only the smallest percentage of oil or fats and the rubber tends to soften. Account has to be taken as to whether the oil or fat is mineral or vegetable based. Also the temperature and possibly acidity has to be taken into consideration.

Food Stuffs

To convey food stuffs, the cover quality must comply with National and International regulations. One has to differentiate between unpacked and packed food-stuffs. The most important regulations at present are:

- FDA/USDA: Food and Drug Administration U.S.
- Unbedenklichkeitsempfehlung XXVII. BGA
- British Code of Practice for Food Contact Applications.

Chemical Effects

If materials contain acids or alkalis the appropriate quality has to be selected. The **concentration** and **temperature** play a decisive part. With increasing concentration and temperature the durability of the covers declines. They are attacked by chemicals and swell. Under certain conditions regard has to be paid to the chemical resistance of the carcase.
Rubber and PVC react differently.

Safety Regulations

Safety regulations can be important criteria in selecting the belt particulary with installations where there is a risk of fire or explosions.
Some requirements demand:

- **Flame resistant quality** (to DIN 22103)

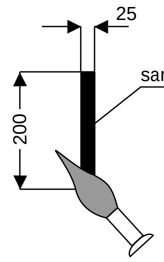
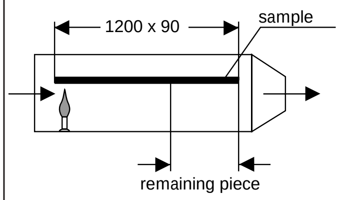
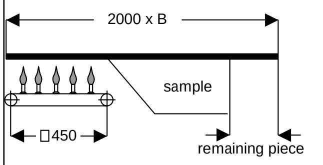
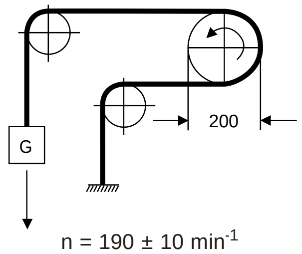
A belt is classed as flame resistant if it does not burn or does not reignite after the source of the flame has been removed.

- **Electrical conductivity (Antistatic)**

A conveyor belt is classed as being antistatic if the average value of the surface resistance of the covers does not exceed $3 * 10^8$ Ohm.

- **Self extinguishing quality (Underground mines)**

The definition of self extinguishing conveyor belting means a quality which does not promote the spread of flame from the initial fire. To determine the combustion characteristics a number of tests have to be conducted.

<table>
<tr><td>Flame Test</td><td>Gallery Test</td><td>Propane Burner Test</td><td>Drum Friction Test</td></tr>
<tr><td>DIN 22103 (12.74)</td><td>DIN 22118 (12.74)</td><td colspan="2"></td></tr>
<tr><td>
</td><td>
</td><td>
</td><td>

n = 190 ± 10 min^{-1}</td></tr>
<tr><td colspan="4">Number of samples</td></tr>
<tr><td>6 (3 + 3)</td><td>5</td><td>1</td><td>6</td></tr>
<tr><td colspan="4">Time in Flame</td></tr>
<tr><td>45 s</td><td>15 min</td><td>10 min</td><td>–</td></tr>
<tr><td colspan="4">Requirements</td></tr>
<tr><td>a) Sum of after flame < 45 s
b) Individual <15
c) No ignition after air current</td><td>a) Average value of residual length ≥ 400mm
b) No sample residual length 0 mm</td><td>Residual piece full width not affected</td><td>a) No reignition
b) With flame glow preventative substitute (slipobservation necessary)</td></tr>
</table>

- **Occupational hygiene characteristics.**

Decomposition products must not come into contact with the skin, must not be harmful to the respiratory system nor increase filter resistance as a result of clogging.

The appropriate data and testing regulations are laid down in National and International Standards.

Design of the Installation

The course of the installation having certain dimensions, troughing, transition, vertical curves, horizontal curves and belt turnover can be decisive when selecting the belt. See Appendix 1.

Method of Tensioning Take-up Travel

The type of tensioning system and the amount of available take-up travel can be important considerations when selecting the belt type having particular reference to elongation behaviour.

Belt Reference Number

In the first instance **the belt is selected** according to the tensile strength requirements. In many cases types selected on this basis alone, would be inappropriate because they do not take into account the **belt width, the bulk density** and the **material lump size.** Tracking, sag and transverse stability need to be taken into account. It is recommended that the belt type selected should be verified according to the **belt type reference number.**

Material characteristics		**Indication number of belt type (mm)**									
Bulks Density ρ (t/m^3)	Lump Size (mm)	400	500	650	800	1000	1200	1400	1600	1800	2000
up to 1	≤ 0.5-10	1	1	1	1	2	3	4	4	4	5
1.1 - 1.5	– 100 + 100	1 -	1 -	1 2	2 2	3 3	3 3	4 5	4 5	5 5	5 5
1.6 - 2.5	– 100 + 100	- -	- -	2 3	3 3	3 4	4 4	5 6	5 6	5 6	5 6
over 2.5	– 100 + 100	- -	- -	3 4	3 4	4 5	5 5	6 6	6 7	6 7	6 7

1. With a lump size over approximately 400 mm select one reference number higher possibly.
2. With lump sizes over 100 mm it is desirable that approximately 30% fines be present. Without such fines the load impact energy has to be taken into account.

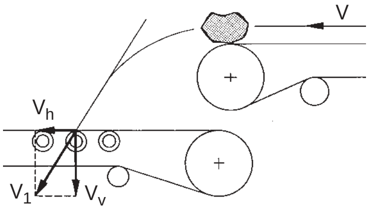
Belt Type Reference	**SUPERFORT** EP Multiply	**DUNLOFLEX** 2 Ply Belt	**TRIOFLEX** 3 Ply Belt
1	EP 200/3	D 200	
2	EP 315/3	D 250 D 315	
3	EP 250/4 EP 315/4 EP 500/3 EP 630/3	D 400	T 250
4	EP 400/4 EP 500/4	D 500	T 315 T 400
5	EP 315/5 EP 400/4 EP 500/5 EP 630/4		T 500
6	EP 500/5 EP 630/5 EP 800/4		T 630
7	EP 800/5		T 800

Instruction for the use of the tables

1. Select the reference number from the first table according to belt width, bulk density and lump size.
2. The second table contains a number of alternatives for the belt type under the same reference number.
3. Final selection may have to take into account other criteria dependent on the working conditions.

Load Demands

At the loading point the demands made of the belt are particularly high. The load has to be accelerated to the belt speed and often at the same time it changes in direction of travel and elevation. With coarse material and high falls the carcase of the belt is subject to additional stresses. These have to be taken into account when dimensioning the belt.



The impact energy E_f is a measure of the demands when loading a belt from a height with large lumps.

$$E_f = m_{st} * g * H_f \qquad (\text{Nm})$$

m_{st}	(kg)	massof a piece (lump)
g	(m/s^2)	acceleration due to gravity
H_f	(m)	delivery height (free fall or via chute)

With the value E_f and with the help of the diagram on Page 12.27. an appropriate belt type can be selected.

Mass of a piece

$$m_{st} = V_{st} * \rho \qquad (\text{kg})$$

V_{st}	(dm^3)	volume of piece (see below)
ρ	(kg/dm^3)	density (see table)

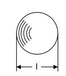
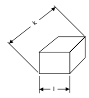
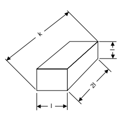
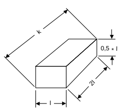
Material Density ρ

Material	Density ρ (kg/dm^3)	Material	Density ρ (kg/dm^3)
Anthracite	1.5 - 1.7	Coke	1.2 - 1.4
Concrete (Gravel)	2.2	Marble	2.5 - 2.8
Brown Iron Stone	3.5 - 4.0	Paper	0.7 - 1.2
Brown Coal (Lignite)	1.2 - 1.5	Coal Briquette	1.3
Manganese Dioxide	5.0	Sandstone	2.2 - 2.5
Earth	1.3 - 2.0	Shale	2.7
Fluorspar	3.2	Slag (Furnace)	2.5 - 3.0
Gypsum Burnt	1.8	Soapstone	2.7
Granite	2.5 - 3.0	Bituminous Coal	1.3 - 1.4
Timber, Green	0.7 - 1.3	Rock Salt	2.3 - 2.4
Timber, Seasoned	0.4 - 1.0	Clay, Dry	1.8
Lime, Burnt	2.3 - 3.2	Clay, Fresh	2.6
Limestone	1.9	Tile	1.4 - 2.0

State of Material

The surface condition of the material influences what effect the impact energy has. With soft rounded material the belt is less stressed compared with that struck by hard sharp edged material.

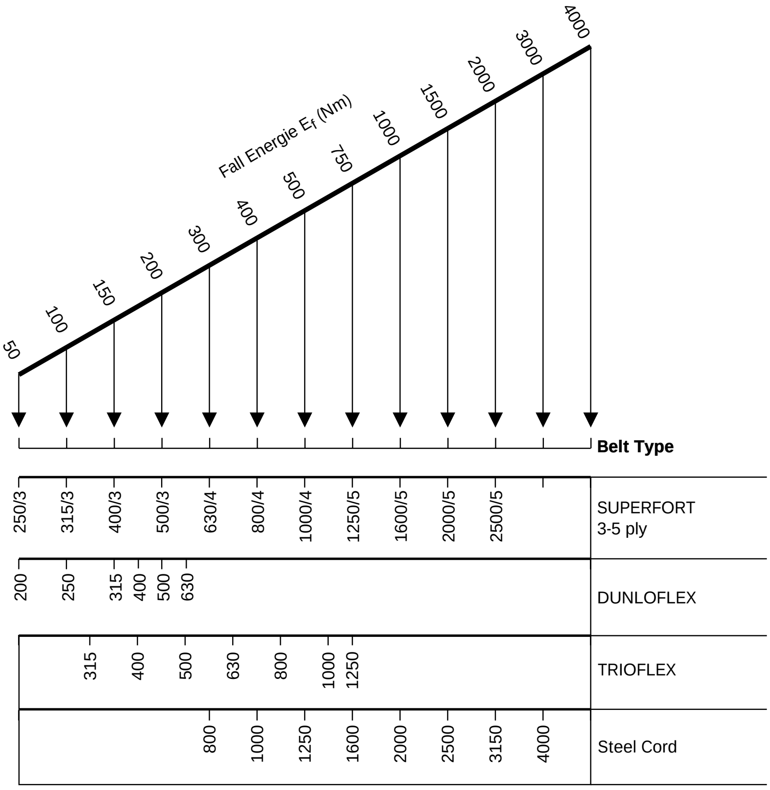
Volumes V_{st}
If the mass of a piece of material is not known, it can be estimated from it volume. With regard to physical shape and by taking the **edge length 1** or the **diagonal measurement corner to corner k,** the volume V_{st} can be established

Physical Shape	Volume V_{st} (dm^3) Edge Length l (dm)	Diagonal Measurement k (dm)
	$V_{st} = 0{,}52 * l^3$	–
	$V_{st} = l^3$	$V_{st} = 0{,}192 * k^3$ $k = \sqrt{3} * l$
	$V_{st} = 2 * l^3$	$V_{st} = 0{,}136 * k^3$ $k = \sqrt{6} * l$
	$V_{st} = l^3$	$V_{st} = 0{,}083 * k^3$ $k = \sqrt{5{,}25} * l$

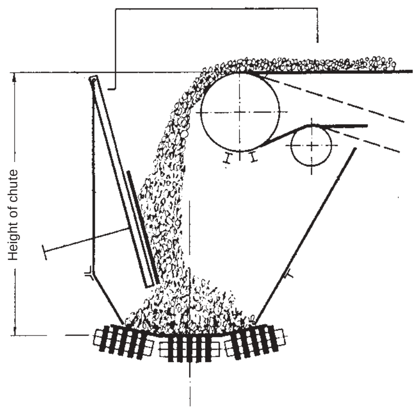
Material	Correction
Angular, irregular such as limestone and gravels ρ ≤ 1.0-2.0 t/m^3	Type according to table. See page 12.27
Rounded, soft such as brown coal ρ = 1.2 t/m^3	1 type lower
Hard, sharp edged such as ores, slag ρ ≥ 2.0 t/m^3	1-2 types higher

Belt Selection

This diagram shows belt types in relation to impact energy and makes an allowance of safety over that of the critical impact energy which could otherwise lead to damage.



Impact Arrangements



To reduce the stress at the loading point there are a number of constructive measures that can be taken such as impact deflection plates, grizzly bars or roller grids, cushioned impact idlers, garland idlers, shaker tables and accelerating belts etc. all of which help to protect the belt.

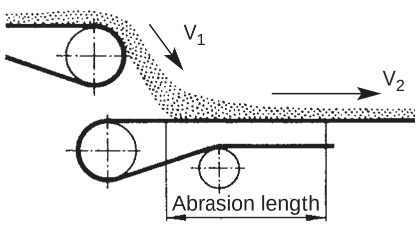
A freely tensioned belt is able to absorb greater impact energy than a supported belt.

Covers

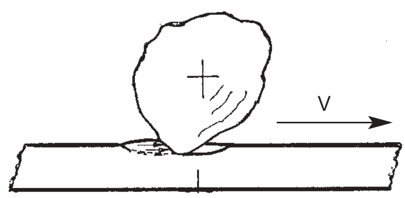
The thickness of the covers also contributes to the reduction of the impact energy and should be calculated to suit. (Recommendations see Page 12.30).

Cover Quality

The **friction due to granulated material acceleration** increases the abrasion and **irregular lump acceleration** causes more cover damage, grooves and tears.



Acceleration relating to friction



Acceleration relating to lump shape

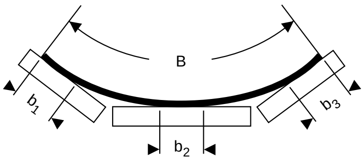
According to the predominating material characteristics either a high abrasion resistant quality or a cut resistant quality has to be used.

Terminal with Slatted Pulley.

Troughability

An appropriate measure of troughability is necessary in order to provide good belt tracking. This depends mainly on the **transverse stiffness** of a belt and its **own weight.** A sufficiently straight run may be expected if approximately 40% of the belt width when running empty, makes contact with the carrying idlers. Approximately 10% should make tangential contact with the centre idler roller.

For normal belt types **minimum belt widths** for a certain troughing angle can be estimated.



$\Sigma b \approx 40\%$ of B
$b_2 \approx 10\%$ of Σb

Minimum Belt Width $B \geq d * c_\lambda * k_W * c_G$ (m)

d (mm) thickness of carcase
c_λ (-) factor for troughing angle
k_W (-) value for carcase material
c_G (-) factor for belt weight

Troughing Angle λ	15°	20°	25°	30°	35°	40°	45°
Factor c_λ	0.64	0.71	0.77	0.83	0.89	0.93	1.00

Material of Weft	B	Z	P	ST
Value k_W	0.44	0.40	0.38	0.29

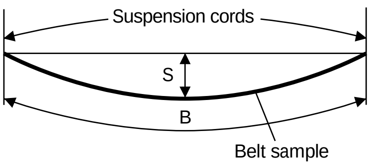
Belt Weight (kg/m^2)	8	10	12	14	16	18	20	22	24	26	28
Factor c_G	0.50	0.47	0.44	0.41	0.38	0.37	0.36	0.355	0.35	0.345	0.34

Test of Troughability

The test of troughability may be carried out according to ISO R 703 (1975) or DIN 22102 (1991).

A 150 mm long full width belt test piece is suspended, friction free, by means of cords whose lengths are according to the belt width. After 5 minutes the sag s is measured and the sag ratio s/B is determined.

Sag Ratio s/B (minimum values)



Troughing Angle λ	20°	25°	30°	35°	40°	45°	50°	55°	60°
Sag s/B	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.23	0.26

Cover Thickness

The cover thicknesses are selected in such a manner as to provide protection to the carcase throughout a belts life and hence economic viability.

The **thickness of the carrying side cover** is mainly dependent upon the characteristics of the load materials (lump size, abrasiveness, density, temperature) and conditions at the loading point.

Chemical effects (oil, acidity etc.) and the effects of material temperature are considered separately. Cover thicknesses should be such that the ratio carrying side to pulley side does not exceed 3:1.

Carrying side cover $d_T = d_m + d_z$ (mm)

d_M (mm) Minimum thickness
For textile belts : 1 to 2 mm depending on fabric
For ST belts: 0.7 * cord diameter, minimum 4mm

d_Z (mm) Addition to minimum thickness
The **value d_z** is determined from the sum of the **rating numbers** for different loading conditions and material characteristics.

Loading Conditions

Loading Conditions	Favourable	Average	Unfavourable
Height of fall (mm)	< 500	500-2000	> 2000
Relative speed $\frac{v - v_o}{v} * 100$ (%)	< 30	30-60	60-100
Damping at loading point	good	normal	moderate

Rating Number

Influence Factor	**Criteria**	**Rating Number**
Loading condition	Favourable Average Unfavourable	1 2 3
Loading cycle $f_B = \frac{2 * L}{v}$ (s)	> 60 20 to 60 < 20	1 2 3
Lump size (mm)	< 50 50 to 100 > 150	1 2 3
Bulk density ρ (t/m^3)	< 1.0 1 to 1.8 > 1.8	1 2 3
Abrasiveness	Slight Average Severe	1 2 3

Value d_z

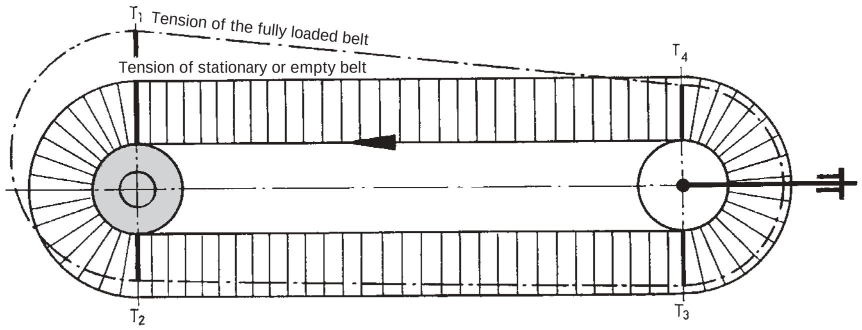
Sum of rating numbers	d_z (mm)
5 to 6	0 to 1
7 to 8	1 to 3
9 to 11	3 to 6
12 to 13	6 to 10
14 to 15	≥ 10

Take-Up

During running the conveyor belt has to accommodate the stress changes that occur between start-up, running, full load running and running empty. These result in elastic elongation of the belt and thus a change in belt length. This difference between the elongation under the installed pre-tension with the conveyor at rest and the total elongation under full load running conditions is the additional running elongation.

The elongation is not spread proportionally over the belt circumference but is dependent upon the effective belt tension and the elongation behaviour of the plies i.e. the carcase materials.
The length change resulting from the pre-tension and the running elongation is accommodated by an appropriately dimensioned take-up unit (see Page 6.1).

The elongation of the belt increases with increasing temperature (hot material and high ambient temperature). The take-up travel is calculated from the relation E = T/B/ϵ (Hooks Law). This is calculated utilizing the average tensions T_{Mo} on the carrying side run and T_{Mu} on the return side run.



$$\epsilon_{ges} = \frac{T_1 + T_2}{2 * B * E} + \frac{T_3 + T_4}{2 * B * E} = \frac{\Sigma T}{2 * B * E} \quad (-)$$

Length Changes

$$\Delta L = \frac{\Sigma T * L * 2}{2 * B * E} \qquad s_p = \frac{\Delta L}{2} \quad (m)$$

Take-up Travel Lengths

$$s_p = \frac{\Sigma T * L}{2 * B * E} = \frac{\Sigma T * L}{2 * B * k_D * k_N} \quad (m)$$

E	(-)	Elastic modulus
B	(m)	Belt width
L	(m)	Conveying length
L_{ges}	(m)	Belt length ≈ 2 * conveying length
k_D	(-)	Elongation value (see Appendix 1.2)
k_N	(N/mm)	Nominal belt strength

For design purposes the following calculation may be used:

Take-Up Travel for Design Purposes

$$S_p = L * \frac{(\epsilon_{bl} + \epsilon_{el})}{100} \quad (m)$$

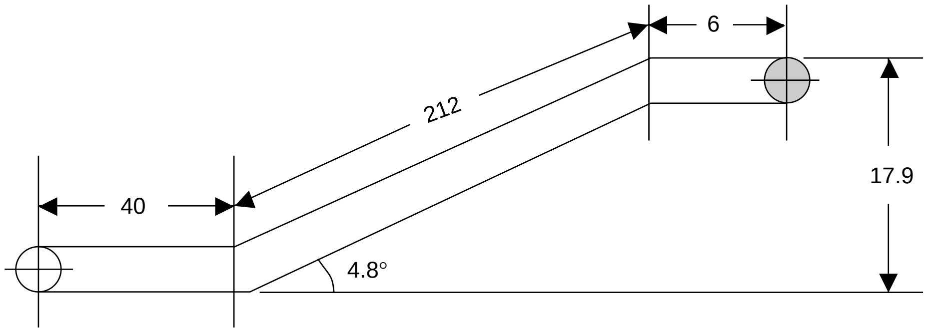
ϵ_{bl}	(%)	Permanent elongation	EP belts 0.2 - 0.4 % ST belts -
ϵ_{el}	(%)	Elastic elongation	EP belts 1 - 2 % ST belts 0.1 - 0.5%

For further information regarding **the elongation behaviour of conveyor belts** see Appendix H.

Service Data

Belt Conveyor with a single pulley head drive conveying sand and gravel, lump size up to 200 mm.

Capacity	Q_m = 1000 t/h	Carrying Idler Rollers	m_{Ro} = 22.3 kg
Bulk Density	ρ = 1.5 t/m^3	Return Idler Rollers	m_{Ru} = 19.3 kg
Degree of Filling	φ = 70%	Surcharge Angle	β = 15°
Conveying Length	L = 258 m	Belt Speed	v = 1.68 m/s
Conveying Height	H = 17.9 m	Layout	horizontal 40 m
Belt Width	B = 1200 mm		inclined 212 m ca 4.8°
Troughing	λ = 35° 3 Roll		horizontal 6 m



Short Calculation

Power Empty + horizontal Conveying

$$P_1 = \frac{c_B * v + Q_m}{c_L * k_f} = \frac{277 * 1.68 + 1000}{52 * 1} = 28.2 \text{ kW}$$

Width Factor	c_B	(see table 11.7)
Length Factor	c_L	(see table 11.8)
Factor	k_f	(see table 11.8)

Power for Lift

$$P_2 = \frac{H * Q_m}{367} = \frac{17.9 * 1000}{367} = 48.8 \text{ kW}$$

Power at drive pulley

$$P_T = P_1 + P_2 = 77 \text{ kW}$$

Required motor power

$$P_M = \frac{77}{0.9} = 85.6 \text{ kW}$$

Installed motor power

P_N = 110 kW (selected)

Belt breaking strength

$$k = \frac{c_R * P_T}{c_v * v} = \frac{15 * 77}{0.75 * 1.68} = 917 \text{ N/mm}$$

Friction factor cR (see table Page 11.9)
Breaking strength loss c_v (see table Page 11.9)
μ used = 0.3
Belt with 4 plies

Belt selection

EP 1000/4 8 + 4 mm covers

Belt weight 25.4 kg
Carcase thickness 5.8 mm

Drive Power and Belt Tension

Peripheral Force

$$F_U = C * f * L * g [m'_R + (2 * m'_G + m'_L) * \cos \delta] + H * g * m'_L$$

Factor C = 1.37
Friction factor f = 0.02

Load due to idlers $m'_R = \frac{22.3}{1.2} + \frac{19.3}{2} = 18.6 + 9.6 = 28.2$ kg/m

Load due to belt $m'_G = 25.4$ kg/m

Load due to material $m'_L = \frac{1000}{3.6 * 1.68} = 165.3$ kg/m

Incline $\delta = 4.8°$ $\cos 4.8° = 0.997$

Start-up factor $k_A = 1.5$ (hydraulic coupling)

Peripheral Force $F_U = 1.37 * 0.02 * 258 * 9.81 [28.2 + (2 * 25.4 + 165.3) * 0.997] + 17.9 * 9.81 * 165.3$

Steady state $F_U = 16897 + 29027 = 45924$ N

Non-steady state $F_A = 45924 * 1.5 = 68885$ N

Power at pulley $P_T = \frac{45923 * 1.68}{1000} = 77.2$ kW

drive degree of efficiency $\eta = 0.9$

Motor power $P_M = \frac{P_T}{0.9} = 85.7$ kW

Installed motor power $P_N = 110$ kW (selected)

Min. belt tension $T_{min} = \frac{(m'_L + m'_G) * g * l_o}{8 * 0.01} = 28061$ N

With belt sag 1%
Carrying side idle pitch $l_o = 1.2$m

Acceleration $a_A = \frac{68885 - 45924}{258 * (165.3 + 25.4 + 50.8)} = 0.368$ m/s^2

Start-up time $t_A = \frac{1.68}{0.368} = 4.56$ s

Start-up distance $s_A = \frac{1.68 * 4.56}{2} = 3.83$ m

Individual Resistances for Sequential Calculation

Main resistance	$F_H = 0.02 * 258 * 9.81 * [\,28.2 + (\,50.8 + 165.3\,) * 0.997\,] = 12333$ N
Secondary resistance	$F_N = (\,C - 1\,) * F_H = 12333 * 0.37 = 4563$ N
Frictional Resistances	
Carrying side	$F_o = 0.02 * 258 * 9.81 * [\,18.6 + (\,25.4 + 165.3\,) * 0.997\,] = 10566$ N
Return side	$F_u = 0.02 * 258 * 9.81 * (\,9.6 + 25.4\,) * 0.997 = 1768$ N
Slope Resistance	
Carrying side	$F_{sto} = 17.9 * 9.81 * (\,25.4 + 165.3\,) = 33486$ N
Return side	$F_{stu} = 17.9 * 9.81 * 25.4 = 4460$ N
Inertial Resistance	
Carrying side	$F_{ao} = 0.368 * 258 * (\,16.7 + 25.4 + 165.3\,) = 19725$ N
Return side	$F_{au} = 0.368 * 258 * (\,8.7 + 25.4\,) = 3237$ N

Sequential Calculation T_1 to T_4

Running (steady state)	$T_2 = F_U * c_2 = 45924 * 0.5 = 22961$ N
	$T_3 = 22962 + 1768 - 4460 = 20269$ N
	$T_4 = T_3 = 20269$ N
	$T_1 = 20269 + 4563 + 10566 + 33486 = 68884$ N
	Sum = 132383 N
Control T_1	$T_1 = F_U * c_1 = 45924 * 1.5 = 68884$ N
Start-up (non-steady state)	$T_{A2} = F_A * c_{2A} = 68886 * 0.3836 = 26424$ N
	$T_{A3} = 26424 + 1768 - 4460 + 3237 = 26969$ N
	$T_{A4} = T_{A3} = 26969$ N
	$T_{A1} = 26969 + 4563 + 10566 + 33486 + 19725 = 95309$ N
	Sum = 175671 N
Control T_{A1}	$T_{A1} = F_A * c_{1A} = 68886 * 1.3836 = 95309$ N

Case 1) Tension Weight at Discharge Point (close to T_2)

1st Adjustment

$T_{A2} = T_2$ (constant tension)
$\Delta T = T_{A2} - T_2 = 26424 - 22961 = 3463$ N
With the value ΔT belt tensions T_1 to T_4 are increased

$T_1 = 72347$ N		$T_{A1} = 95309$ N
$T_2 = 26424$ N	⟷	$T_{A2} = 26424$ N
$T_3 = 23732$ N		$T_{A3} = 26969$ N
$T_4 = 23732$ N		$T_{A4} = 26969$ N

2nd Adjustment

$T_{min} = 28061$ N est nécessaire pour la flèche de 1%, i.e. $T_4 \geq T_{min}$
$\Delta T = T_{min} - T_4 = 28061 - 23732 = 4328$ N
With the value ΔT all belt tensions T_1 to T_4 and T_{A1} to T_{A4} will be increased

Belt Tensions

$T_1 = 76675$ N		$T_{A1} = 99637$ N
$T_2 = 30752$ N	⟷	$T_{A2} = 30752$ N
$T_3 = 28061$ N		$T_{A3} = 31297$ N
$T_4 = 28061$ N		$T_{A4} = 31297$ N
$\Sigma T = 163549$ N		$\Sigma T_A = 192983$ N

The stipulations $T_2 = T_{A2}$ and $T_4 \geq T_{min}$ are fulfilled.

Case 2) - Tension Weight at Tail (close to T_4)

1st Adjustment

$T_{A3} = T_3$ (constant tension)
$\Delta T = T_{A3} - T_3 = 26969 - 20269 = 6700$ N

T_1 = 75584 N	T_{A1} = 95309 N
T_2 = 29661 N	T_{A2} = 26424 N
T_3 = 26969 N	T_{A3} = 26969 N
T_4 = 26969 N	T_{A4} = 26969 N

2nd Adjustment

$\Delta T = T_{min} - T_4 = 28061 - 20269 = 1092$ N

Belt Tensions

T_1 = 76675 N	T_{A1} = 96401 N
T_2 = 30752 N	T_{A2} = 27516 N
T_3 = 28061 N ⟷	T_{A3} = 28061 N
T_4 = 28061 N	T_{A4} = 28061 N
ΣT = 163549 N	ΣT_A = 180039 N

Stipulation

1. $T_4 = T_{A4}$ 2. $T_4 \geq T_{min}$

Case 3) Fixed Take-up Tension, Location no Influence

1st Adjustment

$\Sigma T = \Sigma T_A$
$\Delta T = (175671 - 132383) / 4 =$ 10822 N

T_1 = 79706 N	T_{A1} = 95309 N
T_2 = 33783 N	T_{A2} = 26424 N
T_3 = 31091 N	T_{A3} = 26969 N
T_4 = 31091 N	T_{A4} = 26969 N
Σ = 175671 N ⟷	ΣT_A = 175671 N

2nd Adjustment

$\Delta T = T_{min} - T_{A4} = 28061 - 26969 = 1092$ N

Belt Tensions

T_1 = 80797 N	T_{A1} = 96401 N
T_2 = 34874 N	T_{A2} = 27516 N
T_3 = 32183 N	T_{A3} = 28061 N
T_4 = 32183 N	T_{A4} = 28061 N
ΣT = 180037 N ⟷	ΣT_A = 180037 N

Stipulation

1. $\Sigma T = \Sigma T_A$ 2. T_4 or $T_{A4} \geq T_{min}$

Note

For Case 1 (tension weight at discharge) the belt tensions T_1 and T_2 would suffice after Adjustment 1. To transmit the peripheral force F_U friction cut off wise. The condition $T_1/T_2 \leq e^{\alpha\mu}$ would be fulfilled with $\mu = 0.3$, $\mu_A = 0.35$ and $\alpha = 210°$:

Running : $T_1/T_2 = 72347 / 26424 = 2.74 \leq 3$
Start-up : $T_{A1}/T_{A2} = 95309 / 26424 = 3.6 = e^{\alpha\mu}$

With this condition however the belt would not have the desired degree of sag of 1% at the tail end and fine grained material could escape under the skirtboards at the loading point. The belt would undulate excessively resulting in high wear and tear and extra energy consumption.

Sag without adjustment:

$$h_d = \frac{(165.3 + 25.4) * 9.81 * 1.2}{23732 * 8} = 0.0118$$

The degree of sag desired was 1%

Comparison of various Take-up Systems

Belt Safety

$$S = \frac{k_N * c_v * B}{T_X}$$

k_N	(N/mm)	Nominal belt strength
c_v	(-)	Loss of strength at the joint
B	(mm)	Belt width
T_X	(N)	Belt tension at point X

Working condition	Belt Safety S Case 1	Case 2	Case 3
steady state	11.73	11.73	11.13
non-steady state	9.03	9.33	9.33

Pre-tension Force (kg)

	Case 1	Case 2	Case 3
Pre-tension	$2 * T_{A2}$	$2 * T_{A4}$	$\Sigma T/4 * 2$
Tension force (kg)	6265	5720	9176

Take-up Travel

$$s_p = \frac{\Sigma T * L}{2 * B * k_D * k_N} \quad \text{(mm)}$$

ΣT in Case 1	
at rest	$\Sigma T = 4 * 30752 = 123008$ N
running	$\Sigma T = 163549$ N
at start-up	$\Sigma T = 192983$ N
elongation value	k_D (-) = 10.5

Working condition	Take-up Travel (mm) Case 1	Case 2	Case 3
at rest	1259	1149	1843
steady state	1674	1674	1843
non-steady state	1975	1843	1843

Evaluation of Belt Safety Factor, Pre-tension and Take-up Travel

a) For the belt and installation, Case 2 (take-up weight at tail) is the most favourable. The belt safety is almost the same in all three cases. The pre-tension is substantially less in case 2 and results in the lowest stresses for both the belt and the installation.

b) With the fixed take-up, the calculated travel must be taken into account otherwise a shortening of the belt may be necessary later on.

Result of the Computer Calculation

DUNLOP-ENERKA Comparison of Tension Systems

Safety Factors	Discharge	Tail	Fixed Take-up
Running	11.7	11.7	11.1
Start-up	9.0	9.3	9.3
Pre-tension			
Minimum kg	5387	5500	8955
Maximum kg	6268	5721	9175
Working elongation %	0.28	0.27	
Total elongation %	0.77	0.71	0.71

Pulley Diameter

$$\boxed{D_A = c_{Tr} * d} = 108 * 5.8 = 626 \text{ mm}$$

c_{Tr}	(-)	Parameter for the belts types. See Page 11.10
d	(mm)	Carcase thickness

Drive pulley	D_A = 630 mm (selected)
Bend pulley	D_B = 500 mm
Snub/deflection pulley	D_C = 315 mm

Pulley revolutions

$$\boxed{n_T = \frac{v * 60}{\pi * D_A}} = \frac{1.68 * 60}{3.14 * 0.63} = 51 \text{ R/}_{min}$$

D_A (m) Pulley diameter

Maximum torque

$$\boxed{M_A = \frac{F_{UA} * D_A}{2}} = \frac{68885 * 0.63}{2} = 21699 \text{ Nm}$$

Pulley loading
Drive pulley

$$\boxed{F_{AT} = \frac{(T_{A1} + T_{A2})}{9.81}} = \frac{99637 + 30752}{9.81} = 13291 \text{ kg}$$

Surface pressure

$$\boxed{p_T = \frac{(T_{A1} + T_{A2})}{D_A * B}} = \frac{130389}{63 * 120} = 17.3 \text{ N/cm}^2$$

D_A and B in cm

Troughing Transition

$$\boxed{L_M = x * s * \sin\lambda} = 8 * 367 * \sin 35° = 1684 \text{ mm}$$

x = 8	Factor for carcase
s = 367 mm	Side idler roller contact made by belt
λ = 35°	Troughing angle

Lift of pulley
for troughing 30°

$$\boxed{h = \frac{s^2}{B} * \sin\lambda} = \frac{367^2}{1200} * \sin 35° = 65 \text{ mm}$$

Reduced
transition length

$$\boxed{L_{red} = x * (s * \sin\lambda - h)}$$

$$L_{red} = 8 * (367 * \sin 35° - 65) = 1164 \text{ mm}$$

Convex Vertical Curve

$R_e = x * s * \sin\lambda$ $= 125 * 367 * \sin 35° = 26.3$ m

x = 125 (factor for carcase)

Concave Vertical Curve

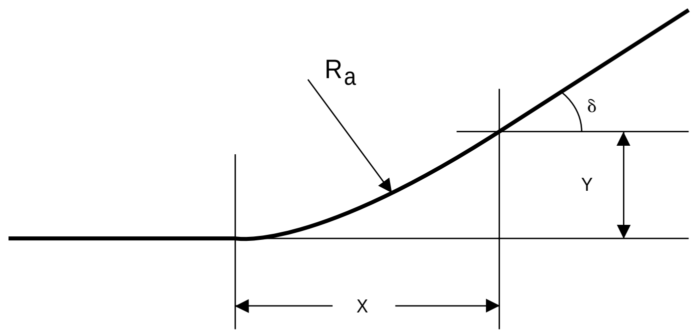
$$R_a = \frac{T_x}{m'_G * 9.81 * \cos\lambda} = \frac{29703}{25.4 * 9.81 * \cos 35°} = 120 \text{ m}$$

$T_x = T_4 + f * L_x * g * (m'_{Ro} + m'_G + m'_L) = 29703$ N Belt tension at start of curve
$T_4 = 28061$ N Belt tension when running
$L_x = 40$ m Horizontal distance before the curve

Co-ordinates of beginning and end of the curve

$X = R_a * \tan\delta = 120 * 0.0839 = 10.1$ m
$Y = 0.5 * R_a * \tan^2\delta = 0.4$ m

$$\sin\delta = \frac{17.9}{212} = 0.0844 \qquad \delta = 4.8°$$



Additional Tension Adjustment

In the regions where the belt goes from flat to troughed, trough vertical and horizontal curves or is turned over, additional stresses and strains occur at the edge zones or in the belt centre. This can result in a reduction in the permissible minimum safety at the belt edges or belt centre. The tension can become negative creating buckling at the centre or flapping at the edges. For additional information see Appendix 1.

Safety in the edge zone

The required values for the calculation for the belt safety running empty at head and tail were extracted from the computer calculation.

Elongation at T_1 (running)

$$\epsilon_0 = \frac{1}{10.5 * 11.7} = 0.00814$$

Curve length

$$\hat{\lambda} = \frac{3.14}{180} * 35° = 0.61056$$

Additional elongation

$$\epsilon_K = \frac{1200^2 * 0.6105^2}{1684^2} * \frac{367^2}{1200^2} * (0.5 - \frac{367}{3 * 1200}) = 0.00705$$

Total elongation

$$\epsilon_{ges} = 0.00814 + 0.00705 = 0.01519$$

$$S_{min} = \frac{1}{10.5 * 0.01519} = 6.27 \geq 4$$ no overstress

Elongation at Belt Centre Discharge Point

Min. safety at running empty

$S_{min} = 25.8$ ⟶ at discharge point (head)

Elongation at T_1 empty

$$\epsilon_0 = \frac{1}{10.5 * 25.8} = 0.003692$$

Additional elongation

$$\epsilon_M = \frac{1200^2 * 0.6105^2}{1684^2} * \frac{367^3}{3 * 1200^3} = 0.0018049$$

Total elongation

$$\epsilon_{min} = 0.003692 - 0.0018049 = 0.0018871$$

$\epsilon_{min} \geq 0$ no buckling

Elongation at Belt Centre at Tail

Min. safety at running empty

$S_{min} = 32.1$ (value from computer calculation)

Elongation at T_1 empty

$$\epsilon_0 = \frac{1}{10.5 * 32.1} = 0.0029669$$

Additional elongation

$\epsilon_M = 0.0018049$

Total elongation

$$\epsilon_{min} = 0.0029669 - 0.0018049 = 0.001162$$

$\epsilon_{min} \geq 0$ no buckling

Computer Calculation

DUNLOP-ENERKA	CONVEYOR BELT CALCULATION	DATE 06.08.93

Drive System	: Single Pulley Head Drive	Firm	: Musterman
Tension	: GTU at Head Drive	Project	: Example
Coupling	: Hydraulic Coupling	Position	: 12

INSTALLATION		**MATERIAL**		**CONVEYOR BELT**	
Capacity	1000 t/h	Material	Sand & Gravel	Belt Type	multiply
Conveying Length	258.0 m	Density	Ambiant	Belt Type	EP 1000/4
Lift	17.9 m	Lump Size (mm)	1.50 t/m3	Covers	8 + 4 mm
belt Width	1200 mm	Temperature	max. 50	Quality	RA
Belt Speed	1.68 m/s	Surcharge Angle	15°	Weight	25.4 kg/m

DRIVE		**PULLEYS**		**IDLER ROLLERS**	
Power Running Empty	7.0 kW	Drive Pulley	630 mm	Roller dia. Carrying	133 mm
Additional Power	0.0 kW	Tension Pulley	500 mm	Roller dia. Return	133 mm
Required Power	85.7 kW	Snub Pulley	315 mm	Roller Weight Carrying	22.3 kW
Installed Power	110.0 kW	Max. Torque	21699 Nm	Roller Weight Return	19.3 kg
Degree of Efficiency	0.90	Drive Pulley Rev	51.0 R/min	Roller Pitch Carrying	1.20 m
Angle of Wrap	210°	Load-Drive Pulley	13290 kg	Roller Pitch Return	2.00 m
Friction Coefficient	0.020	" Tension Pulley	6268 kg	Loading, Carrying	229 kg
Drive Factor	1.5	Friction Factor		Loading, Return	51 kg
Belt Sag	1.0 %	Drive Pulley	0.30	Rev. Carrying Roller	241.4 r.p.m
		Weight	0 kg	Idler Type 3 Port	35°

Safety Factor : Running SB = 11.7
: Start-up SA = 9.0

Acceleration Line : Loaded 4.56 S
: Empty 0.51

PRE-TENSION CRITERIA	**BELT SAG** Running	At Start-up	**SLIP SAFETY FACTOR** Running	At Start-up
Belt Tension T_1 (N)	76667	99629	72349	95311
Belt Tension T_2 (N)	30743	30743	26425	26425
Belt Tension T_3 (N)	28061	31297	23743	26979
Belt Tension T_4 (N)	28061	31297	23743	26979
Belt Tension Sum	163532	192967	146260	175695
Tension Weight	6268 kg		5387 kg	

Take-up Travel (absolute):	Movement Distance Weight:	Belt Elongation:
Weight at Rest ....1259 mm	At Rest .... 0 mm	Under Pre-tension .... 0.49 %
Full Load Running ....1674 mm	Full Load Running ...... 415 mm	Working Elongation ..... 0.28 %
Full Load Start-up ....1976 mm	Full Load Start-up .... 717 mm	Total Elongation........... 0.77 %
Elongation Value ....10.5 (-)		

Measurement of Installation Elements:

Troughing Transition	1686 mm	Convex Vertical Curve	26 m
Pulley Lift	65 mm	Concave Vertical Curve	119 m
Reduced Troughing Transition	1170 mm	Measurement xa horizontal	10.0 m
		Measurement ya vertical	0.4 m

For calculation purposes missing data assumed.
Data with Value 0 were not calculated i.e., not fed in.
We cannot take responsibility for the calculation.

Adjustments of Transition Lengths

Adjustment of Additional Tension **Transition Lengths**

Drive : Single Pulley Head Drive Weight at Discharge
Belt Width : 1200 mm 3 roll 35° KD = 10.5 Belt EP 1000/4

T_1 Full	=	76667 N	Discharge Full SB	=	11.7	Discharge L normal	=	1686 mm
T_1 Empty	=	34939 N	Discharge Empty SB	=	25.8	Tail L normal	=	1686 mm
T_4 Full	=	28061 N	Heck Full SB	=	32.1	Discharge L reduced	=	1170 mm
T_4 Empty	=	28064 N	Heck Empty SB	=	32.1	Tail L reduced	=	1170 mm
			Dyn. Tensile Strength Smin	=	4.0	Pulley Lift	=	65 mm

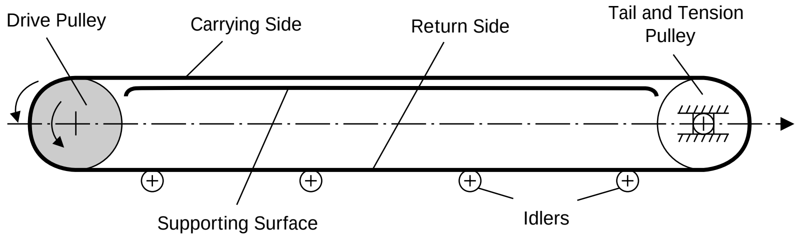
Troughing Transition	Belt Edge		Belt Centre	Criteria
Discharge L normal	0.01516	6.28	0.00189	No buckling
Tail L normal	0.01001	9.51	0.00116	No buckling
Discharge L reduced	0.01586	6.01	0.00146	No buckling
Tail L reduced	0.01071	8.89	0.00074	No buckling

Continue Calculation (1)	Vertical Curve (3)	Belt Turnover (5)
Change Transition Length (2)	Horizontal Curve (4)	

DUNLOP Conveyor Belts at Crusher Plant.

Slider Belts

The conveying of unit loads is generally done by slider belts with conveying lengths up to about 100m. With heavy loads or long length conveying, the belt is tracked over flat or slightly troughed idlers.



Support Surface

The **supporting or slider surface** can be made from various materials with different friction values e.g. steel, wood or synthetic.

To avoid a `suction' effect with the belt, transverse slots can be made at intervals in the supporting surface and perhaps relieving rollers fitted.

Belt Widths

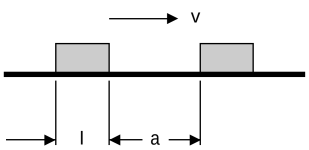
The belt width should be somewhat greater than the longest edge length of a single piece except with pieces loaded in the direction of belt travel.

Belt Width

$$B \approx \text{max. edge length} + 100 \quad (\text{mm})$$

Speed

The speed of the slider belt must be in keeping with the type of unit load. **As a rule V is from 0.2 to 1.5 m/s.**



$$v = \frac{z * (l + a)}{3600} \quad (\text{m/s})$$ With a load of z pieces per hour

$$v = \frac{l + a}{t} \quad (\text{m/s})$$ With a load at time interval t

z	(pieces/hour)	Number of pieces per hour
l	(m)	Length of piece
a	(m)	Spacing of pieces
t	(s)	Loading interval

Conveying Capacity

The conveying capacity can be evaluated from the **given data.**

$$Q_m = 3.6 * v * \frac{m}{l + a} \quad (t/h)$$

$$Q_m = 3.6 * v * \frac{z_m * m}{L} \quad (t/h)$$

$$Q_m = \frac{z * m}{1000} \quad (t/h)$$

v	(m/s)	Belt speed
m	(kg)	Mass of piece
z_m	(St/m)	Number each metre $z_m = L / (l + a)$
z	(St/h)	Pieces per hour
L	(m)	Conveying Length

For **bulk loads** see appropriate tables.

Resistance to Motion

The total resistance to motion and therefore the pulley peripheral force F_U is calculated from the sum of the individual resistances. With slider belts the sum of the **main resistances** from friction on the carrying and return side is relatively high compared with belts supported by idlers.

Peripheral Force

$$F_U = C_g * (F_o + F_u) + F_{st} + F_S\dagger \quad (N)$$

Secondary Resistances

The secondary resistances are independent of the installation length and only occur at certain points. Only with relatively short installations is the value greater that 1.

The secondary resistances are estimated using factor C_g. The value is dependent on the average surface loading. The effect of the secondary resistances is considerably smaller than obtained from the friction value μ_g between belt and supporting surface.

Factor C_g

Conveying Length L (m)	Average Surface Loading p_m (kg/m^2)	
	5 -10	up to 300
2.5	1.8	1.04
5	1.4	1.02
10	1.2	1.01
25	1.09	≈ 1
50	1.05	≈ 1
>100	≈ 1	≈ 1

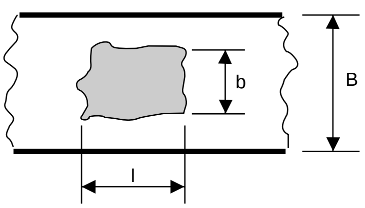
Average Surface Loading

$$p_m = \frac{m'_G}{B} + \frac{m'_L}{b} \quad (kg/m^2)$$

For unit loads

$$p_m = \frac{m'_G + m'_L}{B} \quad (kg/m^2)$$

For bulk loads



m'_G	(kg/m)	Load due to belt
m'_L	(kg/m)	Load due to material (see page 12.6)
B	(m)	Belt width
b	(m)	Load width or width of piece
l	(m)	Length of piece

Frictional Resistances

Frictional resistance occurs on the carrying and return side either sliding or rolling resistance depending on how the belt is supported.

Carrying Side

Sliding

$$F_o = \mu_g * L * g * (m'_L + m'_G) \quad (N)$$

On rollers

$$F_o = 2 * z_R + 0.02 * L * g * (m'_L + m'_G) \quad (N)$$

Return Side

Sliding

$$F_u = \mu_g * L * g * m'_G \quad (N)$$

On rollers

$$F_u = 2 * z_R + 0.02 * L * g * m'_G \quad (N)$$

m'_L	(kg/m)	Load due to material $m'_L = Q_m / 3.6 / v$ (kg/m)
m'_G	(kg/m)	Load due to belt
z_R	(pieces)	Number of rollers
μ_g	(-)	Friction value between belt and supporting surfaces (see table)

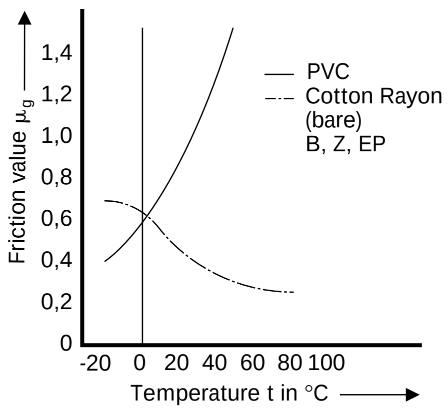
Friction Value μ_g

The average friction value between belt and supporting surface was determined thus:

- Surface load $p = 20 - 500$ N/mm²
- Speed $v = 0.2 - 0.8$ m/s

	Slider Bed Surface - Upper Surface Temperature (°C)					
	Steel Sheet Polished				Synthetic	Hard Wood
	-20°	0°	+18°	+40°	+18°	+18°
Cotton (bare)	0.75	0.70	0.45	0.35	0.40	0.35
EP (bare, impregnated)	0.30	0.30	0.30	0.25	0.30	0.25
Rubber covered			0.90			
Cotton (PVC impregnated)	0.60	0.55	0.50	0.40	0.45	0.35
PVC (film)	0.60	0.60	0.60	0.55	0.60	0.50
PVC (smooth)	0.55	0.70	1.00	1.70	1.30	0.95
PVC (textured)	0.50	0.65	0.80	1.30	1.10	0.75

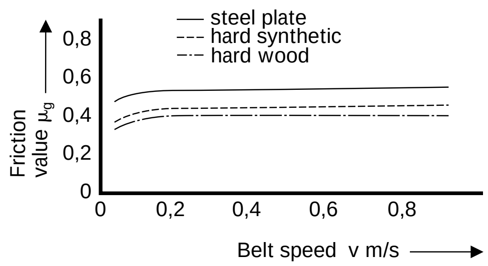
Temperature

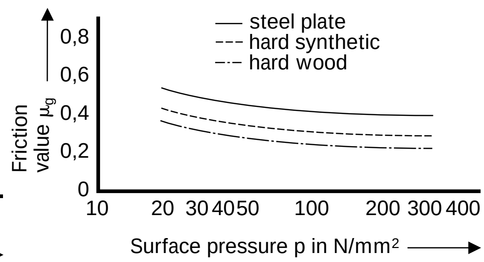


The friction value μ_g is mainly dependent upon the combination of the belt under side surface and the temperature, the speed and the surface loading.

With increasing temperature the friction value μ_g increases with PVC running side surfaces and decreases with fabric surfaces.

Speed Surface Loading





Independent of the characteristics of the slider surface, the friction value μg increases with increasing speed and decreases with increasing surface loading. When selecting the friction value these parameters can be taken into account according to the given working conditions.

Slope Resistance F_{st}

$$F_{st} = H * g * m'_L \qquad (N)$$

H (m) elevation (lift or fall)

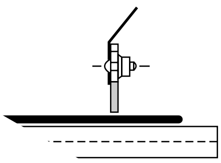
Special Resistance F_S

The special resistances have to be calculated separately according to the conditions existing.

- Resistance F_{A1} Skirtboard seals
- Resistance F_{Aw} Tripper (throw-off carriage)
- Resistance F_{Ab} Scraper
- Resistance F_{Me} Knife edges
- Resistance F_{Sp} Unit load barrier
- Resistance F_{La} Load deflector

$$F_S = F_{Al} + F_{Aw} + F_{Ab} + F_{Me} + F_{Sp} + F_{La}$$

Resistance F_{Al}

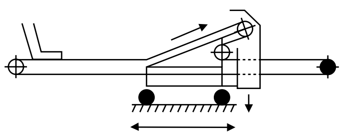


Resistance from **side skirtboards** used to channel the load stream (value).

$$F_{Al} = 160 * l_f \qquad (N)$$

l_f (m) Length of skirtboards per metre of conveying length

Resistance F_{Aw}



Resistance due to **trippers** (values)

$$F_{Aw} = z * k \qquad (N)$$

z (-) number of trippers
k (-) factor

Belt Width (mm)	Factor k for Tripper fixed	movable
300 - 500	1000	1100
650	1500	1700
800	2000	2200
1000	3500	3600
1200	4400	5100
>1200	4700	5400

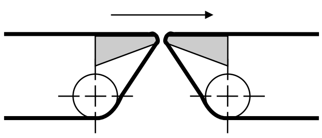
Resistance F_{Ab}

Resistance due to **simple scraper** for cleaning belts upper surface.

$$F_{Ab} = z * 800 * B \qquad (N)$$

z (-) Number of scrapers
B (m) Belt width

Resistance F_{Me}



Resistances due to **knife edges** (value). Knife edges are used to change the direction of very thin slider belts where particularly small transition gaps are necessary. At such knife edges relatively high temperatures may develop, up to approx. 150°C.

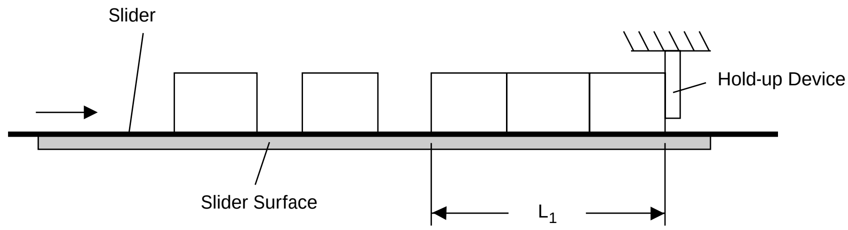
$$F_{Me} = z * 0.1 * B \qquad (N)$$

z (-) Number of knife edges
B (m) Belt width

Resistance F_{Sp}

Resistance due to **unit load blockage.**

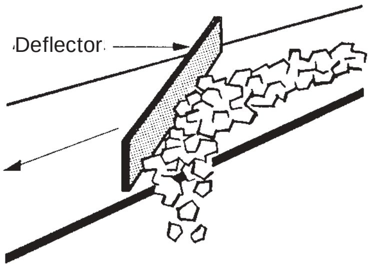
When conveying unit loads temporary hold-ups may occur due to congestion, that is, the belt slides underneath the loaded units resulting in a frictional resistance. This friction resistance can be very high in comparison to the frictional resistance on the belts running side.



$$F_{Sp} = \mu_l * g * L_1 * m'_L * \cos\delta \; - \; m'_L * H_1 \dagger \qquad (N)$$

L_1	(m)	Hold-up length
δ	(°)	Gradient of installation
μ_l	(-)	Friction factor belt/load This friction value can differ widely and depending on the load can be greater than the friction value μ_g between belt and slider surface.
H_1	(m)	Lift (or fall)
m'_L	(kg/m)	Load due to material

Resistance F_{La}



Resistance due to **sideways transfer** of load when conveying horizontally (load deflector). For bulk and unit loads the following values can be used.

Belt Width (mm)	≤ 500	650-800	1000-2000
F_{La} (N)	800	1500	3000-3500

For light bulk and unit loads these values may be reduced. With sideways transfer of the load, the belt is deflected in the opposite direction. Appropriate tracking devices need to be built in.

Drive Power

$$P_T = \frac{F_U * v}{1000} \quad (kW)$$

Motor Power

$$P_M = \frac{P_T}{\eta} \quad (kW)$$

Installed Power

$$P_N \quad (kW)$$

The motor power PN is selected from the standard list with the appropriate rounding up or from existing stock.

Peripheral Force F_A

The drive force of a slider belt installation is as a rule over-dimensioned because of the lower power requirements. Mostly squirrel cage motors with direct on-line starting are used. The start-up torque is approx. 2 - 2.6 times higher than the nominal torque.

$$F_A = 1.5 * F_U * \frac{P_N}{P_M} \quad (N)$$

Note

If $F_A \geq 2.6 * F_U$, then calculate thus: $F_A = 2.6 * F_U$ (N)

This prevents excessive over-dimensioning of the motor and thus the use of an overstrength belt.

Belt Tensions

The belt tensions are calculated from the peripheral forces F_U et F_A.

$$T_1 = F_U * c_1 \quad (N)$$
$$T_{1A} = F_A * c_{1A} \quad (N)$$

In selecting the belt, the highest value is the deciding factor.

Friction Value μ

The friction value μ depends on the **running side** of the belt and **the surface of the drive pulley.**

Running Side of belt	Drive Pulley	Condition	Friction value μ
Rubbered	Lagged	dry damp	0.30 0.25
	Bare	dry damp	0.22 0.20
Fabric	Lagged	dry damp	0.25 0.22
	Bare	dry damp	0.15 0.15

Drive Factor c_1

For factors c_1 and c_{1A}: see Appendix F.1.

At start-up the following may be used: $\mu_A = \mu + 0.05$

Belt Types

In selecting the belt type for a slider application, the tensile strength is largely of secondary consideration because of the low stresses. Nearly always the belts are over dimensioned. **Transverse stability** and the **location** of the belt are particular considerations.

As a rule a check on the factor of safety of the selected belt is sufficient.

Safety Factor
$$S = \frac{k_N * B * c_V}{T_{max}} \geq 10\text{-}12$$

Belt type
$$k_N = \frac{T_{max} * S}{B * c_V} \quad \text{N/mm}$$

K_N	(N/mm)	Nominal breaking strength
B	(mm)	Belt width
C_V	(-)	Factor for the loss of breaking strength at the joint. See Page 11.9.
T_{max}	(N)	Maximum belt tension

$$S = (6 - 8) * \frac{P_N}{P_M}$$

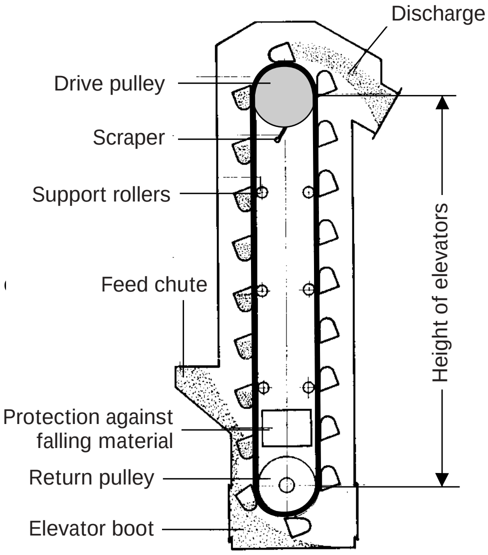
Pulley Diameter

Often with slider belt installations, extremely small pulleys are used, for example, to enable the bridging of gaps between two installations.
The advice of the belt manufacturer should be sought, particularly about problems of making the belt endless.

Rufftop Belt for Sack Conveying

Bucket Elevators

With bucket elevators different bulk loads such as sand, gravel, chippings, coal, cement, grain, flour, fertilizer etc. can be conveyed either vertically or sloping upwards. Belts with buckets attached are used as the carrying medium. With bucket elevators, great heights for example, over 100m can be surmounted with high strength belts. They run with low noise levels and low vibrations at a relatively high speed. They require only a small ground area and are preferred in buildings, silos and warehouses.



Load volume
$$Q_V = 3.6 * v * \varphi * \frac{V_B}{a} \quad (\,m^3/h\,)$$

Load mass
$$Q_m = Q_V * \rho \quad (\,t/h\,)$$

Bucket capacity
$$V_B = \frac{Q_V * a}{3.6 * v * \varphi} \quad (\,litres\,)$$

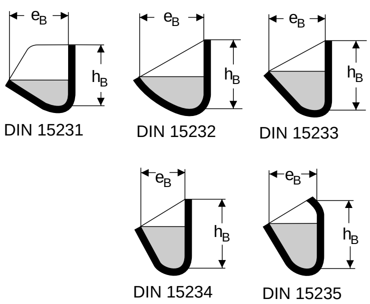
Weight of material per bucket
$$V_m = \frac{Q_m * a}{3.6 * v * \varphi} \quad (\,kg\,)$$

V_B	(l)	Nominal capacity of one bucket
φ	(-)	Degree of filling
v	(m/s)	Speed
a	(m)	Bucket Pitch
ρ	(t/m^3)	Bulk density of load

Nominal Capacity V_B (l) — The nominal capacity V_B of the bucket is determined from the geometric dimensions and a horizontal surface filling level (water filling).

Bucket Types

The type of bucket is determined in the main by the material and the method of discharge either by gravitational or centrifugal emptying.



e_B (mm) Bucket Projection
h_B (mm) Bucket Height

DIN 15231 Flat bucket for light loads such as flour, semolina, grain.
DIN 15232 Flat rounded bucket for light granulated loads such as grain.
DIN 15233 Medium deep bucket for sticky loads such as cane sugar.
DIN 15234 Deep buckets with a flat back wall for heavy pulverized loads or coarse ground loads such as sand, cement, coal.
DIN 15235 Deep buckets with curved back wall for light flowing or rolling loads such as fly ash and potatoes.

For dimensions, capacities and weights, see Page 15.12.

For **high speeds** (V = 1.05 to 4.2 m/s) according to the type of material as a rule the flat, flat h_B rounded or medium deep buckets are used (centrifugal discharge).

For **low speeds** (v = 0.42 to 1.05 m/s) the deep closed buckets are used (gravitational discharge). See also Page 15.3.

Degree of Filling

The measure of the effective utilization of the bucket capacity, is **the degree of filling** φ.

$$\varphi = \frac{\text{eff. bucket filling}}{\text{nominal capacity}}$$

Major influences are:
- Method of loading
- Speed
- Bucket spacing
- Bucket shape

Type of Material	Degree of Filling φ
Potatoes, Turnips, Onions	0.6
Grain, Barley, Rye	0.75
Earth, Sand - damp	0.5
dry	0.7
Sugar beet	0.5
Sliced beet	0.65
Fertilizer	0.75
Coal, Coke - large	0.5
fine	0.7
Gravel, Stone - large	0.5
fine	0.7
Sugar, Salt	0.75
Moulding sand - dry	0.6
Soya beans, Pulses	0.7
Cement	0.75

Speed

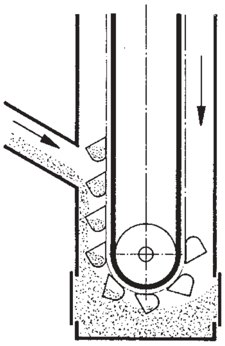
The speed depends upon the function of the bucket elevator, the type of loading and discharge.

The following speeds are recommended:

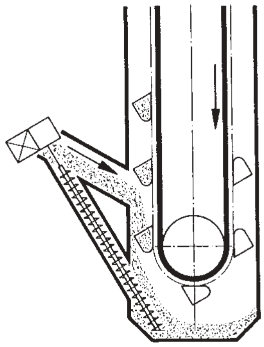
Type of Bucket Elevator	V (m/s)
Loading direct or by scoop action, gravitational discharge, slow moving for heavy materials such as ballast and earth.	Up to 1 m/s
Loading direct or by scoop action, Centrifugal discharge for normal material such as sand and fertilizer.	1 - 2 m/s
Loading direct or by scoop action, As a rule, centrifugal discharge, very fast running, only with free flowing or light easily scooped material such as grain	2 - 4 m/s and more

Loading

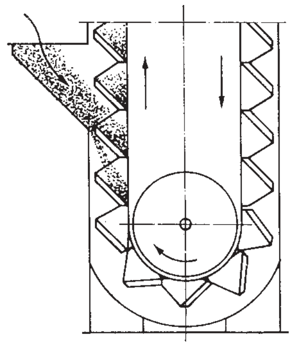
The loading i.e., the filling of the bucket can be done either directly or by dredging action. Scoop loading is only possible with free flowing materials and is to be recommended.
The bucket attachments and bucket brim are subject to arduous duty. With direct loading the base area (boot) has to be kept clean. Danger of buckets being torn off.



Scoop Loading

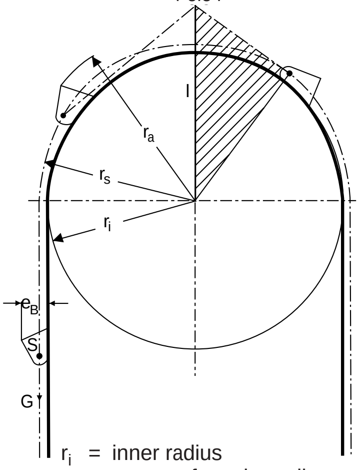


Direct Loading



Direct Loading
close pitch buckets
(continuous buckets)

Discharging



r_i = inner radius
r_s = centre of gravity radius
r_a = outer radius
l = pole distance

The discharging of bucket contents can be done by gravitational or centrifugal means. It depends on the speed and the pulley diameter. When running around the pulley, the resulting force gravitational and centrifugal alter in size and direction but Pole P stays the same for each bucket position.

$$l = \frac{g}{4 * \pi^2 * n^2} = \frac{895}{n^2} \quad (\text{m})$$

$$n = \frac{v * 60}{\pi * D} \quad (\text{R.P.M.})$$

$l < r_i$ Centrifugal Discharge.
The pole lies within the disc. Bucket with large opening is necessary.
The centrifugal discharge only applies when speeds are > 1.2 m/s.

$l > r_a$ Gravitational Discharge.
The pole lies outside the bucket brim. The material slides down over the inner wall of the bucket.

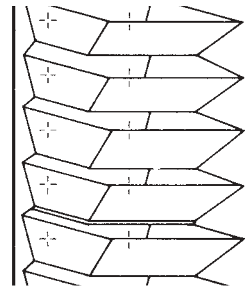
Centrifugal Discharge
With high speeds and light flowing loads or coarse granulated, hard materials.

Gravitational Discharge
With low speeds, heavy loads with large lumps or light dusty materials

Pulley Diameter (mm)	Speed High n (R.P.M.)	Speed High v (m/s)	Speed Low n (R.P.M.)	Speed Low v (m/s)
300	54	0.84	27	0.42
400	50	1.05	25	0.52
500	50	1.31	25	0.66
630	51	1.68	25	0.84
800	50	2.09	25	1.05
1000	50	2.62	25	1.31
1200	53	3.35		
1400	57	4.19		
1600	50	4.19		

Bucket Pitch

The bucket pitch depends on the speed, the bucket geometry, the type of loading and discharge. With **gravitational discharge,** the bucket pitch can be small. The material runs over the back of the preceding bucket.



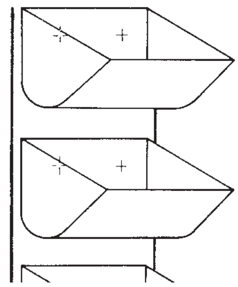
Bucket Pitch
Gravitational Discharge

$$a \approx \text{ ca. bucket height } h_B \quad (\text{mm})$$

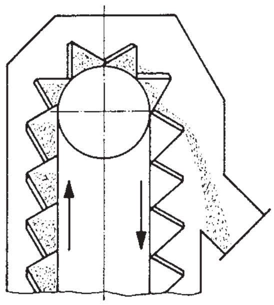
With the **centrifugal discharge** or gravitational discharge with staggered rows of buckets, the pitch has to be greater.

$$a = \frac{0.5 * \pi * D}{2 \text{ to } 4} \quad (\text{m})$$

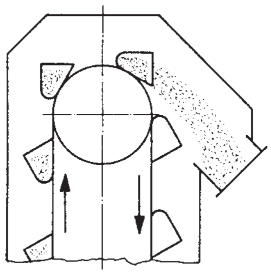
2 to 4 buckets are positioned on half of the pulley circumference.



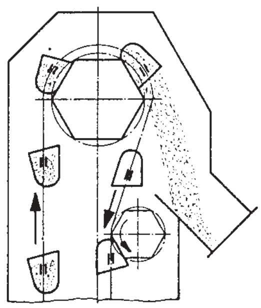
Bucket Pitch
Centrifugal Discharge



Gravitational Discharge



Centrifugal Discharge
v > 1.2 m/s



Centrifugal Discharge with staggered rows of buckets

Belt Width

The belt width is determined by the bucket width.

$$B \approx b_B + (30 \text{ to } 100) \quad (\text{mm})$$

Usual belt widths B (mm)						
150	200	250	300	400	500	550
650	800	1000	1250	1500	1600	

Peripheral Force F_U

The peripheral force F_U is determined from the sum of resistances to motion.

$$F_U = F_H + F_B + F_N \quad (\text{N})$$

Main Resistances F_H

The main resistance is derived from the capacity and the height.

$$F_H = \frac{Q_m * g * H}{3.6 * v} \quad (\text{N})$$

Q_m	(t/h)	Capacity
H	(m)	Height of Elevation
v	(m/s)	Speed
g	(m/s^2)	Acceleration due to gravity (9.81)

Loading Resistance F_B

The method of bucket loading and the force required to accelerate the load to conveying speed determines the loading resistance. F_B is dependent upon the speed. The loading resistance can be accounted for with sufficient accuracy using an additional height factor.

$$F_B = \frac{Q_m * g * H_0}{3.6 * v} \quad (N)$$

The following additional height factor values can be taken according to material and speed.

Type of material	ρ (t/m³)	H_0 (m)
Dry and powdery flour, rice, grain, cement	< 1	4 * v + 1.5
Fine grained sand, salt, sugar	1 - 1.5	4 * v + 4
Coarse grained up to approx. 50 mm gravel, coal, limestone	1.5 - 1.8	6 * v + 4
Rough of sticky Clay, earth, broken stone	> 1.8	6 * v + 6

Secondary Resistance F_N

The secondary resistance F_N takes into account the frictional forces, flex resistances of the belt, pulley bearing resistance and acceleration to the circumferential speed of the drive pulley. F_N is very small by comparison to the other resistances and is adequately covered by the factor c_N as part of the total resistances.

$$F_N = (c_N - 1) * (F_H + F_B) \quad (N)$$

$c_N \approx 1.1$ for bucket elevators

Peripheral Force F_U

$$F_U = c_N * \frac{Q_m * g * (H + H_0)}{3.6 * v} \quad (N)$$

Q_m	(t/h)	Capacity
H	(m)	Elevating height
H_0	(m)	Additional height (see table)
c_N	(-)	Factor for secondary resistances
v	(m/s)	Speed

The determination of F_U including consideration of individual resistances: see Page 15.10.

$$P_T = \frac{F_U * v}{1000} \quad (kW)$$

Drive Power at Pulley

$$P_M = \frac{P_T}{\eta} \quad (kW)$$

Motor Power

η (-) Degree of efficiency (in general 0.5 - 0.95)

Installed Power

P_N from standard range.

Start-up Factor k_A

$$k_A = k * \frac{P_N}{P_M} \quad (-)$$

Values for k

Type of drive	k
Squirrel cage, full voltage starting	2.2
Drive with fluid coupling	1.4 - 1.6
Slip ring motor	1.25

The couplings need to be adjusted to the nominal values.

Determination of Belt Type

From the preceding calculations, the bucket capacity, the bucket width and thus the belt width can be determined. For further calculations especially that for belt safety factor, the belt type has to be assessed as well as the weight of the take-up (tension) pulley.

Nominal Belt Strength

$$k_N = \frac{T_1 * S}{B} \quad (\text{N/mm})$$

S (-) Safety Factor
S = 10 up to 60° C
S = 12 up to 80° C
S = 15 up to 150° C
B (mm) Belt width

Belt Type Assessment

For the assessment of a belt type, initially the belt stress T_1 can be approximately estimated.

Tension T_1	$T_1 = F_U + F_{St} + T_v + T_T$	(N)
Peripheral Force	$F_U = 3 * \frac{Q_m * (H + H_0)}{v}$	(N)
Slope Resistance	$F_{St} = H * 9.81 * (m'_B + m'_G)$	(N)
Pre-Tension	$T_v = c_2 * k_A * F_U - F_{St} - T_T$	(N)
Weight of Take-up Pulley	$T_T = G_T * 9.81 / 2$	(N)

m'_B (kg/m) Weight of bucket and fastenings
m'_G (kg/m) Belt weight estimated depending on bulk density

Bulk Density ρ (t/m³)	Weight of Belt m'_G (kg/m)
≤ 1	8.5 * B
1 - 1.8	11.5 * B
> 1.8	15 * B

k_A (N) Start-up factor estimated
P ≤ 30 kW direct on line — k_A = 1.8 - 2.2
P > 30 kW coupled — $k_A = k * P_N / P_M$ (max 2.5)
depending on coupling — k = 1.2 - 1.6
c_2 (-) Drive factor, see Appendix — $\mu_A = \mu + 0.05$
α = 180°
G_T (kg) Weight of take-up pulley can be ascertained from the table or obtained from exact data if available

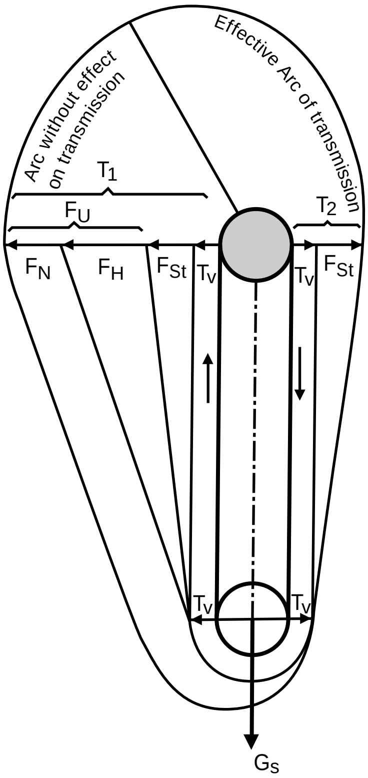
If $T_v \leq 0$ then T_v stays at 0, an additional pretension is not necessary.

Weight G_T (kg) of Take-up Pulley

Pulley Width (mm)	Pulley Diameter (mm) 315	400	500	630	800	1000	1250
125	16	20	27	35	45	70	110
160	20	25	35	45	60	80	135
250	25	30	45	55	80	115	175
350	40	45	70	90	125	180	270
450	42	60	75	115	160	230	310
550	50	70	80	130	190	300	360
650	55	75	95	150	210	350	420
850	65	90	115	180	250	450	550
1000	80	100	140	200	300	540	680
1250	100	115	160	230	350	670	840
1400	110	125	175	260	370	750	959

Once the belt type has been established, a back calculation can be done and safety factors catered for.

Belt Tensions T_1 and T_2



Power Transmission

After the belt nominal tension has been determined, a belt type can be selected and the factor of safety checked.

$T_1 = F_U + F_{St} + T_v + T_T$	(N)
$T_{A1} = F_A + F_{St} + T_v + T_T$	(N)
$T_2 = T_{A2} = F_{St} + T_v + T_T$	(N)

The necessary pre-tension for the power transmission is produced by the belt weight, the bucket weight (empty), the weight of the take-up pulley and possibly additional tension by screw take-up or take-up weight.

Additional pre-tension

$$T_v = c_2 * F_U - F_{St} - \frac{G_T * 9.81}{2} \quad (N)$$

Take-up weight

$$G_S = \frac{2 * T_v}{9.81} \quad (N)$$

If T_v is >0 then this is additional pre-tension.

For a frictional cut off of power transmission the following criteria according to the Eytelwein formula have to be fulfilled.

$$\frac{T_1}{T_2} \leq e^{\mu\alpha}$$

$$\frac{T_{A1}}{T_2} \leq e^{\mu_A \alpha}$$

μ	(-)	Friction factor drive pulley
μ_A	(-)	Friction factor at start-up
		$\mu_A = \mu + 0.05$

Safety Factor

After the calculation T_1 and T_{A1}, the safety factor whilst running and at start-up can be checked.

$$S_B \geq k_N * \frac{B}{T_1}$$

Safety running state

$$S_A \geq k_N * \frac{B}{T_{A1}}$$

Safety at start-up

Values for S_B		Working Temperature (°C) 60°	80°	120°	140°
Bucket Vulcanised to belt	Textile	8	10		
	Steel cord	8	8		
Hole punched according to DIN	Textile	10	12	14	15
	Steel cord	9 to 10			

The **hole punching** can be estimated depending on the choice of S or by subtracting the hole cross section from the belt width B.

Values for $e^{\mu\alpha}$ if α = 180°

μ	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
$e^{\mu\alpha}$	1.37	1.60	1.87	2.19	2.56	4.00	3.52	4.11

Friction Value μ

Pulley Surface	Cage Pulley		Drive Pulley Bare		Drive Pulley Lagged	
	B	A	B	A	B	A
Wet	-	-	0.10	0.15	0.25	0.35
Damp	0.10	0.15	0.15	0.25	0.30	0.40
Dry	0.15	0.25	0.20	0.30	0.35	0.45

B = Running
A = Start-up

Belt Tension T_1

The calculation thus far is based upon the **work dependent belt tension.** Under certain circumstances it is the **load dependent tension** that determines the maximum tension T for the loaded state at rest:

The higher value of T_1 is the determining factor for belt safety and belt type.

$$T_1 = H * g * \left(\frac{Q_m}{3.6 * v} + m'_G + m'_B \right) + \frac{G_T * g}{2} + T_v \qquad (\text{N})$$

Values for Pulley Diameter

Number of plies	Belt Type SUPERFORT	Pulley Diameter (mm)
3	S 315/3 S 400/3 S 500/3 S 630/3	315 315 400 400
4	S 400/4 S 500/4 S 630/4 S 800/4 S 1000/4 S 1250/4	500 500 630 630 800 800

Number of plies	Belt Type SUPERFORT	Pulley Diameter (mm)
5	S 630/5 S 800/5 S 1000/5 S 1250/5 S 1600/5	630 800 1000 1000 1200
6	S 1000/6 S 1250/6 S 1600/6	1000 1200 1400

1) for $k_N < 60\%$, 1 diameter smaller
2) take-up pulley 1 diameter smaller

When installing the buckets a slightly convex arc should be formed. Bucket attachment using flat iron strips should be segmented. For spacing see Appendix.

Table for Number of Plies

Load Bulk Density ρ (t/m³) Lump size (mm)	Bucket width (mm) 100	125	160	200	250	315	400	500	630	800	1000	Ply type
$\rho \leq 1$ t/m³ light flowinge grain, fertilizer oil seed	3	4	4	4	4	4	4	4	5	5	5	EP 100 EP 125 EP 160
$\rho = 1 - 1.5$ t/m³ 0 - 30 mm 0 - 60 mm 0 - 100 mm	3	3 4	4 4	4 5 5	5 5 5	5 5 5	5 5 5	5 5 6	5 6 6	6 6 6	6 6 6	EP 160 EP 200
$\rho \geq 1.5$ t/m³ 0 - 30 mm 0 - 60 mm 0 - 80 mm				4	5 4 4	5 5 5	5 5 6	5 6 6	6 6 6	6 6 6	6 6 6	EP 200 EP 250

Depending on type of bucket attachment and the pull through strength of the bolts, additional numbers of plies may be required.

Peripheral Force F_U

The **peripheral force F_U** can also be derived from the **individual resistances.**

$$F_U = F_H + F_{Aw} + F_S + F_{BA} + F_{BU} \qquad (N)$$

Lift resistance F_U

$$F_H = \frac{H * Q_m * g}{3.6 * v} \qquad (N)$$

Loading resistance F_{Aw}

$$F_{Aw} = \frac{Q_m}{3.6} * (v_1 + v) * g \qquad (N)$$

Dredging resistance F_S

$$F_S = f_k * w_s * \frac{Q_m * g}{3.6} \qquad (N)$$

Bending Resistance F_{BA} at drive pulley

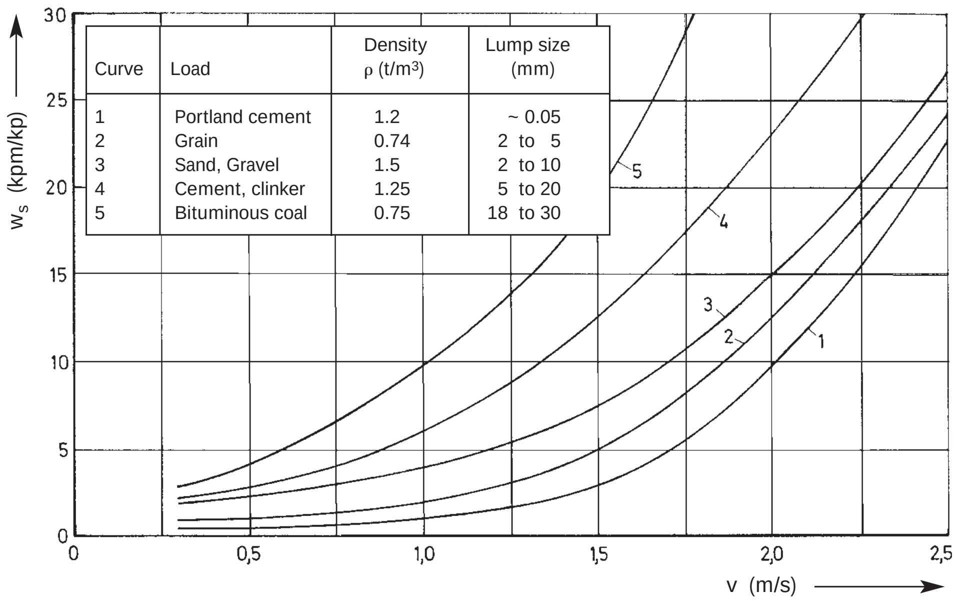
$$F_{BA} = 2 * x * \left(2 * y * B + \frac{T_1 + T_2}{g}\right) * \frac{d}{D} * g \qquad (N)$$

Bending Resistance F_{BU} at take-up pulley

$$F_{BU} = 4 * x * \left(y * B + \frac{T_v}{g}\right) * \frac{d}{D} * g \qquad (N)$$

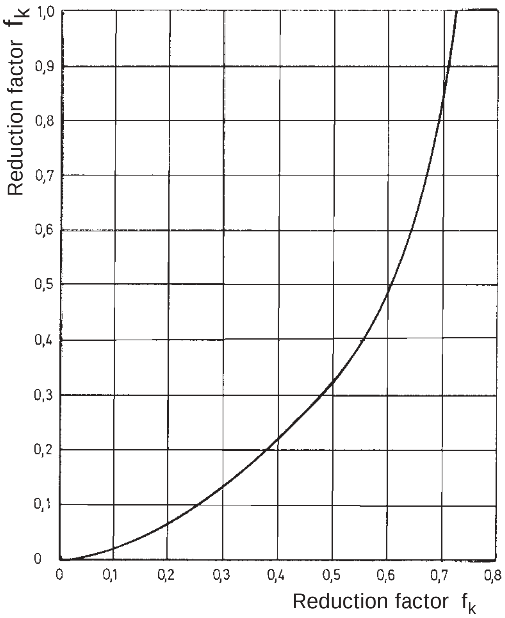
v_1	(m/s)	average loading speed		
v	(m/s)	belt speed		
f_k	(-)	reduction factor depending on relative bucket spacing (see table)		
w_s	(-)	specific scooping factor depending on v and type of material (see table)		
x	(-)	factor	x = 0.09	for textile belts
			x = 0.12	for ST belts
y	(-)	factor	y = 14	for textile belts
			y = 20	for ST belts
D	(cm)	pulley diameter		
d	(cm)	belt thickness		

Specific Scooping Factor W_S



Reduction Factor f_k

The reduction factor or scooping factor depends on the **relative bucket sequence t_F.**



$$t_F = 0.224 * \frac{a}{e_B * v} \quad (s)$$

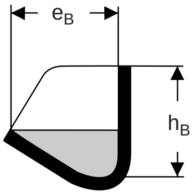
a	(mm)	bucket spacing
e_B	(mm)	bucket discharge
v	(m/s)	speed

Values for fill factor, bulk density and conveying speed.

Load	Bulk Density p (t/m³)	Fill Factor φ	Recommended maximum speed v (m/s)
Ash (slag)	0.9	0.7	2.5
Barley	0.7	0.8	2.0
Basalt	3.0	0.5	1.0
Basalt lava	2.8	0.5	1.2
Beets	0.65	0.5	2.0
Blast furnace slag - ground	0.7	0.8	2.7
Blast furnace slag	1.5	0.5	2.0
Briquette	1.0	0.5	1.8
Brown coal	0.7	0.5	1.9
Cement	1.2	0.8	2.5
Charcoal	0.3	0.6	2.5
Clay	2.0	0.7	1.8
Coal dust	0.7	0.7	2.7
Coke	0.4	0.6	2.5
Crushed coal	0.8	0.8	2.6
Earth	1.7	0.7	2.4
Fly ash	1.0	0.8	2.8
Granite	2.6	0.5	1.3
Gravel wet	2.0	0.7	2.5
Gravel dry	1.7	0.7	2.5
Gypsum	1.3	0.8	2.7
Lime	0.9	0.8	2.5
Limestone	2.6	0.5	1.2
Loam moist	2.0	0.4	1.8
Loam dry	1.6	0.7	2.0

Load	Bulk Density p (t/m³)	Fill Factor φ	Recommended maximum speed v (m/s)
Lump coal	0.9	0.5	1.5
Malt	0.55	0.7	3.0
Marble	2.7	0.5	1.2
Mortar Cement	2.0	0.7	2.0
Mortar Gypsum	1.2	0.7	2.0
Mortar Lime	1.7	0.7	2.0
Moulding sand	1.2	0.8	2.5
Oats	0.55	0.8	3.0
Potatoes	0.75	0.6	2.0
Pulses	0.85	0.7	2.9
Pumice	1.2	0.5	1.8
Pumice ground	0.7	0.7	2.7
Raw flour	1.0	0.8	3.5
Rye	0.7	0.8	3.0
Sand dry	1.6	0.7	2.5
Sand wet	2.1	0.4	2.5
Sawdust	0.25	0.8	3.0
Shell limestone	2.6	0.7	1.4
Slag coal	1.0	0.5	2.0
Slate	2.7	0.5	1.2
Sugar	0.7	0.8	2.7
Sugar beet chopped	0.3	0.7	3.0
Super phosphate	0.8	0.8	2.5
Volcanic limestone	2.0	0.5	1.6
Wheat	0.75	0.7	3.0

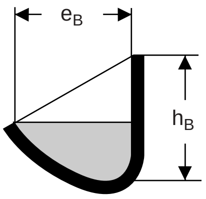
Bucket According to DIN



DIN 15231

Flat buckets to DIN 15231 for light materials such as flour, semolina.

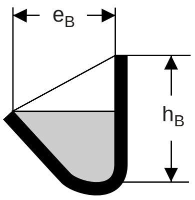
Width b_B (mm)	Projection e_B (mm)	Height h_B (mm)	Weight of Bucket in kg with steelsheet Thickness in mm						Bucket capacity v_B in litres
			0.88	1	1.5	2	3	4	
80	75	67	0.13	0.15					0.1
100	90	80	0.20	0.22	0.33				0.16
125	106	95	0.28	0.32	0.48	0.64			0.28
160	125	112		0.48	0.70	0.96			0.5
200	140	125		0.65	0.95	1.30	1.90		0.8
250	160	140		0.86	1.30	1.75	2.60		1.25
315	180	160			1.80	2.40	3.60	4.80	2.0
400	200	180				3.25	4.90	6.50	3.15
500	224	200					6.60	8.80	5.0--



DIN 15232

Flat rounded buckets to DIN 15232 for light granular materials such as grain.

Width b_B (mm)	Projection e_B (mm)	Height h_B (mm)	Weight of Bucket in kg with steelsheet Thickness in mm						Bucket capacity v_B in litres
			0.88	1	1.5	2	3	4	
80	75	80	0.14	0.16					0.17
100	90	95	0.21	0.24	0.36				0.3
125	106	112	0.30	0.34	0.51	0.68			0.53
160	125	132		0.50	0.75	1.00			0.9
200	140	150		0.68	1.02	1.40	2.10		1.4
250	160	170		0.94	1.40	1.90	2.80		2.24
315	180	190			1.95	2.60	3.85	5.20	3.55
400	200	212				3.55	5.30	7.10	5.6
500	224	236					7.20	9.60	9

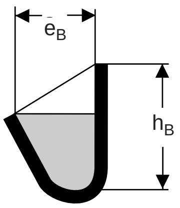


DIN 15233

Medium deep buckets to DIN 15233 for sticky material such as sugar cane.

Width b_B (mm)	Projection e_B (mm)	Height h_B (mm)	Weight of Bucket in kg with steelsheet Thickness in mm						Bucket capacity v_B in litres
			2	3	4	5	6	8	
160	140	160	1.23	1.86					0.95
200	160	180	1.66	2.57	3.46				1.5
250	180	200	2.24	3.36	4.48				2.36
315	200	224		4.56	6.08	7.85			3.75
400	224	250		6.06	8.15	10.3			6
500	250	280			11.5	14.4			9.5
630	280	315			16.1	20.2	24.3		15
800	315	355				27.5	33.3	44.3	23.6
1000	355	400				38.2	46.0	61.2	37.5

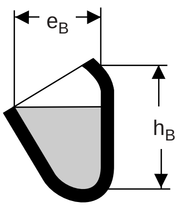
Deep buckets flat rear wall to DIN 15234 for heavy pulverized to coarse materials such as sand, cement, coal.



DIN 15234

Width b_B (mm)	Projection e_B (mm)	Height h_B (mm)	Weight of Bucket in kg with steelsheet Thickness in mm						Bucket capacity v_B in litres
			2	3	4	5	6	8	
160	(125)	160	1.17	1.78					1.2
	140	180	1.38	2.08					1.5
200	(140)	180	1.59	2.41	3.24				1.9
	160	200	1.85	2.80	3.76				2.36
250	(160)	200	2.15	3.26	4.37				3
	180	224	2.49	3.77	4.96				3.75
315	(180)	224		4.44	5.95	7.72			4.75
	200	250		5.09	6.82	8.59			6
400	224	280		7.03	9.40	11.8			9.5
500	250	315			12.8	16.1	19.4		15
630	280	355			17.6	22.1	26.6		23.6
800	315	400				30.6	36.9	49.6	37.5
1000	355	450				42.0	50.3	67.0	60

Deep buckets with curved rear wall to DIN 15235 for light flowing or rolling materials such as fly ash, potatoes.

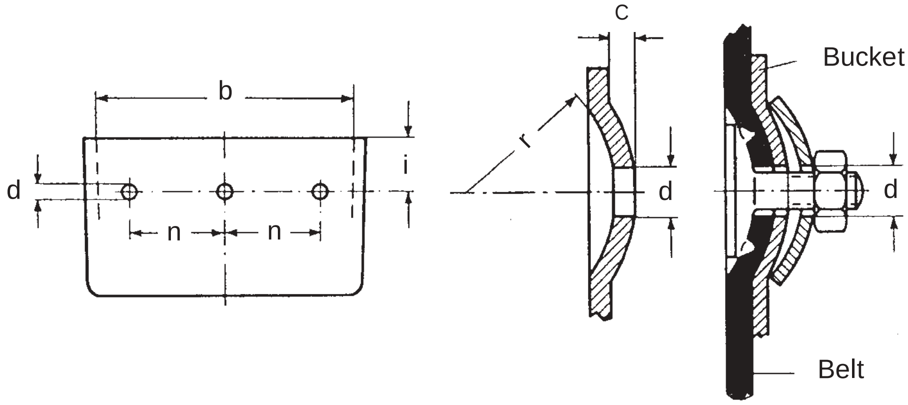


DIN 15235

Width b_B (mm)	Projection e_B (mm)	Height h_B (mm)	Weight of Bucket in kg with steelsheet Thickness in mm						Bucket capacity v_B in litres
			2	3	4	5	6	8	
160	140	200	1.51	2.28					1.5
200	160	224	2.04	3.07	4.15				2.36
250	180	250	2.74	4.14	5.56				3.75
315	200	280		5.59	7.41	9.46			6
400	224	315		7.72	10.4	13.0			9.5
500	250	355			14.1	17.7	21.4		15
630	280	400			19.2	24.1	29.0		23.6
800	315	450				32.5	39.3	37.5	
1000	355	500				44.5	53.5	71.2	60

Bucket Attachment

The bucket made of steel, plastic or rubber may be bolted to the belt. The principal factor to consider is resistance the belt carcase has to bolt pull through.



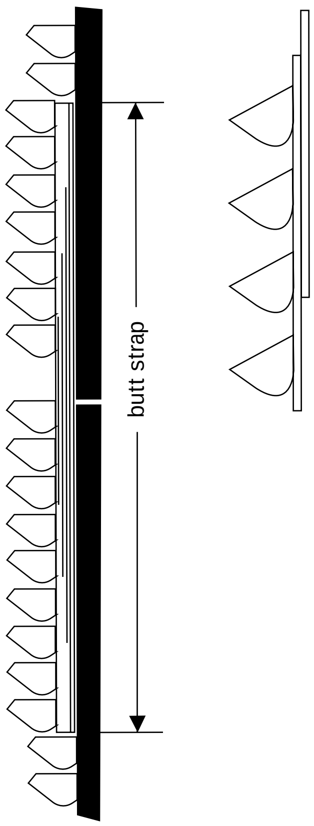
In addition to this most commonly used method of attachment, there are a number of special, partly patented possibilities such as attachment to rear bolt mouldings, vulcanized on rubber plies or flexible steel plate.

Often to prevent dirt penetration and thereby reduce wear, **soft rubber pads** are fitted between belt and bucket.

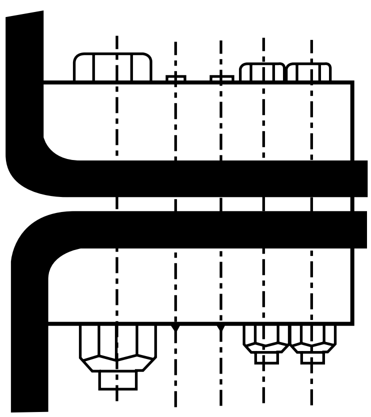
Endless Joint

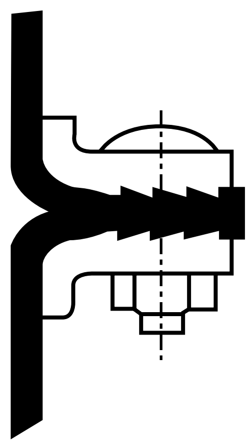
Elevator belts can be made endless by the following methods.

- **Hot vulcanized splice**
- **Bolted lap joint**
- **Bolted on butt strap**
  This method of joining is preferred on fast moving elevators for instance if the velocity is greater than 3 m/s. The plies of the butt strap should be stepped down.
- **Angle joint**
  This type is the simplest and most cost effective but cannot be used in all cases. The pulley diameter and the bending zone of the belt around the angle, have to be taken into consideration.

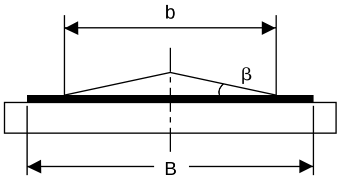


Bolted on butt strap





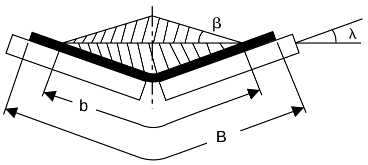
Flat Carrying Idlers



v = 1 m/s
inclination = 0°

Belt Width B (mm)	Surcharge Angle β	Volume Steam Q_V (m^3/h)	Belt Width B (mm)	Surcharge Angle β	Volume Stream Q_V (m^3/h)
300	5°	4	1200	5°	84
	10°	8		10°	168
	15°	11		15°	256
	20°	16		20°	347
400	5°	8	1400	5°	115
	10°	15		10°	232
	15°	23		15°	353
	20°	31		20°	479
500	5°	12	1600	5°	152
	10°	25		10°	306
	15°	38		15°	465
	20°	52		20°	632
650	5°	22	1800	5°	194
	10°	45		10°	391
	15°	68		15°	594
	20°	93		20°	807
800	5°	35	2000	5°	240
	10°	71		10°	486
	15°	108		15°	738
	20°	147		20°	1003
1000	5°	57	2200	5°	300
	10°	114		10°	604
	15°	174		15°	916
	20°	236		20°	1245

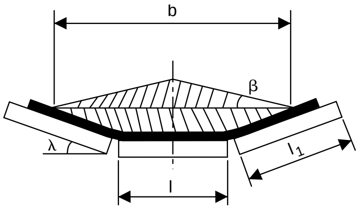
2 Roll Troughing Idlers



v = 1 m/s
inclination = 0°

Belt Width B (mm)	Surcharge Angle β	Troughing Angle λ				
		20°	30°	35°	40°	45°
300	0	14	18	20	21	–
	10	20	24	26	26	–
	15	24	27	28	28	–
	20	28	30	31	31	–
400	0	28	37	40	42	–
	10	41	48	51	51	–
	15	48	55	56	56	–
	20	55	61	62	61	–
500	0	46	62	67	70	–
	10	68	81	85	86	–
	15	80	91	93	93	–
	20	92	101	103	101	–
650	0	82	111	121	126	129
	10	122	145	151	153	151
	15	143	163	167	167	163
	20	165	181	184	182	176
800	0	129	174	190	198	202
	10	192	228	238	240	238
	15	225	255	262	262	256
	20	259	285	288	285	275
1000	0	208	201	305	320	325
	10	309	367	382	387	382
	15	362	412	422	422	412
	20	485	459	464	459	443
1200	0	306	413	449	469	477
	10	455	539	561	568	561
	15	532	605	620	620	605
	20	613	673	681	673	650
1400	0	423	570	619	648	659
	10	628	744	775	785	775
	15	735	835	856	855	835
	20	846	930	941	930	899
1600	0	558	752	817	856	869
	10	829	982	1022	1036	1022
	15	970	1102	1129	1129	1102
	20	1117	1227	1242	1227	1186
1800	0	712	960	1042	1092	1109
	10	1058	1253	1305	1321	1305
	15	1237	1406	1441	1441	1406
	20	1425	1566	1584	1566	1513
2000	0	885	1193	1295	1356	1378
	10	1314	1557	1621	1642	1621
	15	1537	1747	1790	1790	1747
	20	1771	1945	1968	1945	1880
2200	0	1099	1401	1608	1604	1711
	10	1632	1934	2013	2039	2013
	15	1909	2169	2223	2223	2169
	20	2199	2415	2444	2415	2334

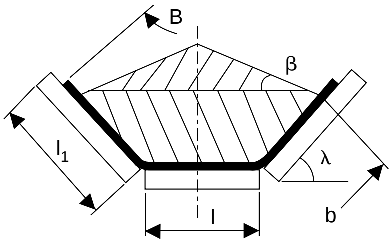
3 Roll Troughing Idlers



v = 1 m/s
inclination = 0°

Belt Width B (mm)	Surcharge Angle β	Troughing Angle λ 20°	30°	35°	40°	45°
400	0	21	30	34	–	–
	10	35	43	47	–	–
	15	42	50	53	–	–
	20	50	57	60	–	–
500	0	36	51	58	–	–
	10	59	73	79	–	–
	15	72	84	90	–	–
	20	85	97	102	–	–
650	0	67	95	108	118	127
	10	109	134	145	153	159
	15	131	155	165	176	176
	20	155	176	184	190	193
800	0	105	149	168	185	198
	10	171	210	227	240	249
	15	206	243	257	268	276
	20	243	276	289	299	303
1000	0	173	246	278	304	326
	10	280	344	370	391	407
	15	336	396	419	436	448
	20	394	449	469	484	492
1200	0	253	360	406	445	477
	10	411	505	543	573	596
	15	493	580	614	640	658
	20	578	659	688	709	722
1400	0	355	504	567	622	666
	10	572	703	755	797	828
	15	685	806	852	888	912
	20	803	915	954	984	1001
1600	0	472	669	753	825	883
	10	758	931	1000	1055	1096
	15	906	1067	1128	1175	1207
	20	1062	1209	1263	1301	1323
1800	0	605	858	965	1057	1131
	10	969	1194	1279	1350	1402
	15	1159	1364	1443	1502	1543
	20	1357	1546	1614	1662	1690
2000	0	750	1064	1197	1311	1404
	10	1204	1478	1588	1675	1741
	15	1439	1694	1791	1865	1916
	20	1685	1919	2003	2064	2099
2200	0	948	1343	1509	1650	1765
	10	1509	1855	1990	2099	2178
	15	1801	2121	2241	2332	2393
	20	2107	2399	2503	2576	2618

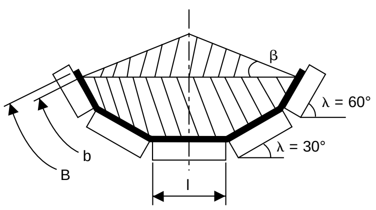
Deep Trough



$l < l_1$
v = 1 m/s
inclination = 0°

Belt Width B (mm)	Surcharge Angle β	Troughing Angle λ 20°	30°	35°	40°	45°
1000	0	196	275	307	333	352
	10	301	369	394	413	424
	15	356	417	440	454	462
	20	413	469	487	498	501
1200	0	286	401	449	487	516
	10	441	540	577	605	623
	15	521	612	644	666	678
	20	605	687	714	731	737
1400	0	393	552	617	670	711
	10	606	743	795	834	859
	15	717	843	888	919	936
	20	834	947	985	1008	1017
1600	0	512	720	806	877	932
	10	794	974	1043	1095	1131
	15	941	1107	1166	1209	1234
	20	1095	1245	1295	1328	1342
1800	0	651	917	1026	1117	1188
	10	1012	1242	1330	1396	1442
	15	1199	1410	1487	1541	1574
	20	1396	1587	1652	1693	1712
2000	0	807	1136	1272	1384	1473
	10	1255	1540	1649	1732	1789
	15	1488	1750	1845	1913	1954
	20	1732	1970	2050	2103	2127
2200	0	1012	1423	1591	1731	1839
	10	1567	1922	2057	2159	2228
	15	1856	2182	2299	2382	2431
	20	2158	2454	2552	2615	2643

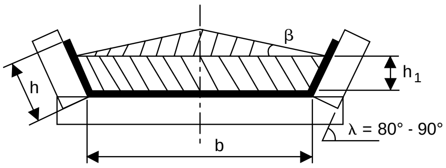
Garland Idlers



$l = l_1 = l_2$
v = 1 m/s
inclination = 0°

Belt Width B (mm)	Surcharge Angle β	Troughing Angle λ 25°/55°	30°/60°
800	0	–	210
	10	–	260
	15	–	286
	20	–	313
1000	0	345	349
	10	425	427
	15	467	468
	20	510	510
1200	0	504	516
	10	623	630
	15	684	689
	20	749	751
1400	0	702	722
	10	864	876
	15	948	957
	20	1036	1041
1600	0	915	947
	10	1132	1153
	15	1244	1260
	20	1362	1371
1800	0	1174	1218
	10	1449	1476
	15	1592	1609
	20	1741	1749
2000	0	1466	1527
	10	1806	1846
	15	1982	2011
	20	2167	2185
2200	0	1838	1868
	10	2255	2252
	15	2472	2452
	20	2699	2661

Box Section Belt

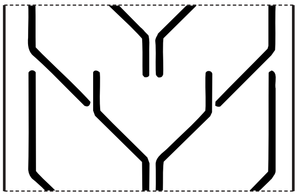


v = 1 m/s
inclination = 0°

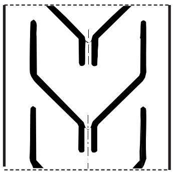
Belt Width B (mm)	Surcharge Angle β	Edge Zone h (mm) 100	125	150
800	0	151	169	187
	10	208	216	226
	15	237	241	244
	20	266	266	266
1000	0	201	234	262
	10	302	320	334
	15	362	367	378
	20	410	417	421
1200	0	252	295	338
	10	410	435	464
	15	489	511	532
	20	576	590	601
1400	0	302	360	414
	10	529	568	604
	15	648	676	705
	20	774	792	810
1600	0	352	421	489
	10	662	709	756
	15	824	860	896
	20	993	1015	1040
1800	0	403	482	565
	10	806	860	921
	15	1018	1058	1105
	20	1238	1267	1299

Conveying of Sinter

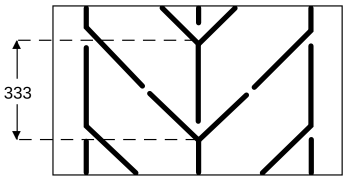
Steep Conveying Belts



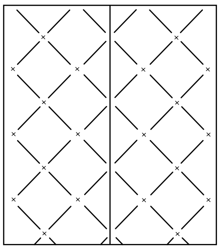
Chevron Profile



Chevron Profile



High Chevron Profile



Multiprof

Beldt Width B (mm)	Surcharge Angle β	CHEVRON 16 mm	Q_v (m^3/h)	HIGH CHEVRON 32 mm	Q_v (m^3/h)
400	0 5 10 15 20	C 330/14	25 33 40 49 58	————	——
400	0 5 10 15 20	C 390/15Z	25 33 40 49 58	————	——
500	0 5 10 15 20	C 430/16	43 56 68 85 100	HC 450/32	48 63 78 94 110
600	0 5 10 15 20	C 530/16	65 86 106 128 151	HC 450/32	42 57 72 88 105
650	0 5 10 15 20	C 530/16	65 86 106 128 151	HC 450/32	42 57 72 88 105
650	0 5 10 15 20	————	——	HC 600/32	88 115 141 160 189
800	0 5 10 15 20	C 650/16	97 129 160 193 227	HC 600/32	78 105 132 160 189
1000	0 5 10 15 20	C 800/16	149 196 244 293 345	HC 1000/32	149 196 244 293 345
1200	0 5 10 15 20	C 1000/16	235 307 384 461 542	HC 1000/32	235 307 384 461 542
1400	0 5 10 15 20	————	——	HC 1200/32	348 454 562 673 789
1600	0 5 10 15 20	————	——	HC 1200/32	326 432 541 652 769

Note

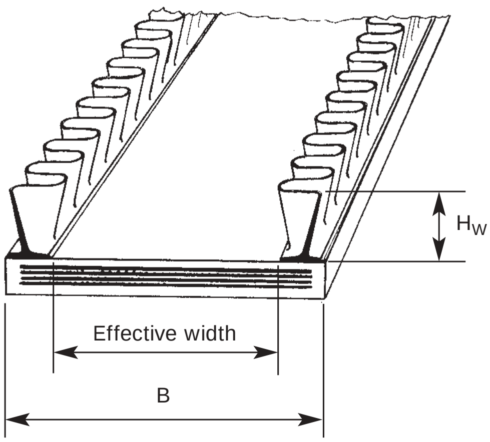
With sticky materials, a surcharge angle ca.5° higher than that for smooth surface belting can be used.

The table values are based on v = 1 m/s
20° troughing

For Multiprof belts the values for CHEVRON belts are valid.

Corrugated Sidewall Belting Without Cleats

The table values are valid for
Speed v = 1 m/s
Gradient angle δ = 0°
Fill factor 1



Corrugated Sidewall height 60 mm

Effective width (mm)	Surcharge Angle				
	0°	5°	10°	15°	20°
200	25	28	32	35	38
300	38	45	52	60	67
400	50	63	76	89	103
500	63	83	103	123	145
600	76	104	133	162	194
700	88	127	166	206	249
800	101	151	202	255	310
900	113	177	242	309	379
1000	126	205	285	367	454

Corrugated Sidewall height 80 mm

Effective width (mm)	Surcharge Angle				
	0°	5°	10°	15°	20°
200	40	43	46	49	53
300	59	66	74	81	89
400	79	92	105	118	132
500	99	119	139	159	181
600	119	147	176	206	237
700	139	177	216	257	299
800	158	209	260	313	368
900	178	242	307	374	444
1000	198	277	357	439	526

Corrugated Sidewall height 120 mm

Effective width (mm)	Surcharge Angle				
	0°	5°	10°	15°	20°
200	68	72	75	78	82
300	103	110	117	124	132
400	137	149	162	175	189
500	171	191	211	231	253
600	205	234	262	292	323
700	239	278	317	358	400
800	274	324	375	428	483
900	308	372	436	503	573
1000	342	421	501	583	670

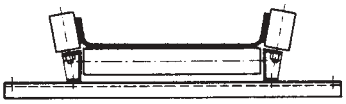
Corrugated Sidewall height 200 mm

Effective width (mm)	Surcharge Angle				
	0°	5°	10°	15°	20°
200	126	129	132	136	139
300	189	196	203	211	218
400	252	265	277	291	304
500	315	335	355	375	397
600	378	406	435	465	496
700	441	480	519	559	602
800	504	554	606	658	714
900	567	631	696	762	832
1000	630	709	789	871	958

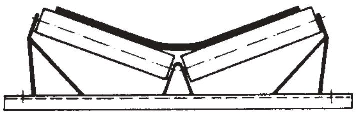
Idler Rollers Carrying and Return

Carrying Side Rollers

Flat Belt

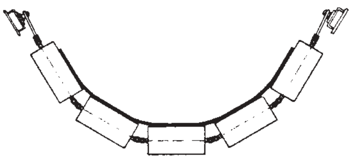


Box Section Belt



2 Part Idlers

3 Part Idlers



5 Part Garland Idlers

Return Idlers

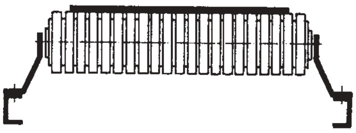
Mass m'_R (kg)

Belt width B (mm)	Idler Rollers	Idler Roller Diameter							
		51	63.5	88.9	108	133	159	193.7	219.1
300	flat	1.6	2.2	3.2					
	2 part	2.3	3.4	4.1					
400	flat	2.0	2.7	3.9	5.6				
	2 part	2.6	3.7	4.7	6.6				
	3 part	2.9	4.4	5.4	7.3				
500	flat	2.2	3.2	4.5	6.6				
	2 part	2.8	4.1	5.5	7.8				
	3 part	3.2	4.6	6.1	8.4				
650	flat		4.0	5.5	8.0	10.8			
	2 part		4.7	6.3	9.0	12.1			
	3 part		5.4	7.0	9.8	13.1			
800	flat		4.7	6.7	9.8	13.3			
	2 part		5.6	7.4	10.6	14.2			
	3 part		6.5	8.3	11.6	15.6			
	5 part			9.0	12.4	16.3			
1000	flat			9.4	11.7	15.9	21.9		
	2 part			11.3	13.2	17.8	24.7		
	3 part			13.0	13.6	18.2	26.3		
	5 part			13.8	14.2	18.9	28.0		
1200	flat				14.2	19.3	26.1		
	2 part				15.0	20.5	28.0		
	3 part				16.3	22.3	24.5		
	5 part				17.2	21.7	31.9		
1400	flat					21.8	29.3		
	2 part					23.3	31.6		
	3 part					25.0	35.5		
	5 part					24.3	35.0		
1600	flat					25.1	33.4		
	2 part					26.5	35.0		
	3 part					28.0	38.7		
	5 part					28.5	39.3		
1800	flat					27.6	37.8		
	2 part					29.1	39.5		
	3 part					30.7	42.4		
	5 part					31.5	42.5		
2000	flat					30.2	40.2	69.1	
	2 part					31.8	43.3	76.4	
	3 part					33.3	47.0	80.1	
	5 part					33.8	46.5	89.5	
2200	flat						46.5	77.8	88.0
	2 part						49.0	82.6	97.1
	3 part						50.1	93.2	111.0
	5 part						51.0	95.5	111.8

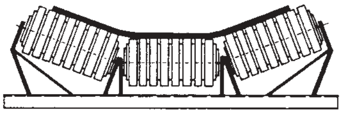
Impact Idlers

To moderate the effect of impact cushion ring idlers may be installed at the loading point.

Loading point - Carrying idler



Flat belt with cushion rings



3 part cushion ring troughing idlers

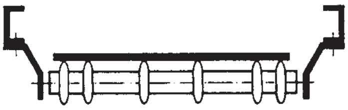
Mass m'_R (kg)

Beldt width (mm)	Tube-dia (mm)	Ring-dia (mm)	Cushion ring idlers 1 part	3 part
1000	88.9	156	19.1	21.1
1200	108	180	30.8	32.8
1400	108	180	35.7	40.5
1600	108	180	42.2	45.0
1800	133	215	67.1	71.1
2000	133	215	73.6	77.6
2200	133	215	80.1	84.1

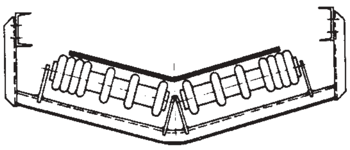
Disc Support Idlers

Disc idlers can be installed on the return side run to reduce the effect of dirt build-up.

To assist tracking of the belt it is recommended that guide rollers be installed at ca. 50 m intervals.



Flat belt with disc idlers



V-troughing with disc idlers

V-troughing - Suspended disc idlers

Mass m'_R (kg)

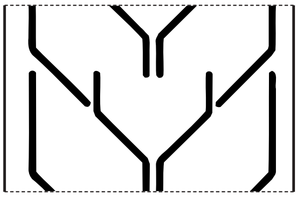
Belt width (mm)	Tube-dia (mm)	Disc-dia (mm)	Return idlers 1 part	3 part
400	51	120	4.0	5.0
500	57	133	5.7	6.8
650	57	133	6.8	8.1
800	63.5	150	11.7	13.2
1000	63.5	150	13.0	14.5
1200	88.9	180	22.2	23.9
1400	88.9	180	24.2	25.9
1600	108.0	180	31.9	33.9
		215	42.0	44.5
1800	108.0	180	34.3	36.3
		215	44.9	47.3
2000	108.0	180	38.3	40.0
		215	48.8	51.8
2200	133	215	59.8	62.8
		250	73.8	76.8

For number and arrangement of disc idlers see Appendix M.1.

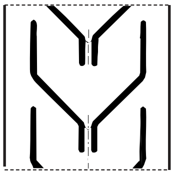
	Belt type (mm)	Carcase thickness (kg/m^2)	Carcase weight	Belt Weight m'$_G$ (kg/m^2) Sum of carrying and pulley side covers (mm)						
				3	4	5	6	8	10	12
SUPERFORT Belts	S 200/3	2.7	3.1	6.6	7.7	8.9	10.0	12.3	14.6	16.9
	S 250/3	2.8	3.2	6.7	7.8	9.0	10.1	12.4	14.7	17.0
	S 315/3	3.0	3.4	6.9	8.0	9.2	10.3	12.6	14.9	17.2
	S 315/4	3.7	4.3	7.8	8.9	10.1	11.2	13.5	15.8	18.1
	S 400/3	3.2	3.7	7.2	8.3	9.5	10.6	12.9	15.2	17.5
	S 400/4	4.1	4.6	8.1	9.2	10.4	11.5	13.8	16.1	18.4
	S 500/3	3.6	4.0	7.5	8.6	9.8	10.9	13.2	15.5	17.8
	S 500/4	4.3	5.0	8.5	9.6	10.8	11.9	14.2	16.5	18.8
	S 630/3	3.9	4.3	7.8	8.9	10.1	11.2	13.5	15.8	18.1
	S 630/4	4.8	5.3	8.8	9.9	11.1	12.2	14.5	16.8	19.1
	S 630/5	5.5	6.2	9.7	10.8	12.0	13.1	15.4	17.7	20.0
	S 800/3	4.5	5.0	8.5	9.6	10.8	11.9	14.2	16.5	18.8
	S 800/4	5.2	5.8	9.3	10.4	11.6	12.7	15.0	17.3	19.6
	S 800/5	6.0	6.7	10.2	11.3	12.5	13.6	15.9	18.2	20.5
	S 1000/4	6.1	6.8	10.3	11.4	12.6	13.7	16.0	18.3	20.6
	S 1000/5	6.5	7.3	10.8	11.9	13.1	14.2	16.5	18.8	21.1
	S 1000/6	7.3	8.1	11.6	12.7	13.9	15.0	17.3	19.6	21.9
	S 1250/4	7.2	8.3	11.8	12.9	14.1	15.2	17.5	19.8	22.1
	S 1250/5	7.6	8.6	12.1	13.2	14.4	15.5	17.8	20.1	22.4
	S 1250/6	7.8	8.8	12.3	13.4	14.6	15.7	18.0	20.3	22.6
	S 1600/4	8.7	9.4	12.9	14.0	15.2	16.3	18.6	20.9	23.2
	S 1600/5	9.1	10.5	14.0	15.1	16.3	17.4	19.7	22.0	24.3
	S 1600/6	9.2	10.4	13.9	15.0	16.2	17.3	19.6	21.9	24.2
	S 2000/5	11.0	11.9	15.4	16.5	17.7	18.8	21.1	23.4	25.7
	S 2000/6	11.0	12.7	16.2	17.3	18.5	19.6	21.9	24.2	26.5
	S 2500/6	13.4	14.4	17.9	19.0	20.2	21.3	23.6	25.9	28.2
STARFLEX belts	SF 250/2	1.5	1.8	5.3	6.4	7.6	8.7	11.0	13.3	15.6
	SF 315/2	1.9	1.9	5.4	6.5	7.7	8.8	11.1	13.4	15.7
	SF 400/3	2.5	2.9	6.4	7.5	8.7	9.8	12.1	14.4	16.7
	SF 500/3	3.0	3.2	6.7	7.8	9.0	10.1	12.4	14.7	17.0
	SF 500/4	3.5	4.0	7.5	8.6	9.8	10.9	13.2	15.5	17.8
	SF 630/4	4.1	4.4	7.9	9.0	10.2	11.3	13.6	15.9	18.2
	SF 800/4	4.6	5.4	8.9	10.0	11.2	12.3	14.6	16.9	19.2
	SF 1000/4	5.3	6.1	9.6	10.7	11.9	13.0	15.3	17.6	19.9
DUNLOFLEX belts	D 160	2.3	2.7	6.2	7.3	8.5	9.6	11.9	14.2	16.5
	D 200	2.7	3.1	6.6	7.7	8.9	10.0	12.3	14.6	16.9
	D 250	3.0	3.6	7.1	8.2	9.4	10.5	12.8	15.1	17.4
	D 315	3.2	3.7	7.2	8.3	9.5	10.6	12.9	15.2	17.5
	D 400	3.7	4.3	7.8	8.9	10.1	11.2	13.5	15.8	18.1
	D 500	4.1	4.7	8.2	9.3	10.5	11.6	13.9	16.2	18.5
	D 630	4.5	5.0	8.5	9.6	10.8	11.9	14.2	16.5	18.8
	D 800	4.8	5.5	9.0	10.1	11.3	12.4	14.7	17.0	19.3
TRIOFLEX belts	T 315	4.0	4.8	8.3	9.4	10.6	11.7	14.0	16.3	18.6
	T 400	4.4	5.3	8.8	9.9	11.1	12.2	14.5	16.8	19.1
	T 500	5.0	5.9	9.4	10.5	11.7	12.8	15.1	17.4	19.7
	T 630	5.5	6.5	10.0	11.1	12.3	13.4	15.7	18.0	20.3
	T 800	6.0	7.2	10.7	11.8	13.0	14.1	16.4	18.7	21.0
	T 1000	6.5	7.8	11.3	12.4	13.6	14.7	17.0	19.3	21.6
	T 1250	7.2	8.1	11.6	12.7	13.9	15.0	17.3	19.6	21.9
FERROFLEX belts	F 500	3.2	5.9	9.4	10.5	11.7	12.8	15.1	17.4	19.7
	F 630	3.2	6.4	9.9	11.0	12.2	13.3	15.6	17.9	20.2
	F 800	4.5	8.9	12.4	13.5	14.7	15.8	18.1	20.4	22.7
	F 1000	4.5	9.8	13.3	14.4	15.6	16.7	19.0	21.3	23.6
	F 1250	6.0	12.5	16.0	17.1	18.3	19.4	21.7	24.0	26.3
	F 1600	6.0	14.1	17.6	18.7	19.9	21.0	23.3	25.6	27.9

Weight of 1 mm cover rubber normal quality: 1.15 kg/m^2.

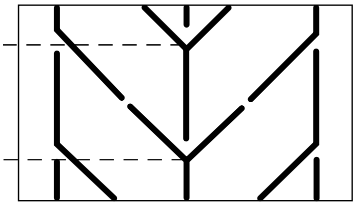
Steep Conveying Belts



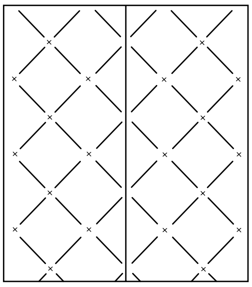
Chevron Profile



Chevron Profile



High Chevron Profile



Multiprof

Belt width (mm)	Belt Type	Profile	Cover Thickness (mm)	Weight (kg/m)
Type CHEVRON				
400	S 200/3	C 330/16	2 + 1	3.5
400	S 200/3	C 390/15Z	2 + 1	3.4
500	S 200/3	C 330/16	2 + 1	4.2
500	S 200/3	C 430/16	2 + 1	4.6
600	S 200/3	C 530/16	2 + 1	5.3
650	S 200/3	C 530/16	2 + 1	5.6
650	S 400/3	C 530/16	3 + 1.5	7.0
800	S 400/3	C 650/16	3 + 1.5	8.5
1000	S 400/3	C 800/16	3 + 1.5	11.2
1200	S 500/4	C 1000/16	3 + 1.5	14.2
Type HIGH-CHEVRON				
500	S 200/3	HC 450/32	2 + 1	5.6
600	S 200/3	HC 450/32	2 + 1	6.3
650	S 200/3	HC 450/32	2 + 1	6.6
650	S 200/3	HC 600/32	2 + 1	7.2
650	S 400/3	HC 450/32	3 + 1.5	8.2
800	S 400/3	HC 600/32	3 + 1.5	9.8
1000	S 500/4	HC 800/32	4 + 2	14.9
1200	S 500/4	HC 1000/32	4 + 2	17.6
1400	S 630/4	HC 1200/32	4 + 2	20.9
1600	S 630/4	HC 1200/32	4 + 2	23.2

Type MULTIPROF

Belt Width (mm)	Belt Type	Profile Width (mm)	Cover Thickness (mm)	Total Thickness (mm)	Weight (kg/m)
550	S 200/3	500	2 + 1	5.7	4.1
650	S 200/3	500	2 + 1	5.7	4.7
800	S 400/3	650	3 + 1.5	7.7	7.6
1000	S 500/4	800	4 + 2	10.3	12.5
1200	S 500/4	800	4 + 2	10.3	15.0

* Belt thickness excluding profile

Profiled Belts

Belt Type	Cover Description	Total Thickness (mm)	Weight (kg/m^2)
S 200/3	Herringbone profile (3) + 1 mm	6.4	6.6
S 200/2	Rufftop profile (3.5) + 1 mm	6.5	5.8

Slider Belting

Belt Type	Cover Description	Total Thickness (mm)	Weight (kg/m^2)
S 200/2	GB-Standard 1.5 + 0 mm smooth	4.9	5.5
S 200/2	GB-Standard 3.5 + 0 mm Rufftop	6.9	6.2
S 200/2	GB-Standard 3 + 0 mm Herringbone	6.1	6.2
S 250/3	GB-Extra 1.5 + 0 mm smooth	5.0	5.6
S 250/3	GB-Extra 3.5 + 0 mm Rufftop	7.0	6.3

Monoply Belts DUNLOPLAST PVG, Quality "RS"

Belt Type	Carcase Thickness (mm)	Belt weight m'_G (kg/m^2) Top and Bottom covers (mm)					
		5 + 3	6 + 3	7 + 3	8 + 3	8 + 4	10 + 4
DLP 630 RS	4	14.85	16.25	17.65	19.05	20.45	21.85
DLP 800 RS	5	15.80	17.20	18.60	20.00	21.40	22.80
DLP 1000 RS	6	16.80	18.20	19.60	21.00	22.40	23.80
DLP 1250 RS	7	18.00	19.40	20.80	22.20	23.60	25.00
DLP 1600 RS	8	20.00	21.40	22.80	24.20	25.60	27.00
DLP 2000 RS	9	22.00	23.40	24.80	26.20	27.60	29.00
DLP 2500 RS	10	24.00	25.40	26.80	28.20	29.60	31.00

Monoply Belts DUNLOPLAST PVC

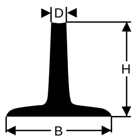
Belt Type	Cover Thickness (mm)	Cover Thickness (mm)	Weight (kg/m^2)
DLP 250 PVC FDA	2 + 0	3	3.75
DLP 250 PVC FDA	2 + 1	4	5.0
DLP 315 PVC FDA	1.5 + 1.5	4	5.0
DLP 400 PVC FDA	1.5 + 1.5	5	6.25
DLP 630 PVC FDA	1.5 + 1.5	7	8.75
DLP 800 PVC FDA	2 + 2	9	11.25
DLP 1000 PVC FDA	2 + 2	11	13.75
DLP 1600 PVC FDA	3 + 3	14	17.5
DLP 2000 PVC FDA	3 + 3	15	18.75
DLP 2500 PVC FDA	3 + 3	16	20.0
DLP 250 PVC BLS	2 + 1	4	5.0
DLP 315 PVC BLS	3 + 1.5	6	7.5
DLP 400 PVC BLS	4 + 2	8	10.0
DLP 630 PVC BLS	5 + 3	11	13.75
DLP 1000 PVC BLS	2 + 2	11	13.75
DLP 315 PVC A	3 + 1.5	6	7.5
DLP 400 PVC A	4 + 2	8	10.0
DLP 630 PVC A	5 + 3	11	13.75
DLP 1000 PVC A	5 + 3	15	18.75
DLP 1600 PVC A	5 + 3	17	21.75

Steel Cord Belts Type `SILVERCORD'

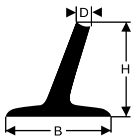
Belt Type	Cord Diameter (mm)	Belt Weight m'$_G$ (kg/m^2) Sum of Carrying and Pulleyside Covers (mm)							
		10	11	12	13	14	15	16	20
ST 500	2.7	16.4	17.5	18.6	19.8	20.9	22.0	23.1	27.6
ST 630	2.7	16.9	18.0	19.2	20.3	21.4	22.5	23.6	28.1
ST 800	3.1	18.4	19.5	20.6	21.7	22.9	24.0	25.1	29.6
ST 900	3.6	19.4	20.5	21.6	22.8	23.9	25.0	26.1	30.6
ST 1000	3.6	19.8	20.9	22.0	23.2	24.3	25.4	26.5	31.0
ST 1150	4.1	20.5	21.6	22.7	23.8	24.9	26.1	27.2	31.7
ST 1250	4.1	21.0	22.1	23.2	24.3	25.5	26.6	27.7	32.2
ST 1400	4.4	22.2	23.3	24.4	25.5	26.6	27.8	28.9	33.4
ST 1600	5.0	23.5	24.6	25.7	26.8	28.0	29.1	30.2	34.7
ST 1800	5.0	24.0	25.1	26.3	27.4	28.5	29.6	30.7	35.2
ST 2000	5.0	24.6	25.8	26.9	28.0	29.1	30.2	31.4	35.8
ST 2250	5.9	28.0	29.1	30.2	31.3	32.5	33.6	34.7	39.2
ST 2500	6.3	29.4	30.5	31.6	32.7	33.9	35.0	36.1	40.6
ST 2750	6.9	30.1	31.2	32.3	33.4	34.5	35.7	36.8	41.3
ST 3150	7.4	32.6	33.7	34.8	35.9	37.1	38.2	39.3	43.8
ST 3500	7.6	34.6	35.7	36.8	37.9	39.1	40.2	41.3	45.8
ST 3750	8.2	36.7	37.8	38.9	40.0	41.1	42.3	43.4	47.9
ST 4000	8.6	38.1	39.3	40.4	41.5	42.6	43.7	44.9	49.3
ST 4250	8.8	39.8	40.9	42.0	43.2	44.3	45.4	46.5	51.0
ST 4500	9.6	41.7	42.8	43.9	45.0	46.2	47.3	48.4	52.9
ST 4750	9.6	43.3	44.5	45.6	46.7	47.8	48.9	50.1	54.5
ST 5000	10.7	45.1	46.2	47.4	48.5	49.6	50.7	51.8	56.3

Weight 1 mm cover rubber normal quality 1.12 kg/m^2.
Minimum cover thickness is 5 mm each side.

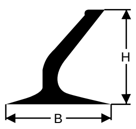
Rubber Cleats



T-Cleat



Sloping T-Cleat



Bucket Cleat

Form	Type Designation	Measurements (mm) H	B	D	Weight (kg/m)
Straight T-Cleat	T 15/20	15	20	8	0.18
	T 20/40	20	40	5	0.27
	T 40/70	40	70	4	0.70
	T 60/80	60	80	4	1.04
	T 80/90	80	90	4	1.55
	T 100/100	100	100	4	2.00
Sloping T-Cleat	TS 50/65	50	65	7	0.88
	TS 70/80	70	80	6	0.82
Bucket Cleat	B 80	80	80	–	1.90
	B 110	110	80	–	2.90
Block Cleat	TB 25/11/ 36	25	36	11	0.49
	TB 50/25/ 80	50	80	25	2.10
	TB 75/10/ 80	75	80	10	1.83
	TB 80/15/100	80	100	15	3.20

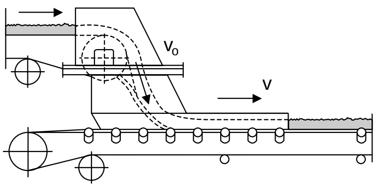
Secondary Resistances

It is possible to obtain a close estimate of secondary resistances from individual components.

$$F_N = F_{Auf} + F_{Sch} + F_{Gr} + F_{Gb} + F_{Tr} \qquad (N)$$

Resistance F_{Auf}

Inertial and frictional resistances due to acceleration of material at the loading point.

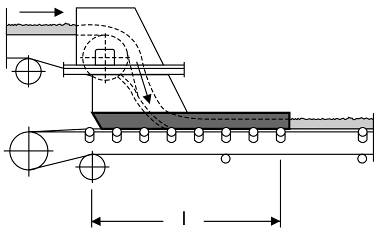


$$F_{Auf} = \frac{Q_m}{3.6} * (v - v_0) \qquad (N)$$

v	(m/s)	Belt speed
v_0	(m/s)	Initial velocity of material in direction of belt travel
Q_m	(t/h	Capacity

Resistance F_{Sch}

Frictional resistance on the skirt plates at the loading point area.



$$F_{Sch} = \frac{Q_m^2 * \mu}{330 * \rho * (v + v_0)^2} * \frac{l}{b^2} \qquad (N)$$

For installations with a normal construction $v > v_0 > 0$

μ	(-)	Friction value between material and skirt plate	
		for grain	0.25
		dry coal	0.35 - 0.4
		damp overburden	0.5 - 0.6
ρ	(t/m^3)	Bulk density	
l	(m)	Length of sidewalls at loading point	
b	(m)	Distance between skirt plates	

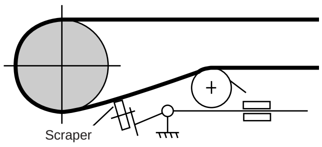
The length of the skirt plates should be not less than the acceleration distance

$$l_{min} = v^2 - v_o^2 / (2 * g * \mu)$$

μ = 0.6 for small piece loads
μ = 0.8 for heavy piece loads

Resistance F_{Gr}

Frictional Resistance due to belt cleaners.



$$F_{Gr} = \mu * p * A \qquad (N)$$

μ	(-)	Friction factor between cleaner and belt in general μ = 0.6 à 0.75
p	(N/mm^2)	Pressure between cleaner and belt p = 0.03 bis 0.1 N/mm^2
A	(mm^2)	Effective contact area between cleaner and belt

For simple scrapers the following estimate formula can be applied:

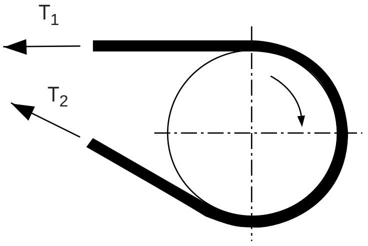
$$F_{Gr} = k_R * B \qquad (N)$$

B	(m)	Belt width
k_R	(N/m)	Frictional resistance

Friction values for k_R

Scraper Thickness (mm)	Pressure (N/mm^2) normale		élevée		
	15	20	20	25	30
k_R	340	450	1500	1875	2250

Resistance F_{Gb}

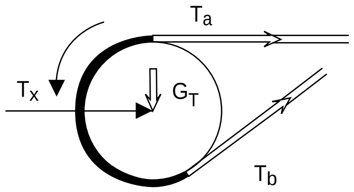


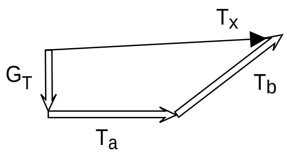
Belt Bending Resistance

$$F_{Gb} = c * B * \left(k + \frac{T_m}{B} \right) * \frac{d}{D} \qquad (N)$$

T_m	(N)	average belt tension
d	(mm)	belt thickness
D	(mm)	pulley diameter
c	(-)	factor for the belt textile belts c = 0.09 steel cord belts c = 0.12
k	(N/mm)	factor for the belt textile belts k = 14 N/mm steel cord belts k = 20 N/mm

Resistance F_{Tr}





Pulley bearing resistance for non-driving pulleys. These resistances like belt bending resistances, are small and need be determined only in special cases.

$$F_{Tr} = 0.005 * T_x * \frac{d_w}{D} \qquad (N)$$

d_w	(mm)	shaft diameter of bearing seating
D	(mm)	pulley diameter
G_T	(N)	pulley weight bearing force vectorial sum from both the belt tension force on the pulley and the weight of the rotating parts of the pulley (axle loading)

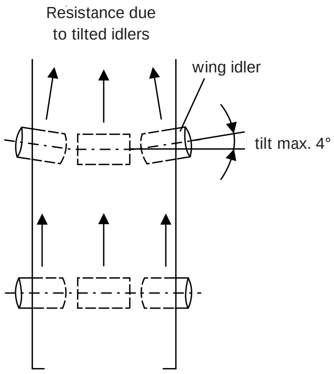
$T_x = \sqrt{(T_a + T_b)^2 + G_T^2}$ (N) with angle of wrap $\alpha = 180°$

0.005 friction coefficient of roller bearings.

Special Resistances

The special resistances occur beyond the loading point area and can be calculated individually.

Resistance F_s



Resistance caused by forward tilt of idler rollers

Carrying side $$F_{so} = Z_{Rst} * C * \mu * \cos\delta * \sin\epsilon\,(m'_G + m'_L) \quad (N)$$

Return side $$F_{su} = Z_{Rst} * C * \mu * \cos\delta * \sin\epsilon * m'_G \quad (N)$$

Z_{Rst}	(pieces)	number of tilted rollers on carrying and return run
C	(-)	load factor for λ = 30° C = 0.4
		for λ = 45° C = 0.5
μ	(-)	friction value between belt and bare roller: 0.2 - 0.7
δ	(-)	gradient angle of the installation
ϵ	(-)	tilt angle of the rollers
		ϵ = 1° à 3° (maximum 4°)

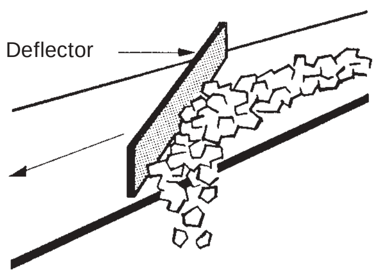
Resistance F_{Sch}

Frictional resistance from skirt plates beyond the loading point area

$$F_{Sch} = \frac{C * Q_m^2 * \mu}{1316 * \rho * v^2} * \frac{l}{b^2} \quad (N)$$

Q_m	(t/h)	capacity
μ	(-)	friction value between material and skirt plate (for values see Page D1)
v	(m/s)	belt speed
ρ	(t/m³)	bulk density
l	(m)	length of skirt plate
b	(m)	distance between skirt plates
C	(-)	factor for surcharge angle
		β = 5° C = 0.83
		β = 10° C = 0.70
		β = 15° C = 0.59
		β = 20° C = 0.49

Resistance F_L

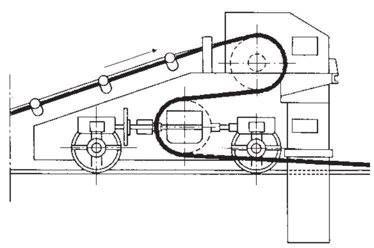


Resistance due to sideways discharge (values)

Load deflector on flat belts running horizontally.

Belt Width (mm)	≤ 500	650 - 800	1000 - 2000
F_L (N)	800	1500	3000 - 3500

Resistance F_B

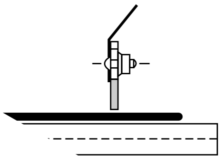


Resistance due to trippers (values)

The resistance due to trippers can be estimated and depends on the belt width.

	Resistance F_B (N)							
Belt Width (mm)	650	800	1000	1200	1400	1600	1800	2000
Fixed tripper	1500	2000	2300	3000	4000	4000	5000	5500
Moveable tripper	1500	2000	2500	4000	4500	5200	5500	6000

Resistance F_{Mf}



Resistance from skirting material beyond the loading point area.

$$F_{Mf} = (80 \text{ to } 120) * l_f \qquad (N)$$

l_f (m) length of skirting material that makes seal with belt
$l_f = 2 *$ conveying length

Resistance F_{Ba}

Resistance to motion of bunker drag-out belts

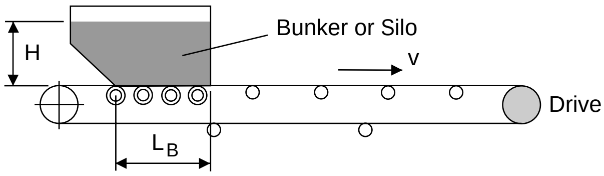
Can be determined according to the following empirical formula:

$$F_{Ba} = 1000 * k * B^2 * L_B * \sigma \qquad (N)$$

B	(m)	drag-out width, silo opening	
H	(m)	material height in bunker	
L_B	(m)	length of out-let	
v	(m/s)	belt speed	
σ	(-)	continuous running	$\sigma = 1$
		up to 30 charges per hour	$\sigma = 1.2$
		over 30 charges per hour	$\sigma = 1.3$

Factor k

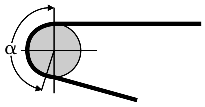
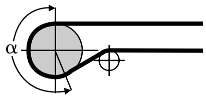
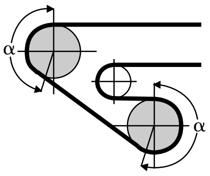
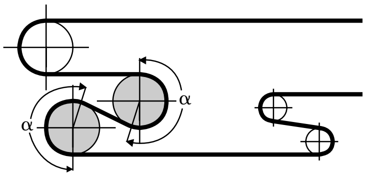
H (m)	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
k	8.0	9.4	10.2	11.0	11.5	12.3	12.7	13.4	13.7	14.5	14.7



Factor c_1

$$c_1 = 1 + c_2 = 1 + \frac{1}{e^{\mu\alpha} - 1}$$

Factor c_2

Angle of wrap $\alpha°$	Friction coefficient μ 0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
160	3.10	1.92	1.34	0.99	0.76	0.60	0.49	0.40
170	2.90	1.78	1.24	0.91	0.70	0.55	0.44	0.36
180	2.71	1.66	1.14	0.84	0.64	0.50	0.40	0.32
190	2.54	1.55	1.06	0.74	0.59	0.46	0.36	0.29
200	2.39	1.45	0.99	0.72	0.54	0.42	0.33	0.26
210	2.26	1.36	0.93	0.67	0.50	0.38	0.30	0.24
220	2.14	1.28	0.87	0.62	0.46	0.35	0.27	0.22
230	2.02	1.22	0.81	0.58	0.43	0.33	0.25	0.20
240	1.92	1.14	0.76	0.54	0.40	0.30	0.23	0.18
360	1.14	0.64	1.40	0.26	0.18	0.13	0.09	0.07
370	1.10	0.61	0.38	0.25	0.17	0.12	0.08	0.06
380	1.06	0.59	0.36	0.24	0.16	0.11	0.08	0.05
390	1.03	0.56	0.35	0.22	0.15	0.10	0.07	0.05
400	0.99	0.54	0.33	0.21	0.14	0.10	0.07	0.05
410	0.96	0.52	0.31	0.20	0.13	0.09	0.06	0.04
420	0.93	0.50	0.31	0.19	0.13	0.08	0.06	0.04
430	0.89	0.48	0.29	0.18	0.12	0.08	0.05	0.04
440	0.87	0.46	0.27	0.17	0.11	0.07	0.05	0.03
450	0.84	0.45	0.26	0.16	0.11	0.07	0.05	0.03
460	0.81	0.43	0.25	0.16	0.10	0.07	0.04	0.03
470	0.79	0.41	0.24	0.15	0.09	0.06	0.04	0.03

Value $e^{\mu\alpha}$

Angle of wrap $\alpha°$	Friction coefficient μ 0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
180	1.37	1.60	1.88	2.20	2.57	3.00	3.51	4.21
190	1.39	1.64	1.94	2.29	2.70	3.18	3.75	4.44
200	1.41	1.69	2.01	2.40	2.85	3.40	4.04	4.82
210	1.44	1.73	2.08	2.50	3.00	3.60	4.32	5.20
220	1.46	1.78	2.16	2.60	3.17	3.89	4.65	5.64
230	1.49	1.83	2.23	2.73	3.32	4.07	4.97	6.09
240	1.52	1.88	2.32	2.85	3.51	4.34	5.35	6.60
320	1.75	2.32	3.06	4.05	5.35	7.05	9.35	12.40
340	1.81	2.44	3.27	4.40	5.95	8.00	10.70	14.50
360	1.88	2.57	3.52	4.80	6.58	9.00	12.40	17.00
380	1.94	2.71	3.77	5.25	7.30	10.20	14.10	19.80
400	2.01	2.86	4.05	5.74	8.15	11.60	16.50	23.20
420	2.08	3.01	4.35	6.25	9.00	13.00	18.60	27.10

Belt Safety

The safety factor S is reduced by additional stresses which cannot be determined by calculation.

Steady State

$$S_B = \frac{1}{1 - (r_0 + r_1 + r_2)}$$

Non-Steady State

$$S_A = \frac{1}{1 - (r_0 + r_1)}$$

With= the factor r_0 and r_2 the various additional stresses are determined.

Reduction r_0

With this factor the time-strength behaviour under dynamic conditions is determined. Low r_0 values result from favourable working conditions with few additional stresses from the physical and chemical characteristics of the load, few changes of conveying elevation, infrequent stopping and starting, regular servicing and no extreme environmental influences. Belts with planned relatively low working durations, can be calculated with low r_0 values.

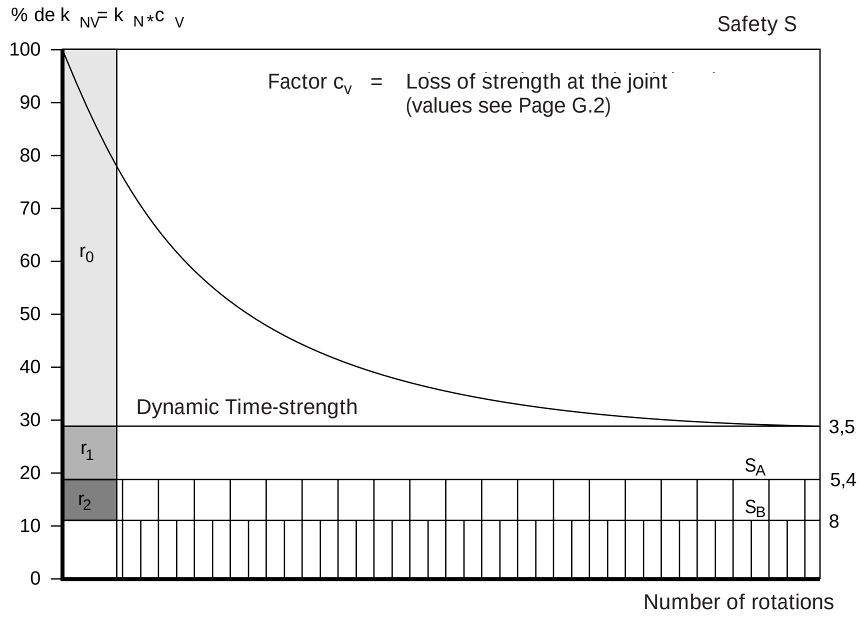
Reduction r_1

This factor takes into consideration the influence of additional strains caused by the belt flexing around pulleys, transition zones, belt turnovers and vertical curves. The values of r_1 apply to standard dimensioning of the pulleys and transitions without belt turnover.

Reduction r_2

This factor takes into account the peak stresses of temporary occurrence for instance at start-up and stopping.

Time-Strength Behaviour

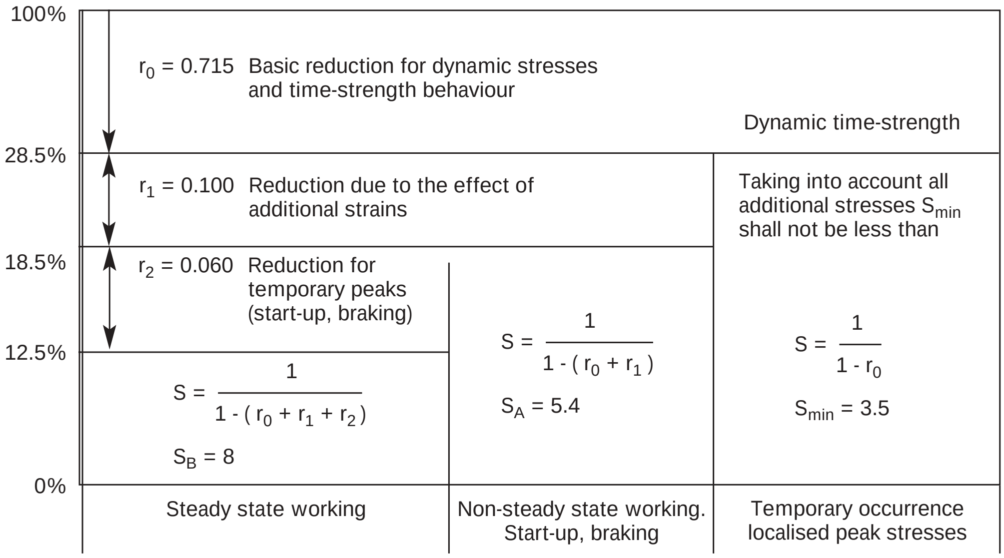


The above reductions r and safety S apply to EP belts, normal running conditions.

Safety Factor

K_{NV} = Breaking Strength at Joint

Belt Safety Factor (DIN 22101)



The values given are valid for a textile belt under normal working conditions.

Minimum values based on joint strength, safety factor S and reduction r

		Safety factors and reductions [1]					
carcase material	working conditions	basic reduction	non-continuous peak stresses		maximum stress steady-state working		
		r_0	r_1	S_{insta}	r_1	r_2	S_{sta}
B (Cotton) P (Polyamide) E (Polyester)	favourable normal unfavourable	≥ 0.691 ≥ 0.715 ≥ 0.734	≥ 0.100	≥ 4.8 ≥ 5.4 ≥ 6.0	≥ 0.100	≥ 0.060	≥ 6.7 ≥ 8.0 ≥ 9.5
Steel Cord	favourab le normal unfavourable	≥ 0.641 ≥ 0.665 ≥ 0.684	≥ 0.150	≥ 4.8 ≥ 5.4 ≥ 6.0	≥ 0.150	≥ 0.060	≥ 6.7 ≥ 8.0 ≥ 9.5

[1]) With extraordinary favourable installation conditions (for instance untroughed belt or belt conserving start-up conditions) lower values for r_1 and r_2 can be used. The mathematical inter relationship between the additional stresses and the values r_1 and r_2 is at present still unknown.

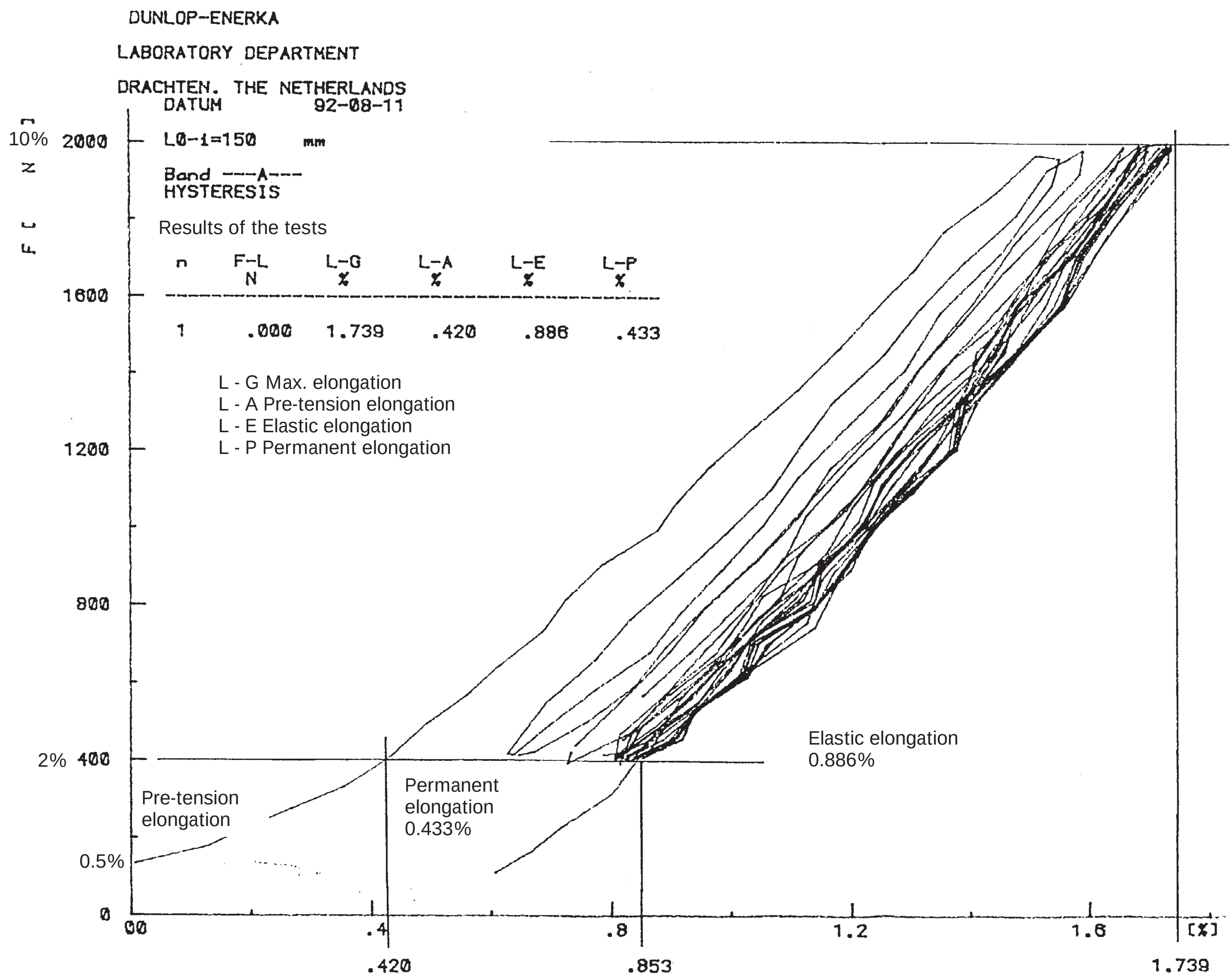
S_{insta} belt safety at start-up (braking)
S_{sta} belt safety running

Elastic Characteristics	When working a conveyor belt has to undergo constant load changes and hence constant length changes. These are catered for by the elastic characteristics of the carcase and are also partly reduced by the take-up system. As a result of constant stress changes, a fatigue and diminution of elasticity occurs compared as with the as manufactured belt.
Stress-Strain Curve	The elongation behaviour of a belt can be determined with the aid of a stress-strain curve. The test piece is subject to a pre-tension of 0.5% of the nominal tensile strength of the belt. It is then subjected to a test programm of 200 load cycles of between 2% and 10% of the nominal tensile strength of the belt.

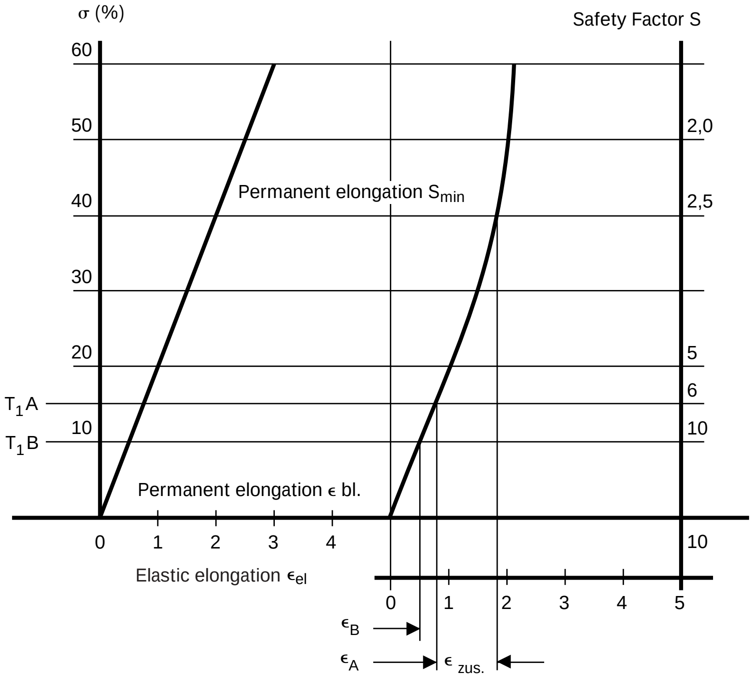
The following example is derived from the hysteresis loop thus obtained:

Initial Elongation	0.42%	Elastic Elongation	0.88%
Permanent Elongation	0.43%	Total Elongation	1.74%

The practical stress of the belt should be within these limits.



The quasi elastic area remaining above the 12% stress means an elongation reserve for temporary or local occurring over-stresses such as at start-up, braking, edge stresses at troughing transitions, terminal pulleys, belt turnover and pulley build-up etc.
In addition to this over elongation results in a slow destruction of the plies or carcase structure.



Stress-Strain curve for a belt with S_{min} = 2.5

T_{1A}	(N)	Belt tension at start-up
T_{1B}	(N)	Belt tension working
ϵ_B	(%)	Elongation in working condition
ϵ_A	(%)	Elongation at start-up
$\epsilon_{zus.}$	(%)	Additional elongation due to additional stresses
σ	(%)	Working stress/breaking stress

For a flat untroughed belt with uniform tension distribution over it’s cross section the belt safety can be calculated from the breaking or nominal tensile strength of the belt and the working tension at point x.

Belt Safety

$$S = \frac{k_N * B}{T_x} = \frac{\text{belt breaking strength}}{\text{working tension}}$$

In reality the belt is subject to the effects of temporary and local additional stresses in particular at the edge region and in the belt centre which are caused by such as:

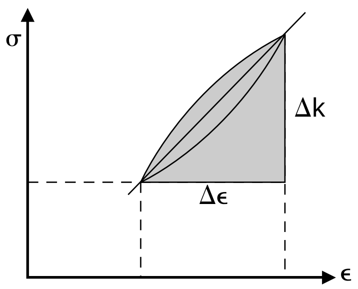
troughing transitions
convex vertical curves
concave vertical curves
horizontal curves
belt turnover

These additional stresses, positive or negative may give rise to
- **Overstraining** of belt edges and thereby a reduction of the belt safety or
- **Compression,** that is, negative elongation; the belt compresses resulting in arching or buckling.

Elastic Modulus E

With the help of the elastic modulus which can be determined from the **stress-strain** diagram and the following relationships, the conditions at the relevant points can be checked:

$$E = \frac{\sigma}{\epsilon} = \frac{T}{B * \epsilon} \qquad \frac{B}{T} = \frac{1}{E * \epsilon}$$



Hysteresis loop of a conveyor belt under initial tension

The belt safety appears thus:

$$S_x = \frac{k_N}{E * \epsilon} = \frac{k_N * B}{T_x} \qquad \epsilon_0 = \frac{T_x}{E * B} = \frac{k_N}{E * S_x} \quad (\%)$$

Belt safety Factor — Elongation at working tension

ϵ	(%)	Belt elongation
k_N	(-)	Nominal belt strength
k_D	(-)	Elongation value of the belt

Elongation ϵ_0 is the elongation under working tension at point x under the stress T_x or the safety S_x. If additional tensions or peak tensions and therefore additional elongations occur the elongation increases by the value $\Delta\epsilon_K$ at the belt edge that is to say $\Delta\epsilon_M$ in the centre of the belt. The total elongation results thus:

Belt edge

$$\epsilon_{ges} = \epsilon_0 + \Delta\epsilon_K = \frac{k_N}{E * S_{min}}$$

Belt centre

$$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_M \geq 0$$

For further calculations the elongation value $\mathbf{k_D = E / k_N}$ is introduced.

- With $\Delta\epsilon$ **the geometric relations** which result from belt tracking are determined.
- With S_x the **actual belt stresses** at point x apply.
- With S_{min} the **minimum permissible belt tension,** that is to say, the dynamic time-strength behaviour, is considered.
- With the elongation value k_D the **belt characteristics** that is to say, the elongation behaviour of the belt is taken into account.

Values for S_{min}

Carcase	Working Conditions	S_{min}
Textile belts	favourable normal unfavourable	3.2 3.5 3.7
ST-belts	favourable normal unfavourable	2.8 3.0 3.2

In practice the operation as a rule is determined by geographical or working conditions. These conditions decide the dimensions for the belt path.

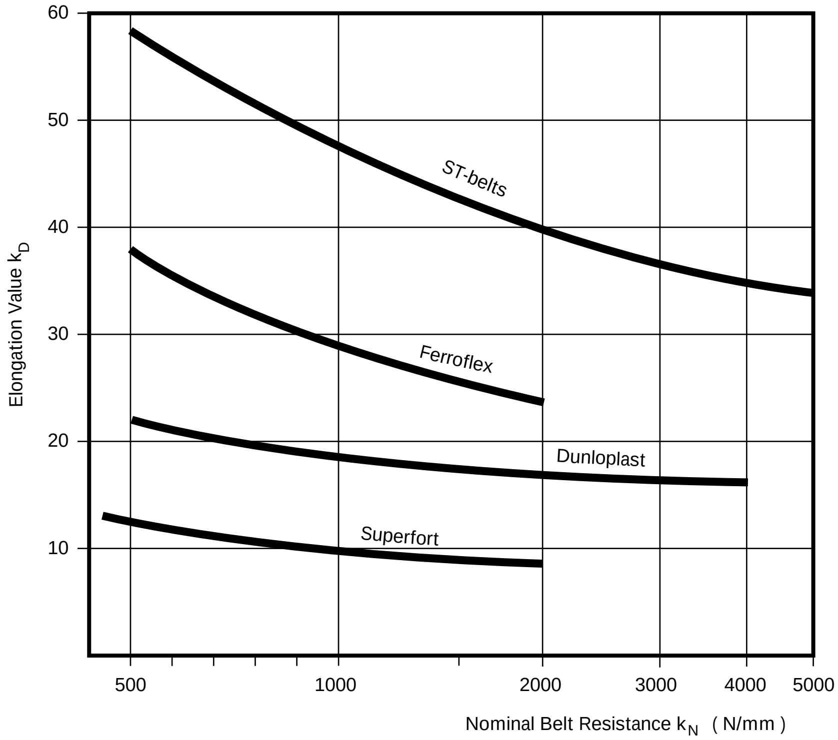
Edge Elongation

High belt elongations always occur if the belt is troughed or turned and the belt tension T_x at this point is especially high for instance at discharge with T_{max}.
Therefore a check of S_{min} is recommended.

Buckling

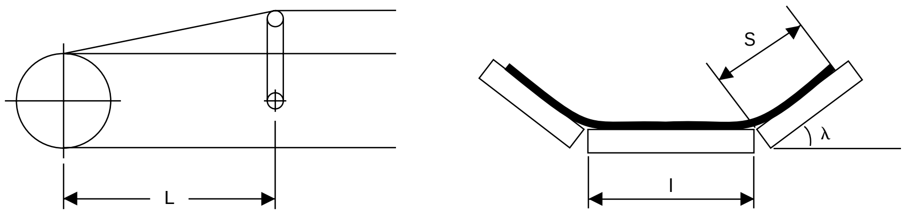
Buckling occurs mostly at those points where the belt tension T_x is particularly low for instance at the tail when running empty.
A check of ϵ_{min} is recommended.

Elongation Value k_D



The elongation values of the various Dunlop belt types may be taken from the above diagram. They are values for the planning of an installation. In individual cases the k_D value of a certain belt type can differ from the values in the diagram. In separate cases it is recommended that the advice of our engineers be sought.

Troughing Transition



Belt Edges

The belt edges are examined to maintain S_{min}, that is to say actual belt tensions present at discharge and tail.

Safety Factor

$$S_{min} = \frac{1}{k_D * \epsilon_{ges}}$$

permissible values see Appendix I.2

Total Elongation

$$\epsilon_{ges} = \epsilon_0 + \Delta\epsilon_K$$

Average Elongation

$$\epsilon_0 = 1 / (k_D * S_B)$$

Additional Elongation

$$\Delta\epsilon_K = \frac{s^2 * \hat{\lambda}^2}{L^2} * \left(\frac{1}{2} - \frac{s}{3 * B} \right)$$

k_D	(-)	Elongation value of belt
l	(mm)	Length of centre carrying idler
s	(mm)	Lenght of belt in contact with the side carrying idler roller
B	(mm)	Belt width
$\hat{\lambda}$	(-)	Arc measurement $\hat{\lambda} = \pi * \lambda° / 180$
L	(mm)	Transition Length
S_B	(-)	Safety factor at discharge with T_1 at tail with T_4

Belt Centre

The belt centre is checked to determine whether the **total elongation $\epsilon > 0$** that is no buckling of the belt is expected.

Total Elongation

$$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_M \geq 0$$

Average Elongation

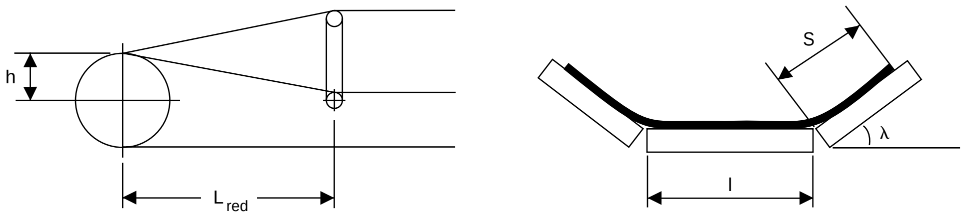
$$\epsilon_0 = 1 / (k_D * S_{min})$$

Additional Elongation

$$\Delta\epsilon_M = \frac{s^3 * \hat{\lambda}^2}{3 * B * L^2}$$

S_{min} (-) Safety factor with the lowest belt tension at discharge and tail with T_{empty}.

Troughing Transition with Raised Pulley



Belt Edge

Check of Minimum Safety S_{min}.

Safety Factor

$$S_{min} = \frac{1}{k_D * \epsilon_{ges}}$$

Permissible values see Appendix I.2

Total Elongation

$$\epsilon_{ges} = \epsilon_0 + \Delta\epsilon_K$$

Average Elongation

$$\epsilon_0 = 1 / (k_D * S_B)$$

Additional Elongation

$$\Delta\epsilon_K = \epsilon_1 - \epsilon_2$$

$$\epsilon_1 = \frac{s^2 * \hat{\lambda}^2}{L^2} * \left(\frac{1}{2} - \frac{s}{3 * B} \right)$$

$$\epsilon_2 = \frac{h * B * \sin\lambda}{4 * L_{red}^2} \left(1 - \frac{l^2}{B^2} \right)$$

h (mm) lift of pulley
L_{red} (mm) reduced transition length

Centre of Belt

Check of Elongation i.e. Possible Buckling.

Total Elongation

$$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_M \geq 0$$

Average Elongation

$$\epsilon_0 = 1 / (k_D * S_{min})$$

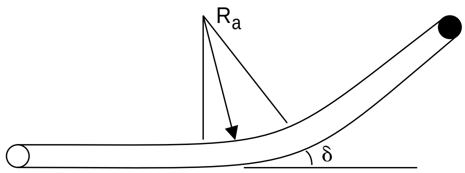
Additional Elongation

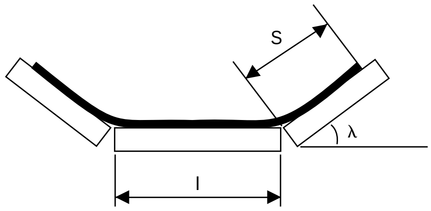
$$\Delta\epsilon_M = \epsilon_1 - \epsilon_2$$

$$\epsilon_1 = \frac{s^2 * \hat{\lambda}^2}{L^2} * \left(\frac{1}{2} - \frac{s}{3 * B} \right)$$

$$\epsilon_2 = \frac{1}{8} * \frac{h * B * \sin\lambda}{L_{red}^2} \left(1 - \frac{l}{B} \right)^2$$

Concave Vertical Curve





Belt Edges

Check on Buckling Risk.

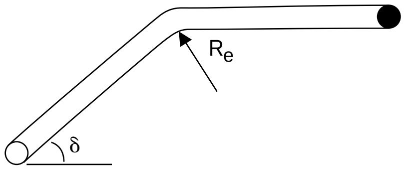
Total Elongation	$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_K \geq 0$
Average Elongation	$\epsilon_0 = 1 / (k_D * S_{empty})$
Additional Elongation	$\Delta\epsilon_K = \frac{s * \sin\lambda}{R_a} * (1 - \frac{s}{B})$

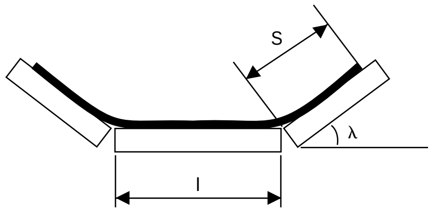
Belt Centre

Check on Minimum Safety S_{min}.

Total Elongation	$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_M$	
Average Elongation	$\epsilon_0 = 1 / (k_D * S_B)$	
Additional Elongation	$\Delta\epsilon_M = * \frac{s^2}{B} \quad \frac{\sin\lambda}{R_a}$	
Safety Factor	$S_{min} = \frac{1}{k_D * \epsilon_{ges}}$	Permissible values see Appendix 1.2

Convex Vertical Curve





Belt Edge

Check on Minimum Safety S_{min}.

Safety Factor

$$S_{min} = \frac{1}{k_D * \epsilon_{ges}}$$

Permissible values see Appendix I2

Total Elongation

$$\epsilon_{ges} = \epsilon_0 + \Delta\epsilon_K$$

Average Elongation

$$\epsilon_0 = 1 / (k_D * S_B)$$

Additional Elongation

$$\Delta\epsilon_K = \frac{s * \sin\lambda}{R_e} * \left(1 - \frac{s}{B} \right)$$

Belt Centre

Check on Buckling Risk.

Total Elongation

$$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_M \geq 0$$

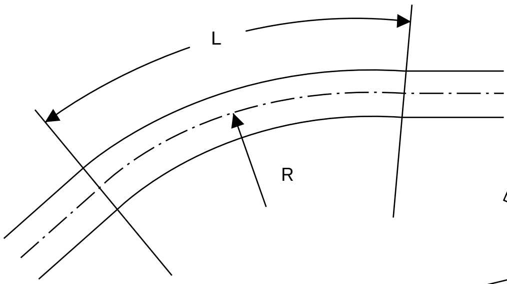
Average Elongation

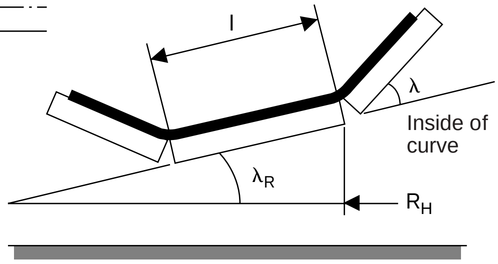
$$\epsilon_0 = 1 / (k_D * S_{empty})$$

Additional Elongation

$$\Delta\epsilon_M = \frac{s^2}{B * R_e} * \sin\lambda$$

Horizontal Curve





Belt Edge
Outer Radius

Check on Minimum Safety S_{min}.

Safety Factor

$$S_{min} = \frac{1}{k_D * \epsilon_{ges}}$$

Permissible values see Appendix I2

Total Elongation

$$\epsilon_{ges} = \epsilon_0 \;\; \Delta\epsilon_K$$

Average Elongation

$$\epsilon_0 = 1 / (k_D * S_B)$$

Additional Elongation

$$\Delta\epsilon_K = \frac{l}{2R} + \cos\lambda_R + (\frac{B-1}{2R}) \cos(\lambda + \lambda_R)$$

Belt Edge
Inner Radius

Check on Buckling Risk.

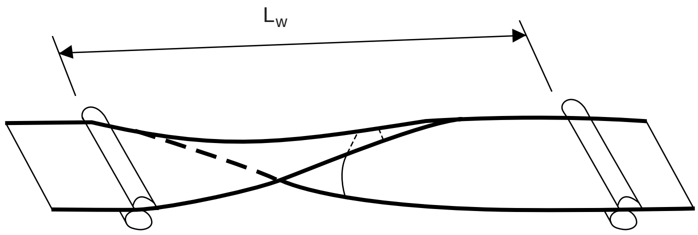
Total Elongation

$$\epsilon_{ges} = \epsilon_0 - \Delta\epsilon_K \geq 0$$

Average Elongation

$$\epsilon_0 = \frac{1}{k_D * S_{empty}}$$

Belt Turnover



T_x	(N)	Belt Tension at region of turnover T_x = ca. T_2 at discharge T_x = ca. T_3 at tail
S_x	(-)	Belt safety with belt tension T_x
L_w	(m)	Length of turnover section
k_N	(N/mm)	Nominal strength of the belt

Belt Edges

Control of minimum safety S_{min}.

Safety Factor	$S_{min} = \dfrac{1}{k_D * \epsilon_{ges}}$
Total Elongation	$\epsilon_{ges} = \epsilon_x + \Delta\epsilon_K$
Elongation under Working Tension	$\epsilon_x = \dfrac{1}{k_D * S_x}$
Safety Factor at Working Stress	$S_x = \dfrac{k_N * B}{T_x}$
Additional Elongation	$\Delta\epsilon_x = c * \dfrac{B^2}{L_w{}^2}$

Turnover Section

The length of the turnover section can be calculated once S_x and S_{min} are determined.

$$L_W = c * B * \sqrt{k_D * \frac{S_x * S_{min}}{S_x - S_{min}}} \quad (\text{ m })$$

B	(m)	Belt Width
c	(-)	Factor c (see table)

Factor c	Turnover Guided	Turnover Supported
Textile Belts	1.55	1.36
ST. Belts	1.36	1.13

Belt Centre

Minimum Turnover length when buckled.

$$L_W = \frac{c * B}{1.4} * \sqrt{S_x * \frac{k_D}{c_V}} \quad (\text{ m })$$

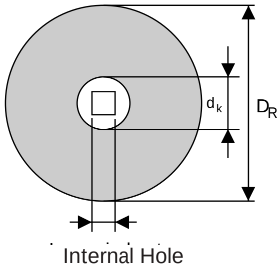
c_V (-) Loss of breaking strength at the joint.

For transport purposes, conveyor belts are coiled on wooden cores or drums. The length or weight of the belt determines the size of core to use.

Core Diameter d_k (mm)	Internal Square Hole (mm)	Application
150 250	55 110	Stock belts, general
400	205	Wide and heavy belts
600	205	Steel cord belts

The roll diameter can be calculated using the following formulae:

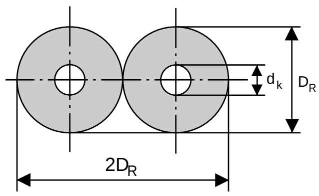
Coiled Roll



$$D_R = \sqrt{1.27 * d * L + d_k^2} \quad (m)$$

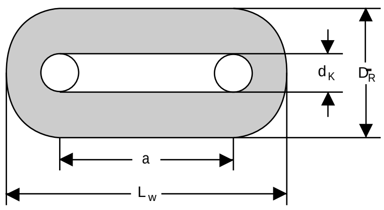
D_R	(m)	roll diameter
d	(m)	belt thickness
d_k	(m)	core diameter
L	(m)	belt length

Double Coiled Roll



$$D_R = \sqrt{1.27 * d * \frac{L}{2} + d_k^2} \quad (m)$$

Roll on Oval Core

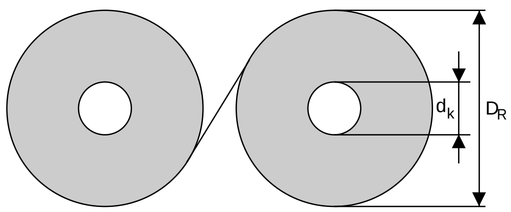


$$a = \frac{d * L}{D_R - d_k} \quad \frac{\pi}{4} * (D_R + d_k) \quad (m)$$

$$L_W = a + D_k \quad (m)$$

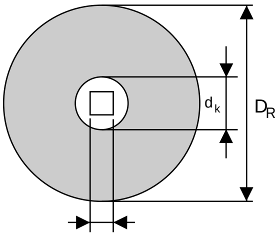
$$D_R = \sqrt{1.27 * d * L + (d_k + \frac{2}{\pi} * a)^2 - \frac{2}{\pi} * a} \quad (m)$$

S-Form Coiled Roll



$$L = \frac{\pi\ (D_R^2 - d_k^2)}{2 * d} \quad (m)$$

Roll Diameter
in m

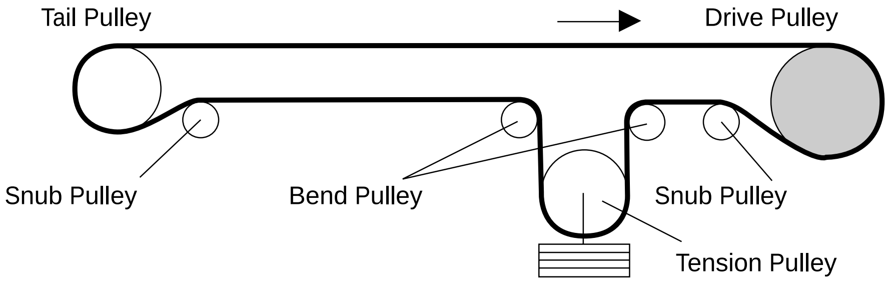


Internal Hole

L (m)	Belt Thickness (mm) 5	6	7	8	9	10	11	12
20	0.44	0.46	0.49	0.52	0.54	0.56	0.58	0.61
30	0.50	0.54	0.57	0.61	0.64	0.67	0.69	0.72
40	0.56	0.61	0.65	0.68	0.72	0.76	0.79	0.82
50	0.62	0.67	0.71	0.76	0.80	0.84	0.87	0.91
60	0.67	0.72	0.77	0.82	0.87	0.91	0.95	0.99
70	0.71	0.77	0.83	0.88	0.93	0.98	1.02	1.06
80	0.76	0.82	0.88	0.94	0.99	1.04	1.09	1.13
90	0.80	0.87	0.93	0.99	1.05	1.10	1.15	1.20
100	0.84	0.91	0.98	1.04	1.10	1.15	1.21	1.26
120	0.91	0.99	1.06	1.14	1.20	1.26	1.32	1.38
140	0.98	1.06	1.14	1.22	1.29	1.36	1.42	1.48
160	1.04	1.13	1.22	1.30	1.38	1.45	1.52	1.58
180	1.10	1.20	1.29	1.38	1.45	1.53	1.61	1.68
200	1.16	1.26	1.36	1.45	1.53	1.61	1.69	1.76
220	1.21	1.32	1.42	1.52	1.61	1.69	1.77	1.85
240	1.26	1.38	1.48	1.58	1.68	1.76	1.85	1.93
260	1.31	1.43	1.54	1.64	1.74	1.83	1.92	2.00
280	1.36	1.48	1.60	1.71	1.81	1.90	1.99	2.08
300	1.40	1.53	1.65	1.76	1.87	1.97	2.06	2.15
320	1.45	1.58	1.71	1.82	1.93	2.03	2.13	2.22
340	1.49	1.63	1.76	1.88	1.99	2.09	2.19	2.29
360	1.53	1.68	1.81	1.93	2.05	2.15	2.26	2.36
380	1.57	1.72	1.85	1.98	2.10	2.22	2.32	2.42
400	1.61	1.77	1.90	2.03	2.15	2.27	2.38	2.48

Core diameter d_k = 250 mm

The pulley diameter in general depends upon the thickness of the carcase (see Appendix C). In regions of low tension such as at the tail pulley or with small angles of wrap such as snub pulleys (90°) smaller pulley diameters may be used.



Tension Pulley (take-up) see Tail Pulley

Pulley diameters
for belt types

SUPERFORT S
STARFLEX SF
DUNLOFLEX D
TRIOFLEX T
FERROFLEX F

Belt Type	Pulley Diameter (mm) A	B	C
S 200/3	250	200	160
S 250/3	250	200	160
S 315/3	315	250	200
S 315/4	400	315	250
S 400/3	315	250	200
S 400/4	400	315	250
S 500/3	400	315	250
S 500/4	400	315	250
S 630/3	400	315	250
S 630/4	500	400	250
S 630/5	630	500	315
S 800/3	500	400	400
S 800/4	630	500	315
S 800/5	630	500	400
S 1000/4	630	500	400
S 1000/5	800	630	400
S 1000/6	800	630	500
S 1250/4	800	630	500
S 1250/5	800	630	500
S 1250/6	1000	800	630
S 1600/4	1000	800	630
S 1600/5	1000	800	630
S 1600/6	1000	800	630
S 2000/5	1200	1000	800
S 2500/6	1400	1200	1000

Belt Type	Pulley Diameter (mm) A	B	C
SF 250/2	200	160	125
SF 315/2	250	200	160
SF 400/3	250	200	160
SF 500/3	315	250	200
SF 500/4	400	315	250
SF 630/4	500	400	315
SF 800/4	500	400	315
SF 1000/4	630	500	400
D 160	250	200	160
D 200	250	200	160
D 250	250	200	160
D 315	250	200	160
D 400	315	250	200
D 500	315	250	200
D 630	400	315	250
D 800	500	400	315
T 315	315	250	200
T 400	400	315	250
T 500	500	400	315
T 630	630	500	400
T 800	800	630	500
T 1250	1000	800	630
F 500	500	400	315
F 630	500	400	315
F 800	630	500	400
F 1000	630	500	400
F 1250	800	630	400
F 1600	800	630	400

CHEVRON Belts

Belt Type de bande	Profile	Drive Pulley (mm)	Bend Pulley (mm)	Snub Pulley (mm)
S 200/3	C 330/16	250	250	160
	C 390/15Z	250	250	160
	C 430/16	250	250	160
	C 530/16	250	250	160
S 400/3	C 650/16	315	250	200
	C 800/16	315	250	200
S 500/4	C 1000/16	500	400	250

HIGH CHEVRON Belts

Belt Type de bande	Profile	Drive Pulley (mm)	Bend Pulley (mm)	Snub Pulley (mm)
S 200/3	HC 450/32	315	315	200
S 400/3	HC 450/32	315	315	200
	HC 600/32	315	315	200
S 500/4	HC 800/32	500	400	250
	HC 1000/32	500	400	250
S 630/4	HC 1200/32	500	400	315

DUNLOPLAST DLP

Belt Type	Cover Thickness* (mm)	Total Thickness (mm)	Weight (kg/m^2)	Pulley minimum Diameter A (mm)	B (mm)	C (mm)
PVC						
DLP 250 PVC FDA	2 + 0	3	3.75	100	80	60
DLP 250 PVC FDA	2 + 1	4	5.00	100	80	60
DLP 315 PVC FDA	1.5 + 1.5	4	5.00	200	160	125
DLP 400 PVC FDA	1.5 + 1.5	5	6.25	250	200	160
DLP 630 PVC FDA	1.5 + 1.5	7	8.75	400	315	250
DLP 800 PVC FDA	2 + 2	9	11.25	500	400	315
DLP 1000 PVC FDA	2 + 2	11	13.75	500	400	315
DLP 1600 PVC FDA	3 + 3	14	17.50	630	500	400
DLP 2000 PVC FDA	3 + 3	15	18.75	800	630	500
DLP 2500 PVC FDA	3 + 3	16	20.00	1000	800	630
DLP 250 PVC BLS	2 + 1	4	5.00	160	125	100
DLP 315 PVC BLS	3 + 1.5	6	7.50	250	200	160
DLP 400 PVC BLS	4 + 2	8	10.00	315	250	200
DLP 630 PVC BLS	5 + 3	11	13.75	500	400	315
DLP 1000 PVC BLS	2 + 2	11	13.75	500	400	315
DLP 315 PVC A	3 + 1.5	6	7.50	250	200	160
DLP 400 PVC A	4 + 2	8	10.00	315	250	200
DLP 630 PVC A	5 + 3	11	13.75	500	400	315
DLP 1000 PVC A	5 + 3	15	18.75	630	500	400
DLP 1600 PVC A	5 + 3	17	21.75	800	630	500
RUBBER						
DLP 630 RS	6 + 3	13	16.25	400	315	250
DLP 800 RS	8 + 3	16	20.00	500	400	315
DLP 1000 RS	8 + 3	18	21.00	630	500	400
DLP 1250 RS	10 + 4	21	26.25	800	630	500
DLP 1600 RS	10 + 4	23	27.00	800	630	500
DLP 2000 RS	10 + 4	25	29.00	1000	800	630
DLP 2500 RS	10 + 4	26	31.00	1000	800	630
DLP 630 DB RS	15 + 5	27	31.00	500	400	315

* Total cover thickness (including reinforcing fibre)

PVC-Quality

FDA	Colour: white. For transport of food stuffs, meeting International recommendations. Resistant to Fats, Oils and Solvents. Temperature range: -15° to +80°C
BLS	Colour: white. Food quality and flame resistant according to International recommendations. Temperature range: -15° to +80°C
A	Colour: grey. Non-toxic, highly abrasion resistant, Chemical, Fat and oil. Temperature range: -20° to +80°

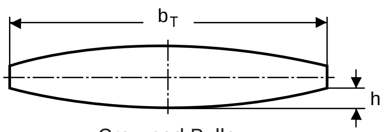
Rubber-Quality

RA	Highly abrasion resistant under normal working conditions for bulk and unit loads.

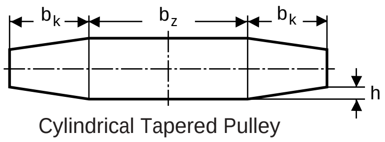
Steel Cord Belts
SILVERCORD

Belt Type	Min. Cover Thickness Top (mm)	Min. Cover Thickness Bottom (mm)	Belt Thick-ness (mm)	Belt Weight (kg/m^2)	Minimum Pulley Diam. A (mm)	Minimum Pulley Diam. B (mm)	Minimum Pulley Diam. C (mm)
ST 500	5	5	12.5	16.40	630	500	400
ST 630	5	5	12.5	16.92	630	500	400
ST 800	5	5	13.2	18.37	630	500	400
ST 900	5	5	13.8	19.39	630	500	400
ST 1000	5	5	13.8	19.79	630	500	400
ST 1150	5	5	13.9	20.45	630	500	400
ST 1250	5	5	13.9	20.98	630	500	400
ST 1400	5	5	14.7	22.16	800	630	500
ST 1600	5	5	15.3	23.48	800	630	500
ST 1800	5	5	15.3	24.01	800	630	500
ST 2000	5	5	15.3	24.63	800	630	500
ST 2250	5	5	16.4	27.98	1000	800	630
ST 2500	5	5	16.8	29.38	1000	800	630
ST 2750	6	6	19.0	32.29	1250	800	630
ST 3150	6	6	19.6	34.81	1250	800	630
ST 3500	7	7	22.0	39.05	1400	1000	630
ST 3750	7	7	22.4	41.14	1400	1000	630
ST 4000	7	7	22.8	42.61	1400	1000	630
ST 4250	8	8	25.6	46.51	1400	1250	800
ST 4500	8	8	26.2	48.40	1600	1400	800
ST 4750	8	8	26.6	50.05	1600	1400	1000
ST 5000	9	9	29.0	54.08	1800	1600	1000
ST 6000	9	9	30.4	60.69	2000	1800	1250
ST 7000	10	10	33.3	68.50	2250	2000	1600

Additional weight for each 1mm additional cover thickness: 1.12 kg/m^2.
Belt thickness and belt weight apply to belt with the minimum cover thickness top and bottom.

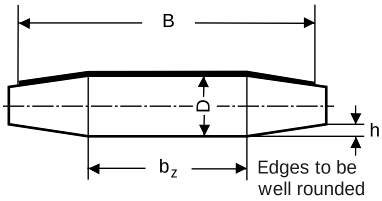


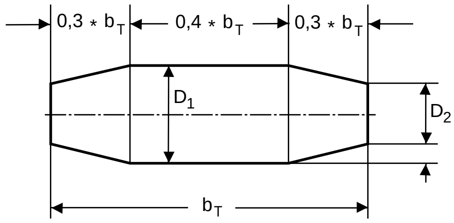
Crowned Pulley



Cylindrical Tapered Pulley

The tracking of conveyor belts will in practice be effectively achieved using crowned pulleys. In most cases for reasons of price and manufacture, the pulley is shaped in a cylindrical-tapered manner.
Pulley crown should be carefully dimensioned to ensure belt life, wear and tear and working behaviour are not influenced determinately.





B (mm) Belt width
D (mm) Pulley diameter
h (mm) Taper
b_z (mm) Width of cylindrical section

$$f = \frac{D_1 - D_2}{2} \qquad f < 0.005 * \frac{D_1}{2}$$

Taper to ISO/DIN 5286

Cylindrical Mid Section b_z

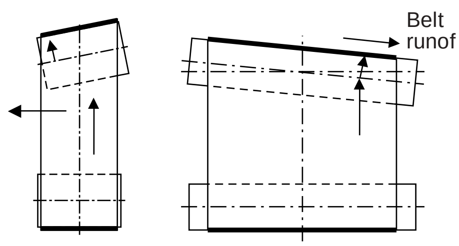
Belt Width B (mm)	b_z (mm)	B/b_z
> 300	0.4 * B	2.5
> 300 à 800	0.5 * B	2
> 800	0.65 * B	1.5

Taper h

Pulley Diameter (mm)	Taper h (mm)		
	L < 3000 mm	L > 3000 mm	
		B ≤ 800 mm	B > 800 mm
< 200	0.5	0.5	0.7
> 200 à 400	1	1.25	1.5
> 400 à 800	1.5	2	2.5
> 800	2	2.5	3

Notes

- If the taper h is too great or if b_z is too small differential belt tensions result which has an adverse effect on belt life.
- If the belt in the taper area does not make full contact this will lead to wear and tear on the pulley side of the belt.
- If the taper h is too great diagonal creases can appear in the belt. The tracking effect is reduced.
- The transition from the cylindrical to the tapered part of the pulley must be well rounded.
- Crowned pulleys can also be lagged.



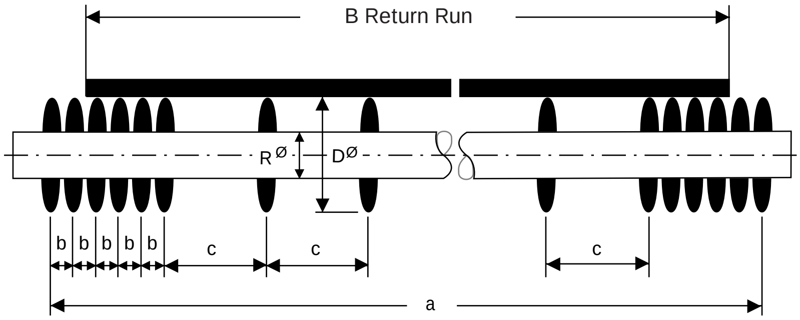
Empirical determinations:

- The belt runs towards the slack side.
- The belt wanders towards the side to which it first makes contact with the pulley surface.

Supporting Discs on Return Side Run

Rubber supporting discs on the return side run will behave as a belt surface cleaner. These idlers sometimes influence the tracking of the belt in a negative manner.

Installation Instruction



Measurements (Values)

Belt Width B (mm)	a (mm)	b (mm)	c (ca. mm)	Number of Discs each roller end	Number of Discs total	Tube Diam. R (mm)	Disc Diam. D (mm)
400	425	25	108	3	8	51	120
500	530	30	137	3	8	57	133
650	680	30	140	3	9	57	133
800	890	30	130	5	14	63.5	150
1000	1090	30	122	5	16	63.5	150
1200	1305	35	128	5	17	88.9	180
1400	1505	35	136	5	18	88.9	180
1600	1800	40	140	6	21	108	180 215
1800	2000	40	145	6	22	108	180 215
2000	2200	40	156	7	24	108	180 215

The conveyor belt must not only meet the mechanical and thermal demands of the load but also be resistant to diverse chemicals. The chemicals can be in liquid, solid form or gaseous form. In many cases the quantity, concentration and temperature of the substances being handled is not exactly known or defined.

The chemical resistance of the belt will be judged by the swelling of the covers. At ca. 8-10% swelling the physical properties decline, the belt loses its working capability and its life will be reduced.

The reaction will intensify through higher concentration, also with relatively low temperature, or with lower concentration and higher temperature.

Resistance of the Carcase

Chemical	Carcase Material			
	B	P	E	D
Acetic Acid 10%	+	0	+	+
Acetone	+	+	+	+
Ammonia 10%	+	0	+	+
Benzine	+	+	+	+
Benzol	+	+	+	+
Creosote	+	–	+	+
Diesel Oil	+	+	+	+
Drilling Oil	+	+	+	+
Ethylacetate	0	0	+	+
Formaldehyde 10%	+	0	+	+
Formic Acid 10%	+	0	+	+
Glycerine	+	+	+	+
Hydrochloric Acid 10%	–	–	+	+
Latic Acid 10%	+	0	+	+
Machine Oil	+	+	+	+
Methylalcohol	0	–	+	+
Oil, Heavy Swelling	+	+	+	+
Oil, Light Swelling	+	+	+	+
Phenol 10%	+	–	+	+
Pine Tar	+	+	+	+
Potassium Bichromate 10%	+	0	+	+
Sulphuric Acid 10%	–	–	+	+
Trichloroethylene	+	+	+	+
Urea 50%	+	0	+	+
Water 20°C	+	+	+	+
Water 80°C	+	+	+	+

+ = resistant   0 = limited resistance   – = not resistant

B Cotton
P Polyamide
E Polyester
D Aramid

Resistance of the Covers

Chemical	Condition K %	Condition °C	Basic Material NR SBR	Basic Material NBR NCR	Basic Material Butyl	Basic Material PVC
Acetic Acid	100	20	–	–	0	–
Acetic Acid	10	20	+	–	+	0
Acetic Acid	10	80	–	–	–	–
Acetone	100	20	+	–	+	–
Ammonia	10	20	+	+	+	+
Aniline	100	20	0	–	+	–
Axle Oil	100	80	–	0	–	0
Axle Oil	100	20	–	+	–	+
Benzine	100	20	–	0	–	+
Benzine/Benzol 1:1	50/50	20	–	–	–	+
Benzol	100	20	–	–	–	–
Caustic Soda	10	20	+	+	+	+
Caustic Soda	40	20	0	0	+	–
Caustic Soda	40	80	0	0	+	–
Caustic Soda	10	80	0	0	+	–
Chloroform	100	20	–	–	–	–
Copper Sulphate	10	20	0	0	+	+
Creosote	50	20	0	0	+	–
Cyclohexane	100	20	–	–	–	–
Detergent P3	3	20	+	+	+	+
Detergent P3	3	80	+	0	+	0
Diesel Oil	100	20	–	0	–	+
Drilling Oil	2	20	0	+	0	+
Drilling Oil	10	20	–	+	–	+
Drilling Oil	100	20	–	+	–	+
Ethylacetate	100	20	0	–	0	–
Ethylalcohol	100	20	+	0	+	0
Formaldehyde	38	20	+	0	+	+
Formic Acid	10	20	+	–	+	+
Formic Acid	100	20	0	–	+	0
Glycerine	100	20	+	+	+	+
Glycerine	100	80	+	+	+	+
Hydrochloric Acid	10	80	–	–	–	–
Hydrochloric Acid	10	20	+	+	+	+
Hydrochloric Acid	30	80	–	–	–	–
Hydrochloric Acid	30	30	+	+	+	+
Latic Acid	10	80	–	–	0	0
Latic Acid	10	20	+	0	+	+
Machine Oil	100	20	–	+	0	+
Machine Oil	100	80	–	0	–	0
Magnesium Sulphate	10	80	+	+	+	+
Magnesium Sulphate	10	20	+	+	+	+
Maleic Acid	10	80	–	0	–	–
Maleic Acid	10	20	+	0	+	0
Methylalcohol	100	20	+	0	+	0
Methylene Chloride	100	20	–	–	–	–
Nitro Solvent	100	20	0	–	0	–
Oil Heavy Swelling	100	20	–	+	–	+
Oil Heavy Swelling	100	80	–	0	–	–
Oil Swelling	100	80	–	0	–	–
Oil Swelling	100	20	0	+	–	+
Olive Oil	100	80	–	0	–	0
Olive Oil	100	20	–	+	+	+
Oxalic Acid	10	80	–	–	+	+
Oxalic Acid	10	20	+	+	+	+

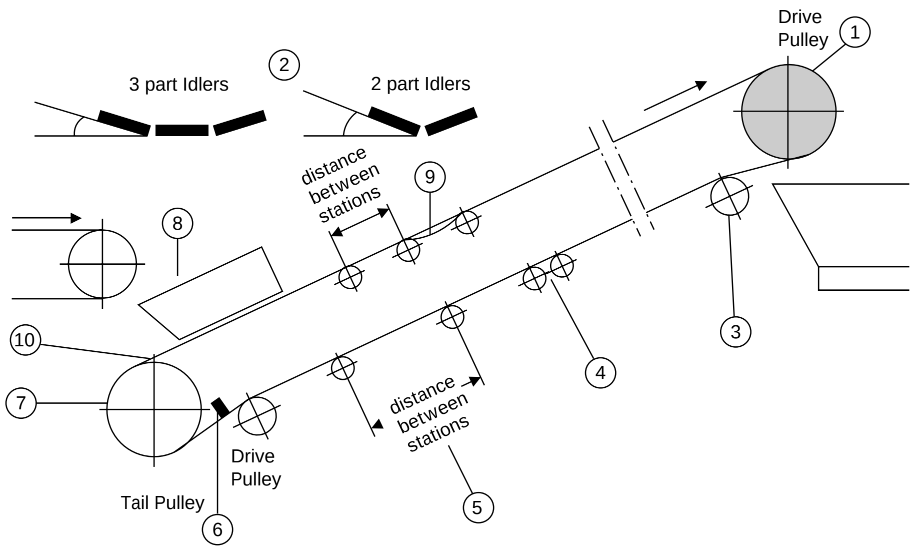
Chemical	Condition K %	°C	Basic Material NR SBR	NBR NCR	Butyl	PVC
Paper Pulp (damp)	Pulp	20	+	–	0	–
Paraffin Oil	100	80	–	0	–	0
Paraffin Oil	100	20	–	+	–	+
Phenol	10	80	–	–	–	–
Phenol	10	20	0	–	+	–
Phosphoric Acid	10	20	+	+	+	+
Phosphoric Acid	50	20	+	+	+	+
Phosphoric Acid	10	80	+	–	+	0
Pitch for Wood Impregnation	100	20	–	–	–	–
Potash	40	20	0	0	+	–
Potash	10	20	+	+	+	+
Potash	10	80	0	0	+	–
Potash	40	80	0	0	+	–
Potassium Dichromate	10	20	+	+	+	+
Potassium Dichromate	10	80	0	+	+	+
Potassium Permanganate	10	20	–	–	–	–
Sea Water	100	80	+	+	+	+
Sea Water	100	20	+	+	+	+
Stearic Acid Powder	100	20	+	+	+	+
Stearic Acid Powder	100	80	–	0	–	0
Sulphuric Acid	50	20	–	–	–	–
Sulphuric Acid	10	20	+	+	+	+
Sulphuric Acid	50	80	–	–	–	–
Sulphuric Acid	10	80	–	–	+	–
Tallow	100	20	0	+	+	+
Tallow	100	80	–	–	–	–
Tetralin	100	20	–	–	–	–
Toluol	100	20	–	–	–	–
Trichloroethylene	100	20	–	–	–	–
Urea	50	20	+	+	+	+
Water (tap)	100	20	+	+	+	+
Water (tap)	100	80	+	+	+	+

K % Concentration in %

NR Natural Rubber
SBR Styrene Butadiene Rubber
NBR Nitrile Rubber
NCR Nitrile Chloroprene Rubber
Butyl Butyl Rubber
PVC Polyvinyl Chloride

Hints for Installation Improvements

To obtain optimum performance from steep incline belts in the CHEVRON and HIGH-CHEVRON range with regard to smooth running and working safety, a range of constructive measures are recommended which have proved themselves in practice.



1. Drive Pulley — The drive pulley should be rubber lagged. The belt tension will be less, slip at start-up under load, prevented, and the running of the belt stabilized.

2. Troughing Angle — If 3 part idlers are used, the troughing angle should not be greater than 20° and with 2 part idlers, greater than 25° (for jute sacks etc.)

3. Snub Pulley — The angle of wrap on the drive pulley should not be more than ca. 190°. The pressure on the profile caused by the snub pulley would otherwise be too high such that the profile can be pressed through to the pulley side of the belt. Changes in the direction of bending of the belt are not recommended.

4. Return Side — To improve the running of the return side belt, two idlers with reduced diameter and set next to one another, can be fitted.

5. Idler Spacing — On the carrying and return side run, as with smooth surface belts, standard idlers are used. The idler pitch should not be divisible by the pitch of the profile pattern on the belt.

6. Deflector — A plough scraper should be fitted immediately before the tail pulley so that material does not get trapped between belt and pulley. Where there is back flowing water on the pulley side of the return belt, a water scraper (plough scraper) should be used.

7. Tail Pulley — If wet materials are conveyed, water shedding grooved pulley lagging is recommended because of the risk of off-track running. The tail pulley should as a minimum be one profile length behind the back of the loading chute so that back flowing material can be re-admitted.

8. Loading Height — The height of fall of material being loaded to the belt should be kept as small as possible, the loading being in direction of belt travel.

9. Belt Sag — The belt must not sag excessively between idlers (pre-tension), otherwise the load stream becomes broken up causing loss of conveying capacity and an increase in power demand.

10. Cleaning

Steep incline belting cannot be scraped by normal mechanical cleaning devices. Clinging and sticky materials will result in build-ups. If conditions permit, the belt surface beyond the discharge pulley can be hosed with water or cleaned by compressed air. For lumpy and dry materials, sledge-hammer devices can be used which impact the pulley side of the belt during rotation. Eccentric rapping rollers that vibrate the belt can also be fitted.

Methods of Making Endless

Conveyor belting can be made endless using the following jointing methods:

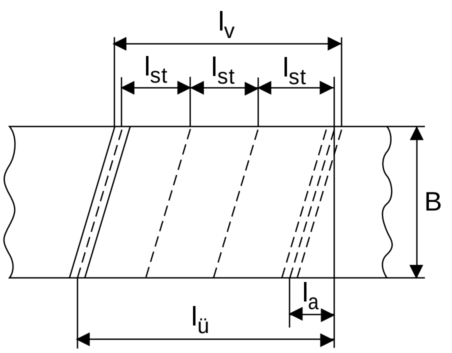
- Hot Vulcanised
- Warm Vulcanised
- Cold Vulcanised
- Mechanical Fastener

The durability obtained from these various methods depends on environmental conditions, splicing materials and the proficiency of the splicing personnel. The advice of the belt manufacture should be observed particularly the special recommendations that apply to certain belt types and qualities.

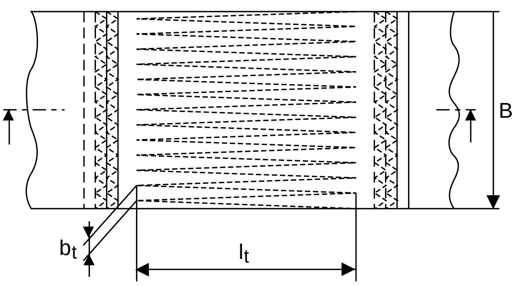
Hot Vulcanisation

With hot vulcanising, vulcanising presses are necessary. The vulcanisation takes place under a certain pressure and depending on cover quality, at a certain time and temperature. In general the vulcanising temperature lies between 140°C and 155°C. Vulcanisation times depend on the belt thickness and are of 20 to 45 minutes duration. With almost all belt types, the best results are achieved by the hot vulcanisation process with regard to strength and the retention of strength after prolonged periods of usage.

With **textile belts,** depending on carcase construction there are the stepped splice method (1 or more steps) and the finger splice method.

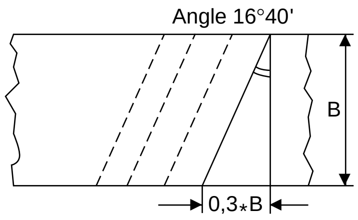


Stepped Splice

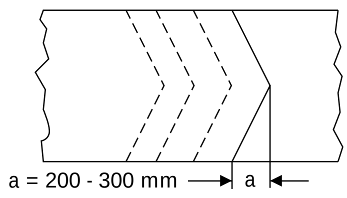


Finger Splice

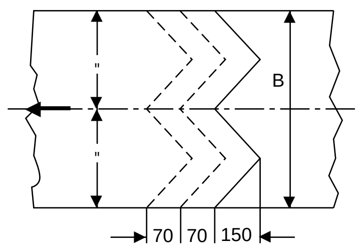
Depending on the **application** one differentiates between the straight joint (90° across the belt) the diagonal splice, the Chevron splice and the zig-zag splice (for short centre to centre distances or for weigher belts).



Diagonal Splice

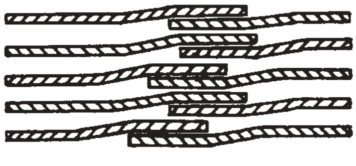


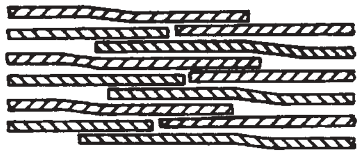
Chevron Splice

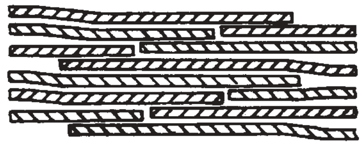


Zig-Zag Splice

For **Steel Cord Belts** depending on tensile strength and the cord diameter the 1 to 4 step splice is used. With this the cords are laid together in a particular design.


1 Step


2 Step


3 Step

Warm Vulcanisation

For the warm vulcanisation method, a press is likewise required. The joint can however be constructed with shorter overlap lengths. The vulcanisation temperature lies between 100°C to ca. 115°C. For the warm vulcanising process special vulcanising materials are required.

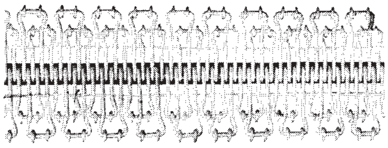
Cold Vulcanisation

With cold vulcanisation, the process takes place under ambient temperature conditions. The optimum strength is reached in ca. 24 hours. The quality of the joint is very dependent upon the ambient conditions, humidity, dustiness etc. Expenditure on equipment and time is much reduced.

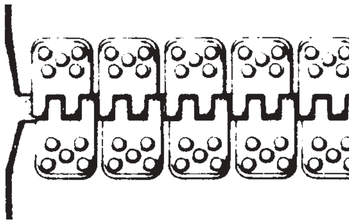
Mechanical Splice

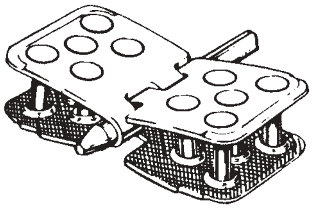
The quickest and most cost effective joint is the mechanical splice. The strength however is significantly lower than that of a vulcanised splice joint.

The mechanical fastener is well suited for quick emergency splicing or belt damage repair.

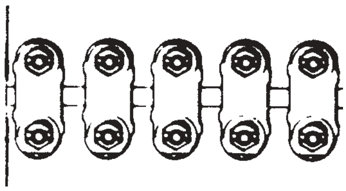


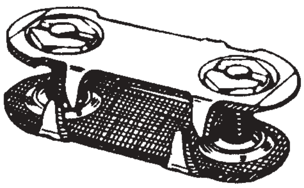
Steel Wire Hook Fastener





Hinged Fastener





Bolted Plate Fastener (Flexco)

Material	Bulk Density ρ (t/m³)	Surcharge Angle β (°)	Maximum Angle of Inclination Smooth Belts	height of profiles 6 mm	16 mm	32 mm	Recom-mended Dunlop Quality
Alum	0.80 - 1.04	25	17	-	-	-	-
Aluminium, coarse	0.95 - 1.05	15	20	-	-	-	-
Aluminium, fine	0.70 - 0.80	6	20	-	25	30	-
Aluminium - Oxide	1.12 - 1.92	15	17	-	25	30	RS
Aluminium - Silicate	0.78	-	-	-	-	-	-
Aluminium - Sulphate (granular)	0.86	20	17	-	-	-	-
Aluminium - Turnings	0.11 - 0.24	-	-	-	-	-	-
Ammonium - Chloride (crysalline)	0.83	15	15	-	-	-	RA
Ammonium - Nitrate	0.72	25	23	-	-	-	RA
Ammonium - Sulphate, dry	1.10	10	15	-	-	-	-
Ammonium - Sulphate, granulated	0.72 - 0.93	15	15	-	-	-	RA
Ammonium - Sulphate, wet	1.30	15	17	-	-	-	-
Anthracite Coal	0.90	15	17	20	35	40	-
Apples	0.35	15	10 - 12	-	-	-	-
Asbestos ore	1.30	20	18	-	-	-	-
Asbestos, shred	0.32 - 0.40	15	22	-	-	-	-
Ash, caustic salt	0.81	-	-	-	-	-	-
Ash, coal - dry 80mm and under	0.56 - 0.64	10	18	-	-	-	RA - RS
Ash, coal - wet 80mm and under	0.72 - 0.80	25	20	-	30	35	RA - RS
Ash, fly	0.45 - 0.80	15	15	-	25	30	-
Ash, fly coal slack	0.64 - 0.72	15	15	-	25	30	RA - RS
Asphalt	1.00 - 1.30	-	-	-	-	-	-
Asphalt, binder for motorways	1.28 - 1.36	-	-	-	-	-	-
Asphalt, broken	0.70	-	-	-	-	-	RA
Asphalt, paving	1.30	30	20	-	-	-	RA
Bagasse	0.13	30	25	-	-	-	RA
Bakelite, fine	0.48 - 0.64	-	-	-	-	-	-
Ballast, broken	1.50 - 1.80	15	20	22	35	40	-
Barium carbonate	1.15	-	-	-	-	-	RA
Barley	0.60 - 0.70	5	12	20	30	35	RA
Barytes, coarse grained	2.4 - 2.9	-	-	-	-	-	-
Barytes, powdered	1.9 - 2.3	20	18	-	-	-	-
Basalt	1.6 - 2.3	15	17	-	25	30	RE - RS
Basalt lava	2.30	15	17	-	20	25	RE - RS
Bauxite, broken 80mm and under	1.20 - 1.36	20	20	-	25	30	RE - RS
Bauxite, earth broken	1.09	20	18	-	25	30	RE - RS
Bauxite, raw	2.55	15	20	22	35	40	-
Beans	0.85	10	12	-	-	-	RA
Beet pulp	0.40	15	14	-	25	25	RA
Beets	0.65	15	12	-	20	30	-
Bezonite, raw	0.54 - 0.64	-	-	-	-	-	-
Bicarbonate	0.65	20	15	-	-	-	RA
Bituminous coal, fine or nuts	0.90 - 1.00	15	18 - 20	22	30	35	-
Bituminous coal up to 50mm	0.80	20	24	20	35	40	RA - BV
Blast furnace slag, broken	1.30 - 1.60	15	15 - 20	-	25	30	-
Blast furnace slag, granulated, damp	1.50	15	18	-	25	30	RA - RS - RAS
Blast furnace slag, granulated, dry	0.50 - 0.60	10	15	18	25	30	RA - RS - RAS
Bone meal	0.90	20	15	-	-	-	-
Bones	0.60	20	17	-	-	-	-
Borax, coarse	0.96 - 1.04	15	15	-	25	30	RA
Bran	0.25 - 0.30	10	12	-	25	30	RA
Brown Coal briquettes	0.65 - 0.85	20	15	-	-	20	-
Brown Coal, dry	0.65 - 0.80	15	20	22	30	40	RA
Calcium carbide	1.12 - 1.28	15	18	-	25	30	RA
Calcium oxide	0.70	20	18	-	-	-	RA

Material	Bulk Density ρ (t/m^3)	Surcharge Angle β (°)	Maximum Angle of Inclination				Recom-mended Dunlop Quality
			Smooth Belts	height of profiles 6 mm	16 mm	32 mm	
Carbon pellets	0.35	5	15	20	25	30	RA
Carborundum up to 80mm	1.60	10	15	-	-	-	RS
Casein	0.60	15	15	-	-	-	-
Cast Iron swarf	2.08 - 3.20	20	22	-	-	-	RA
Cement, clinker	1.20 - 1.30	15	18	-	30	30	RE - RS
Cement, dry	1.20	10	20	22	30	30	-
Cement Mortar	2.00	10	8	-	-	-	-
Chalk, broken	1.35 - 1.45	15	17	-	30	30	RA
Chalk, pulverised	1.10 - 1.20	15	15 - 18	-	30	30	RA
Charcoal	0.35	15	20	-	25	30	RA
Chestnuts	0.80	5	8 - 10	-	30	40	-
Chrome Ore	2.00 - 2.24	10	17	-	-	-	RE - RS
Clay, calcined	1.28 - 1.60	15	18	20	30	40	RA - RS
Clay, dry	1.60	15	15	-	25	30	-
Clay, dry	1.60 - 1.80	15	28	20	30	35	-
Clay, dry, lumps	0.96 - 1.20	15	15	20	30	35	RA - RS
Clay, wet	2.00	15	20	-	25	30	-
Clay, wet	1.80 - 2.00	15	15 - 18	20	30	40	-
Coal	0.95	10 - 15	10	-	-	30	-
Coal, anthracite up to 3mm	0.95	20	18	20	35	35	RA - BV
Coal dust	0.06 - 0.11	10	5	-	-	-	RA
Coal, pelletized	0.32 - 0.40	-	-	-	-	-	-
Cocoa beans	0.55	10	12	-	-	-	RA
Cocoa powder	0.50	5	20	-	25	30	RA
Coffee beans, green	0.55	10	12	-	25	30	RA
Coffee beans, raw	0.45 - 0.65	5	5	-	25	30	RA
Coffee beans, roasted	0.30 - 0.45	5	5	-	25	30	RA
Coke breeze up to 7mm	0.40 - 0.56	20	20	-	-	-	RA - RS - BV
Coke, loose	0.37 - 0.56	20	18	-	35	35	RE - RS
Coke, petroleum calcined	0.56 - 0.72	20	18	-	-	-	RA
Coleseed	0.65	5	15	20	25	30	-
Compost	0.80	15	15	20	25	30	ROM - ROS
Concrete, stone	2.08 - 2.40	-	-	-	-	-	-
Concrete, wet (Readymix)	1.76 - 2.40	-	-	-	-	-	-
Concrete with gravel	1.80	10	18	20	25	30	-
Concrete with limestone	2.00 - 2.20	15	10	22	27	32	-
Copper ore	1.92 - 2.40	15	15	-	-	-	RE - RS
Copper sulphate	1.20 - 1.36	15	15	-	-	-	RA
Copra flakes	0.40 - 0.60	10	17	-	-	-	ROM - ROS
Cork, broken	0.10 - 0.20	15	15	-	25	30	-
Cork, fine	0.20 - 0.25	15	17	-	25	30	RA
Cork, granulated	0.19 - 0.24	10	17	-	25	30	RA
Cornmeal	0.60 - 0.65	10	15	20	25	30	-
Cotton seed	0.4 - 0.5	20	15	-	-	-	-
Cotton seed flock	0.35	-	-	-	-	-	-
Cryolite, dust	1.20 - 1.44	-	-	-	-	-	-
Cryolite, Lumps	1.44 - 1.60	-	-	-	-	-	-
Dicalcium Phosphate	0.69	-	-	-	-	-	-
Disodium Phosphate	0.40 - 0.50	-	-	-	-	-	-
Dolomite, broken	1.60	15	10	-	30	35	-
Dolomite, pieces	1.44 - 1.60	15	17	-	30	35	RE - RS
Earth, dry	1.60	10	15	20	25	40	-
Earth, excavated dry	1.12 - 1.28	10	17	20	30	30	RA
Earth, wet	2.00	15	20	25	30	45	-
Earth, wet loamy	1.60 - 1.76	15	20	23	30	30	RA

Material	Bulk Density ρ (t/m³)	Surcharge Angle β (°)	Maximum Angle of Inclination Smooth Belts	height of profiles 6 mm	16 mm	32 mm	Recom-mended Dunlop Quality
Ebonite up to 13mm	1.04 - 1.12	-	-	-	-	-	-
Felspar, broken	1.6	20	18	-	25	30	RS
Felspar, lump size 40-80mm	1.44 - 1.76	-	-	-	-	-	-
Felspar, screened to 13mm	1.12 - 1.36	-	-	-	-	-	-
Fertilizer	0.90 - 1.20	15	20	25	30	35	-
Filter cake	1.15	10	15	-	-	-	-
Filter mud	0.60 - 0.80	10	12	-	-	-	-
Fishmeal	0.55 - 0.65	5	14	25	30	30	MORS-ROM
Flaxseed	0.70 - 0.75	5	14	20	30	35	MORS
Flourcalcium 40-80mm	1.76 - 1.92	-	-	-	-	-	-
Flourcalcium, screened	0.56 - 1.68	-	-	-	-	-	-
Flourspar	2.50	15	17	-	25	30	RE - RS
Fly ash	0.45 - 0.80	15	15	-	25	30	-
Foundry waste	1.12 - 1.60	-	-	-	-	-	-
Fruit	0.35	10	15	-	-	-	-
Fullers earth, dry	0.48 - 0.56	10	15	-	-	-	RA
Fullers earth, oil filter, burnt	0.64	-	-	-	-	-	-
Fullers earth, oil filter raw	0.56 - 0.64	-	-	-	-	-	-
Fullers earth, oily	0.96 - 1.04	20	20	-	-	-	RA
Glass, broken	1.30 - 1.60	15	18 - 20	-	-	-	-
Grain	0.60	-	12	-	-	-	RA
Granite, 40-50mm lumps	1.36 - 1.44	-	-	-	-	-	-
Granite ballast	1.40 - 1.80	15	15	-	25	30	-
Granite, broken	1.52 - 1.60	-	-	-	-	-	-
Granite, screened, up to 13mm	1.28 - 1.44	-	-	-	-	-	-
Graphite flock	0.60 - 0.65	10	15	-	-	-	RA
Graphite, pulverized	0.45	5	15	-	-	-	RA
Grass seed	0.22	5	15	25	25	30	RA
Gravel	1.44 - 1.60	-	-	-	-	-	-
Gravel, dry	1.40 - 1.50	10	17	-	30	35	RA
Gravel, wet	1.80 - 1.90	15	18	-	30	35	RA
Green fodder	0.35	20	10 - 15	-	25	30	-
Ground nut kernels	0.35	10	14	18	20	25	ROM
Ground nuts, with shells	0.30	10	14	20	25	30	ROM
Ground nuts, without shells	0.35	18	12	-	20	25	-
Guano, dry	1.12	-	-	-	-	-	-
Gypsum, broken	1.35	15	15	-	25	30	-
Gypsum, burnt	1.80	10	15	-	25	30	RA
Gypsum mortar	1.20	10	-	8	-	-	-
Gypsum, pulverized	0.95 - 1.50	10	18	22	30	30	RA
Gypsum, sieved	1.45	15	17	-	25	30	RA
Household refuse	0.80	10	15 - 20	20	25	30	-
Husks, dry	0.45	22	20	-	-	-	RA - ROS
Husks, wet	0.90	22	20	-	-	-	RA - ROS
Iron Ore, broken	2.00 - 4.50	15	18	25	30	35	RS - RAS
Iron Ore, crushed	2.16 - 2.40	-	-	-	-	-	-
Iron Ore, pellets	5.00	5	12	-	20	25	RS - RAS
Iron Oxide, pigment	0.40	-	-	-	-	-	-
Iron turnings (Swarf)	2.00	-	-	-	-	-	-
Kaolin, broken	1.00	20	19	-	-	-	RA
Kaolin clay up to 80mm	1.00	20	19	-	-	-	RA
Kaolin, pulverized	0.70 - 0.90	30	20	-	-	-	RA
Kieselguhr	0.17 - 0.22	5	15	-	-	-	-
Kiln Brick	1.60	17	17	-	25 - 30	25 - 30	RA - RS

Material	Bulk Density ρ (t/m³)	Surcharge Angle β (°)	Maximum Angle of Inclination: Smooth Belts	height of profiles: 6 mm	16 mm	32 mm	Recom-mended Dunlop Quality
Lactose (milk sugar)	0.51	-	-	-	-	-	RA - RS
Lead - Arsenate	1.15	-	-	-	-	-	-
Lead Ore	3.20 - 4.70	15	17	-	25	30	RA
Lead - Oxide	0.96 - 2.40	15	15	-	25	30	RA
Legumes	0.85	5 - 10	8 - 10	-	-	20	-
Lime, hydrated	0.60	15	15	-	-	-	RA
Lime, lumps	1.20 - 1.36	-	-	-	-	-	-
Lime, pulverized	1.00 - 1.20	5	20	22	25	27	-
Lime, slaked	0.64	-	-	-	-	-	-
Lime up to 3mm	0.96	15	18	-	25	30	RA
Limestone, Broken	1.40 - 1.50	15	18	22	30	35	RA - RS
Limestone, dust	1.30 - 1.40	5	15	-	25	30	RA - RS
Linseed	0.72	5	12 - 15	20	30	35	-
Linseed cake	0.75 - 0.80	10	15	-	-	-	-
Magnesite	3.00	15	17	-	-	-	-
Magnesite, fine	1.04 - 1.20	-	-	-	-	-	-
Magnesium chloride	0.52	20	20	-	-	-	RA
Magnesium oxide	1.90	10	17	-	25	30	RA
Magnesium sulphate	1.10	10	16	-	25	30	RA
Magnetite	3.00	15	17	-	-	-	-
Maize	0.70 - 0.75	5	10 - 12	-	25	30	-
Maize, shelled	0.70	10	10	-	25	30	BV - ROM - BVO
Malt, dry	0.30 - 0.50	5	15	20	25	30	-
Manganese dioxide	1.28	-	-	-	-	-	-
Manganese ore	2.0 - 2.3	25	20	-	-	-	-
Manganese sulphate	1.12	10	15	-	-	-	-
Manure	1.10	15	20	25	30	30	ROS
Marble, broken	2.70	15	15	-	25	25	-
Marble, crushed, up to 13mm	1.44 - 1.52	15	17	-	25	25	RA - RS
Marl, dry	1.20 - 1.30	10	17	-	25	30	RA - RS
Meal	0.60 - 0.70	5	20	22	30	35	-
Molybdenite, powdered	1.71	15	18	-	-	-	RA
Mortar, cement	2.00	10	-	-	20	-	RA
Mortar, Gypsum	1.20	10	-	-	20	-	RA
Mortar, Lime	1.70	10	8	-	20	-	-
Moulding sand, core sand	1.04	15	20	-	-	-	-
Moulding sand, knock-out	1.45 - 1.60	10	18	-	-	-	-
Moulding sand, prepared	1.30 - 1.45	10	20	25	30	35	-
Mushrooms	0.40	10	15	-	25	30	-
Nickel Ore	2.40	-	-	-	-	-	-
Oats	0.55	5	15	20	25	30	RS
Oil Sand	1.50	15	15	-	25	30	-
Ore, Copper	1.92 - 2.40	15	15	-	-	-	RE - RS
Ore, Iron	2.00 - 4.50	15	18	25	30	35	RS - RAS
Ore, Lead	3.2 - 4.7	15	17	-	25	30	RA
Ore, Manganese	2.0 - 2.3	25	20	-	-	-	-
Ore, Zinc	2.40	15	15	-	25	30	-
Overburden	1.7	15	17	-	-	-	-
Peas, dried	0.70 - 0.80	5	14	18	20	25	RA
Peat, dry	0.32 - 0.80	15	15	25	25	30	-
Peat, wet	0.65 - 1.00	15	12	-	25	30	-
Phosphate, broken	1.20	15	15	-	25	30	-
Phosphate, fertilizer	0.96	-	-	-	-	-	-
Phosphate, sand, cement	1.36	-	-	-	-	-	-

Material	Bulk Density ρ (t/m³)	Surcharge Angle β (°)	Maximum Angle of Inclination Smooth Belts	height of profiles 6 mm	16 mm	32 mm	Recom-mended Dunlop Quality
Phosphate, pulverized	0.96	-	-	-	-	-	-
Phosphate rock, broken	1.35 - 1.45	15	15	-	25	30	-
Plaster	1.70	10	15	-	20	-	RA
Portland cement	1.50	20	18	-	30	30	RA
Portland cement, loose	0.96 - 1.20	15	15	-	-	-	RA
Potash	1.35	10	17	20	25	30	RA - RS
Potash	1.10 - 1.60	10	20	-	30	35	-
Potash, broken	1.20 - 1.35	-	-	-	-	-	-
Potash salts, sylvite etc	1.28	-	-	-	-	-	-
Potassium (Saltpetre)	1.22	-	-	-	-	-	-
Potassium chloride pellets	1.92 - 2.08	-	-	-	-	-	-
Potassium sulphate	0.67 - 0.77	-	-	-	-	-	-
Potatoes	0.75	15	12	-	20	25	RA
Pulp, dry	0.20 - 0.25	15	15	-	25	30	-
Pulp, wet	0.40	15	12	-	25	25	-
Pumice stone	1.20	15	17	-	25	30	-
Pumice stone sand	0.70	10	15	20	25	30	-
Pyrites, Iron lump size 50-80mm	2.16 - 2.32	-	-	-	-	-	-
Pyrites, Iron Sulphide	2.00 - 2.50	15	15	-	25	30	-
Pyrites, pellets	1.92 - 2.08	15	15	-	25	30	RA - RS
Quartz, broken	1.60 - 1.75	15	17	-	25	30	-
Quartz, lump size 40-80mm	1.36 - 1.52	-	-	-	-	-	-
Quartz sand	1.70 - 1.90	15	17	20	25	30	RA - RS
Rape seed	0.80	5	12	15	25	25	-
Rice	0.70 - 0.80	5	8	12	20	25	RA
Roadstone, Broken (Porphyry)	1.50 - 1.70	15	20	22	30	35	-
Rock salt	1.00 - 1.20	10	15	-	30	35	-
Rubber, dust	0.60 - 0.65	10	20	-	25	30	-
Rubber, pelletized	0.80 - 0.88	10	20	-	25	30	RA
Rubber, reclaim	0.40 - 0.48	10	20	-	25	30	RA
Run of mine coal	0.80 - 1.00	15	18	22	35	40	-
Rye	0.70 - 0.80	5	15	20	30	35	RA
Salt, coarse	0.70 - 0.80	10	17	-	30	35	RA
Salt, common, dry	0.64 - 0.88	-	-	-	-	-	-
Salt, common, fine	1.12 - 1.28	10	17	25	30	35	RA
Saltpetre	1.10	5	15	20	25	30	-
Saltpetre	1.70	10	16	-	-	-	RA
Sand and gravel, dry	1.50 - 1.80	10	18	20	30	35	RA
Sand and gravel, wet	1.80 - 2.10	15	20	25	30	35	RA
Sand, dry	1.30 - 1.60	10	17	20	25	30	RA
Sand, foundry, prepared	1.44 - 1.60	20	17	-	30	35	RA
Sand pebble, dry	2.00	5	15	20	25	30	RA
Sand, wet	1.60 - 2.00	15	20	25	35	35	RA
Sandstone	1.36 - 1.44	15	17	-	30	35	RA - RS - RAS
Sawdust	0.20 - 0.30	20	18	20	25	25	ROM
Sewage Sludge	0.64 - 0.80	-	15	-	-	-	RA - ROS
Shale	2.70	15	18	-	25	30	RA - RS - RAS
Shale, broken	1.44 - 1.60	-	-	-	-	-	-
Shale dust	1.12 - 1.28	-	-	-	-	-	-
Shale, lump size 40-80mm	1.36 - 1.52	-	-	-	-	-	-
Sinter, blast furnace, dry	1.50	15	15	-	25	30	-
Slag, blast furnace	1.50	15	-	25	30	-	-
Slag, blast furnace, broken	1.28 - 1.44	10	15	-	25	30	RA - RS - RAS
Slag, porous, broken	0.60	15	15	-	25	30	-
Slate, broken	1.40 - 1.55	15	17	-	25	30	RA - RE

Material	Bulk Density ρ (t/m^3)	Surcharge Angle β (°)	Maximum Angle of Inclination Smooth Belts	height of profiles 6 mm	16 mm	32 mm	Recom-mended Dunlop Quality
Soap beads or granules	0.24 - 0.40	10	12	18	25	-	RA
Soap flakes	0.15 - 0.35	10	18	22	25	30	RA
Soda	0.90 - 1.20	-	17	-	20	25	RA
Soda ash, heavy	0.88 - 1.04	-	-	-	-	-	-
Soda ash, light	0.32 - 0.56	-	-	-	-	-	-
Sodium Bicarbonate	0.65	-	-	-	-	-	-
Sodium nitrate	1.12 - 1.28	-	-	-	-	-	-
Soot	0.40 - 0.75	5	15	20	25	30	-
Soya beans	0.80	15	18	-	-	-	ROS
Steel trimmings, shredded	1.60 - 2.40	-	-	-	-	-	-
Stone, broken, small	1.50 - 1.80	15	15	-	25	30	-
Stone, large, sized	1.40 - 1.60	15	15	-	20	25	-
Stone, sized	1.55 - 1.70	10	15	-	22	27	-
Sugar cane, raw	0.88 - 1.04	10	19	22	25	-	RA
Sugar, granulated	0.60 - 0.80	5	15	20	25	-	RA
Sugar, powdered	0.80 - 0.96	0	12	16	20	25	RA
Sugar, raw	0.90 - 1.50	15	20	20	25	-	-
Sugar-beet	0.70	18	18	-	-	30	RA
Sugar-beet, sliced, damp	0.40	15	18 - 20	-	25	30	-
Sugar-beet, sliced, dry	0.20 - 0.25	-	18	-	25	30	-
Sugar-beet, wet	0.88 - 1.04	-	-	-	-	-	-
Sulphate, ferrous	0.80 - 1.20	-	-	-	-	-	-
Sulphur, broken	1.30	15	17	20	30	35	RA
Sulphur, pulverized	0.90	5	19	25	30	35	RA
Table Salt	0.80	10	20	25	30	35	-
Tile, hard	2.00	-	-	-	-	-	-
Tile, soft	1.60	-	-	-	-	-	-
Titanium, washed	0.96 - 1.12	15	17	-	-	-	RA - RS - RAS
Tomatoes	0.90	10	10 - 12	-	20	30	-
Wheat	0.75	10	14	20	30	35	ROM
Wheat, flour	0.55 - 0.65	5	20	22	30	35	-
Wood chips	0.30 - 0.40	-	15	-	20	-	-
Wood chips, wet	0.60 - 0.85	10	20	-	20	-	-
Wood shavings	0.16 - 0.48	5	12	18	20	25	RA - ROM
Zinc Ore	2.40	15	15	-	25	30	-
Zinc Ore, burnt	1.60	-	-	-	-	-	-
Zinc Ore, crushed	2.56	15	17	-	25	30	RA - RS - RAS
Zinc Oxide, heavy	0.48 - 0.56	15	17	-	20	25	RA
Zinc Oxide, light	0.16 - 0.24	-	-	-	-	-	-
Zinc Sulphate	3.70	15	17	-	25	30	RA