Edition

Conveyor Belt System Design Service Manual

Conveyor Belt Group

Continental CONTITECH

ContiTech
Specialist in rubber and plastic technology

Conveyor Belt System Design

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An insignificant **side-note** for all those who are already accustomed to using SI units or units derived from them and who take advantage of this “standard” settlement by the international units system.

For all other readers it is pointed out that an “Act on Units in Measuring Procedure”, regulating the use of technical units in business and official transactions, was passed on 2nd July 1969 and became mandatory for us when the transitional period expired on 1st January 1978.

The effects of this Act on materials handling technology are minimal as most of the units already widely used in this sphere have undergone no change. One essential point is the uniform stipulation of the

kilogram (kg) as a unit for mass

It was furthermore stipulated that the word “weight” may be used only as a mass quantity. The weights listed in a large number of tables and reference books thus retain their numerical value. Other units of weight or of mass are the gram (g) and the ton (t).

We recall: one kilogram was accelerated by the gravitational attraction of the earth at approx. 9.81 m/s^2 and thus exercised on the base it is resting upon a **weight** of one kilogram-force. In the Space Age we now depart from this unit of force, that applied solely under the gravitational conditions of the earth, and use exclusively the

Newton (N) as a unit for force

One kilogram is accelerated by one Newton at 1 m/s2 or one kilogram acts under the influence of the acceleration due to gravity on its base with the force of approx. 9.81 Newton. All forces that were formerly expressed in kilogram-force attain a numerical value increased approximately by the factor 10 with Newton as a unit.

The breaking strength of a conveyor belt in kilogram-force related to the unit of width in cm retains approximately its familiar numerical value if the breaking strength in N is related, as practised below, to the width in mm.

1 kilogram-force/cm ≈ 1 N/mm

It follows from the introduction of the Newton as a unit for force that the acceleration due to gravity g occurs in all calculation formulae in which weights are determined or other forces with their origin in weights are calculated. For this purpose

Acceleration due to gravity g = 9.81 m/s^2

can be inserted with sufficient accuracy. A simplification results in the calculation of power in W (Watt) by the multiplication of force in N and speed in m/s. A conversion factor is then superfluous.

This short introduction is aimed at drawing your attention to the changeover resulting from these regulations for the following calculation principles and data.

D – Conveyor belt calculation

This section of the ContiTech Conveyor Belt Service Manual comprises design fundamentals and data design of the belt system and the practical application.

The calculation of the belt conveyor for bulk material set out at the beginning of these calculation principles is based on specifications contained in DIN standard 22101 (February 1982) and is supplemented by extending design procedures. Essential calculation stages not included in the DIN 22101 standard specifications have been described for instance in publications by Vierling*

The designations of formula quantities have been adapted as far as possible to general guidelines. Designations already well known from technical literature in this field have also been used, however, where we felt them to be beneficial.

Only the “statutory units” (SI units or units derived therefrom) on which the “International Units System” (SI) is based are used in the formulae, tables and diagrams. To facilitate comprehension, the units used in the further course of the calculation are stated after the respective formulae, whereby the units listed in the index of symbols are to be used. A brief dimensional consideration helps in cases of doubt. The units to be used are stated in the case of formulae with conversion factors or dimension-affected constants.

* Vierling, A.: Zum Stand der Berechnungsgrundlagen für Gurtförderer Braunkohle Wärme und Energie 19 (1967) No 9,P 309-315
Vierling, A.: Zur Theorie der Bandförderung Continental-Transportband-Dienst 1972, 3rd edition.

In general, the following data are known before a belt conveyor is designed:

- **Type of material to be conveyed (e. g. bulk weight, lump size, angle of repose)**
- **Flow of material to be conveyed (e. g. mass flow, volume flow)**
- **Conveying track (e. g. conveying length, conveying height).**

These data are a basis for selection of the type of belt conveyor and thus for the type of conveyor belt. The decision on whether it is a plain standard conveyor belt, a steep-angle conveyor belt or a special conveyor with the corresponding special-purpose belt serves as a reference point for the **stipulation of the principal data**, in particular of conveying speed and belt width, but may necessitate for instance in addition the stipulation of the troughing design, the belt surface profile or the pitch of elevator buckets.

The next factors to be determined are motional resistances and required power of the belt conveyor. If the design is already established, it is the size, position and type of the driving motors that have to be recorded besides the data stated hitherto, as these may have a decisive influence on the selection of the conveyor belt. The **calculation of the belt conveyor** leads on to the peripheral forces at the driving pulleys. Their magnitude depends on the extent of the motional resistances but varies for temporary operating conditions such as starting and stopping.

The **design of the tension member** in the conveyor belt then follows. Its tensile strength is determined mainly by the magnitude of the belt tensions. Further influences result from a variety of criteria relating to operation and design.

As the strength of the tension member must always have a specific safety margin over the maximum stress, it is essential to stipulate the safety coefficient or to check the available safety margin on selection of the tension member. Special attention must be paid here to the durability of the conveyor belt at the joints.

The tension member of the conveyor belt is enclosed in the covers, which thus form an effective protection against external influences. For this reason the material and construction are selected to counteract the effects of the material conveyed and of the environment. Conformity to the tension member must also be observed on **selection of the covers**.

Conti conveyor belts are supplied to all parts of the world, using those dispatch facilities best suited to the destination; the packaging guarantees safe transport even in exceptional cases. The optimum part lengths can be determined for each particular instance to provide inexpensive shipment and simple assembly. The thickness, width and weight of the belt are to be given special consideration.

The conveyor belt calculation process described also conforms with the arrangement of the following chapters. The chart shows both the normal sequence of design stages and the feasibility of starting at any section, provided that specific data or parameters are known. In general the single chapters go deeper into the respective subject with increasing sub-division, so that a rough assessment of the belt structure can be made by consulting the general chapter alone. Special reference is made to those calculation points at which an assessment is recommended.

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➲ **Stipulation of principal data.... D–2**
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➲ **Steep-angle conveyor belts.... D–6**

➲ **Elevator belts..... D–7**

➲ **Piece goods handling and belts for sliding bed operation D–8**

➲ **Appendix..... D–9**

The staff of the ContiTech with its sophisticated research and development resources is available for direct consultation and to answer any questions. Please address enquiries to:

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The computer programmes filed in the constantly expanding library of the Department for Application Technique enable enquiries to be dealt with promptly Apart from specific individual problems, this facility permits above all extensive calculations and design alternatives to be investigated without loss of time Reference can be made to the currently complied programmes at those points of the conveyor belt calculation marked with the key word ‚CONTI-COM'.

D – 1 Basic expressions and definitions

Continuous conveyors: Continuous conveyors are mechanical, pneumatic and hydraulic conveying devices with which the material to be handled can be moved continuously on a fixed conveying track of restricted length from feeding point to discharge point, possibly at varying speed or in a fixed cycle. These conveyors are available in stationary or mobile versions and are used for the handling of bulk materials or piece goods.

Belt conveyors: Continuous conveyors whose belts have a tension member consisting of synthetic fabrics or steel cables with rubber or synthetic covers; the belts are supported by straight or trough-shaped idlers or have sliding support on a smooth base as a tension and support member. The actual conveying is done on the top run, in special cases on the top run and the return run. Belts with cleated top covers, special-purpose belts or sandwich belts are used for steep-angle conveying.

1 Feed
2 Discharge
3 Head pulley (drive pulley)
4 Snub or deflecting pulley
5 Tail pulley (takeup pulley)
6 Top run (tight side)
7 Return run (slack side)
8 Top run idlers
9 Return run idlers
10 Feed rollers
11 Flat-to-trough transition
12 Trough-to-flat transition
13 Feed chute
14 Belt cleaner (transverse scraper)
15 Belt cleaner (plough-type scraper)
16 Drive unit
17 Counterweight

Belt-type elevators: Continuous conveyors with buckets or similar containers as carriers; these either scoop the material or are filled by metering hoppers and emptied at specific discharge points. The tension unit consists of belts to which the containers are attached. Transport is effected at any angle from vertical to horizontal.

Flow of material: Mass or volume of the conveyed bulk material or piece goods per unit of time in continuous conveying. In contrast, the capacity is not time-related.

Conveying capacity: The volume capacity or capacity of material conveyed that can be attained with the given conveying speed and the available cross-section area or the container volume and spacing.

Conveying speed: Speed of the material conveyed. The conveyor belt as the support member determines the speed of the material on it.

Centre distance: The distance between head and tail pulley of the conveyor. The belt length as the inside circumference of the endless, slack belt results from this distance only when the pulley circumference and any belt loops (tension loops, discharge loops etc.) present are taken into account.

Conveying length: The distance between the centre of the material feeding point and the axle of the discharge pulley. If the material conveyed is stripped off, the centre of the material discharge is to be taken instead. In general the conveying length is approximately equal to the centre distance. The conveying length may, however, be smaller than the centre distance or be variable during the conveying process.

Conveying height: The difference in height between material feed and material discharge. Belt conveyors with sections at different gradients yield the section heights allocated to the section lengths.

Belt support: The belt is generally supported by fixed idlers or by suspended idlers. The belt can be flat or troughed by multi-roll idlers. Troughing permits a greater flow of material and promotes improved belt training. The idler spacing is normally larger on the return run than on the top run and can also be graduated within one belt conveyor. Special idler designs or arrangements are frequently selected at the feeding points and for belt cleaning.

Flat belt (with CONTIWELL® side wall) | V-trough | 3-roll troughing | 5-part troughing | Square-troughing belt

Conveyor belt: The task of the conveyor belt is to carry the material handled and simultaneously to transmit the driving forces to overcome the motional resistances. The conveyor belt consists in general of the tension member and the top and bottom covers, which form a core-protecting covering.

Those conveyor belts used in belt conveyors are to be regarded as continuous conveying elements composed of one or more belt sections joined together at their ends. Short conveyor belts can also be manufactured in endless versions.

Steel cable belt:

Fabric carcass belt:

Steep angle conveyor belts: Rollback of the material handled can be prevented by means of chevron cleats, fins or cross partitions on the carrier side when the conveying gradient is steep. The design can be adapted to the material to be conveyed and to the transverse stability required. Cross partitions and corrugated sidewalls can be combined to form the corrugated box-section belt.

Steep angle conveyor belt with chevron cleats

HÖCKER® conveyor belt

CONTIWELL® box-section belt

Elevator belts: These are used in belt-type bucket elevators and are provided with special brackets for the attachment of containers, buckets or belt slings.

Tension member: The task of the tension member is to transmit the forces induced at the drives for overcoming the system resistances to the point where they are needed. In the fabric ply belt, the tension member consists of one or more plied fabrics. In the case of a steel cable belt, the tension member consists of steel cables arranged on a single plane and running parallel to each other longitudinally; these cables are imbedded in core rubber.

Core rubber: The core rubber envelops the steel cables of the steel cable tension member, providing good adhesion to the Cover material with a high dynamic carrying capacity. The physical properties are maintained even after repeated curing, for instance on splicing.

Belt covers: The covers protect the tension member from damage and other environmental influences. The top and bottom covers may vary in thickness. The bottom cover can be omitted from bare-bottom-ply belts for sliding bed operation. Additional elements designed to increase impact resistance or for monitoring purposes may be located in the covers. The Cover materials can be selected to suit any application.

Surface patterning: For improved holding of materials on gradients or for special applications, the top cover can be manufactured in a patterned or a cleated version.

D – 2 Stipulation of principal data

➲ **Material handled 2.1**

➲ **Flow of material handled 2.2**

➲ **Conveying track 2.3**

➲ **Type of belt conveyor 2.4**
Belt width 2.4.1
Belt speed 2.4.2
Belt support 2.4.3

➲ **Carrying capacity 2.5**

The designing of a conveyor belt begins with an investigation into the service requirements and the stipulation of the principal data characterizing the specific application. Data already available can be checked against the guide values stated in this section.

The optimum conveyor belt cannot be selected by means of the principal data alone, as the operating method and the belt conveyor design also have a considerable influence. If the stress and strain on the conveyor belt are not known in detail, a calculation of the belt conveyor must be executed with reference to the bulk material transport up to approx. 30° system gradient in section D – 3. Sections D – 6 to D – 8 contain supplementary data for the calculation of steep-angle conveyors for bulk materials, of elevators and piece goods conveyors.

D–2.1 Material handled

The physical and chemical properties of the material to be handled must be taken into account when selecting the belt conveyor and designing the conveyor belt.

Properties of material to be handled (guide values)

Material to be handled	Bulk density ρ in t/m³	Angle of repose in°	Max. gradient of fight δ	Possible effect		
				mech.	chem.	temp.
Ammonium sulphate	0.75–0.95		22	+	+ +	
Artificial fertilizer	0.9–1.2	12	12–15			
Ash, dry	0.65–0.75		16			+
Ash, wet	0.9	15	18			+
Asphalt, crushed	0.7		22			
Bauxite, crushed	1.2–1.4		18–20	+ +		
Bauxite, fine	1.9–2.0		18	+		
Beet	0.65–0.75	10	15–17			
Beet chip, wet	0.5	18	15–17		+	
Berry and Lorraine iron, fine	3.2	15	18			
Blast furnace slag	1.2–1.4		18	+ +		+
Cement	1.2–1.5	0–10	15–20			+ +
Cereals (not oats)	0.7–0.85	10–12	14		+	
Clay, damp	1.8	15–18	18–20			
Clinker	1.2–1.5	10–15	18	+ +		+ +
Coal, fine	0.8–0.9	10	18–20			
Coal, raw	0.75–0.85	18	18	+		
Coke	0.45–0.6	15	17–18	+ +		+ +
Concrete, wet	1.8–2.4	0–5	16–22	+ +		
Copper ore	1.9–2.4	15	18	+ +		
Crushed rock	1.5–1.8	10–15	16–20	+ +		
Felspar, crushed	1.6		18	+ +		
Fish meal	0.55–0.65		25			
Flour	0.5–0.6	15	15–18			
Foundry sand	1.2–1.3		20	+		+ +
Foundry waste	1.2–1.6		16	+ +		+
Fruit	0.35	10	12–15		+	
Glass, crushed	1.3–1.6		12–15	+ +		+
Granite, crushed	1.5–1.6		20	+ +		
Graphite, powder	0.5		20			
Gravel and sand, wet	2.0–2.4	15	20			
Gravel, graded, washed	1.5–2.5	18	12–15			
Gravel, ungraded	1.8	15	18–20			
Gypsum, crushed	1.35		18			
Gypsum, powdered	0.95–1.0		23			
Household refuse	0.6		18–20	+	+	
Iron ore	1.7–2.5	15	18	+ +		
Iron ore, pellets	2.5–3.0	12	15			
Lignite briquettes oval	0.7–0.85	15	12–13			
Lignite, dry	0.5–0.9	15	15–17			

Properties of material to be handled (guide values)

Material to be handled	Bulk density ρ in t/m³	Angle of repose in°	Max. gradient of fight δ	Possible effect		
				mech.	chem.	temp.
Lignite, wet	0.9	15–20	18–20			
Lime, lumps	1.0–1.4	15	15–20			+
Limestone, crushed	1.3–1.6	15	16–18	+		
Maize	0.7–0.75	12	15		+	
Manganese ore	2.0–2.2	15	18–22	+ +		
Matchwood	0.2–0.35		22–24			
Oil cake	0.7–0.8	10	15		+ +	
Oil sand	1.6–1.8	15	20		+ +	
Overburden	1.6–1.7	15	17	+ +		
Peat	0.4–0.6		16			
Phosphate, crushed	1.2–1.4	15	18–20	+ +		
Phosphate, fine	2	12–15	18	+		
Potash	1.1–1.6	15	18	+		+
Rock salt	1–2	15	18	+		
Rubber, pellets	0.8–0.9		12			+
Salt, coarse	1.2	15	18–20			
Salt, fine	1.2–1.3	15	15–18			
Sand, dry	1.3–1.4	15	16–20	+ +		
Sand, wet	1.4–1.9	15	20–25	+		
Sandstone, crushed	1.3–1.5			+ +		
Scrap metal	1.2–2.0			+ +	+	
Slag	1.3		18	+ +		+
Slate, crushed	1.4–1.6			+		
Sludge	1.0	0				
Soap flakes	0.15–0.35		15			
Soil, damp	1.5–1.9	15–20	18–20	+		
Sugar cane	0.2–0.3					
Sugar, raw	0.9–1.1		15		+	
Sugar, refined	0.8–0.9		20			
Timber, pieces	0.25–0.5		20–25			

Key:

+ medium wear
chemically aggressive
temperatures above 70°

+ + heavy wear
chemically highly aggressive
temperatures above 120°C

The dynamic angle of repose is in general lower than the natural angle of incline of the material handled and depends on the type of material handled, the conveying speed, the design of the feeding point and the gradient of the system.

It permits an assessment of the cross-sectional form of the bulk material on the belt. The maximum gradient stated for the flight applies to a standard, uncleated belt surface.

D – 2.2 Flow of material handled

If a specific quantity of material is to be handled in a prescribed time, the

Mass flow Q_m in t/h

results with the mass of the material handled, and the

Volume flow $Q_v = Q_m / \rho$ in m³/h

for bulk materials with bulk density ρ.

The necessary Conveying capacity of the conveyor belt or belt conveyor to be selected is determined by these two values. To what extent downtime may occur through maintenance, breakdown and repairs and to operation-related interruptions in conveying must be taken into account when making the selection.

	Operating hours per year		
Working days per year	**1 shift***	**2 shift**	**3 shift**
365	2920	5840	8760
250	2000	4000	6000
200	1600	3200	4800

*1 shift ≙ 8 hours

Guide values for maximum material flows

	Q_v in m³/h	Q_m in t/h
Belt conveyors	approx. 25 000	approx. 40 000
Steep angle conveyor belts	approx. 1 400	approx. 3 000
Elevators	approx. 1 500	approx. 2 500

D – 2.3 Conveying track

Owing to its continuous conveying process, the belt conveyor has relatively low flight loads and can consequently be adapted to any routing. The belt gradient can be changed at random, which may provide the most economic solution for longhaul conveying systems in particular. Certain minimum radii of the concave or convex curves must be adhered to. Laying in horizontal curves is also feasible with belt conveyors. The use of a high-strength tension member permits considerable conveying lengths and conveying heights to be attained. Centre distances in the magnitude of 5 to 20 kilometres and conveying heights of up to several hundred meters are no longer a rarity.

Belt conveyors with steep angle belts are normally designed for shorter flights at a very steep gradient. The belt guidance can be adapted largely to the specific application in this case too with the use of appropriate supporting elements.

Belt elevators are used almost exclusively for vertical transport with conveying heights of up to almost 100 meters. The operating principle permits no curves or significant gradients in the flight. The use of steep angle conveyor belts and elevator belts overlaps in the range of very steep to vertical transport. Individual adaptation of the conveyor element is frequently essential in this instance.

D – 2.4 Type of belt conveyor

The transport of bulk material and piece goods on conveyor belts with no surface partitioning or on hugger belts is restricted by the gradient at which the material being handled begins to slip or to roll. Nor is faultless transport guaranteed if starting or stopping induces this process. The critical conveying gradient angle for a smooth belt is between 15° and 20° for the majority of different types of material handled. Furthermore, special belts permitting steeper gradients have to be used. The stated guide values apply to ascending transport.

1 Conveyor belts with no surface partitioning (Conveyor belts with cover patterning for bulk materials)
2 Piece goods conveyor belts with cover patterning , Rollgurt*
3 Belts with chevron cleats
4 Fin belts, box-section belts with corrugated sidewalls
5 Conveyor belts in sandwich design
6 Elevator belts

The maximum gradients are to be selected on the small side, if easily flowing bulk material is to be conveyed at an angle on conveyor belts of groups 1,2 or 3. With increasing surface pattern depth and with transport in buckets etc., the type of material handled is of essential significance only with regard to granularity or piece size.

The velocity data, too, represent guide values that are intended to help in selecting the appropriate type of belt but have to be determined more precisely in the subsequent design. If a steep angle conveyor belt or an elevator belt is found to be suitable for the application or if belts for piece goods or for sliding bed operation are to be designed, special attention should be paid to the information in the relevant chapters.

For bulk material transport on conveyor belts with cover patterning (group 1) the maximum conveying gradients can be set approx. 5° higher than corresponding to the respective angle of the material concerned.

If a control of all requirements has led to the selection of a conveyor with a belt from groups 2 to 6, the layout information given for these special conveyor belts in Sections D – 6, D – 7 and D – 8 is to be observed.

* Tubular belt, developed by ContiTech and PWH AS (please ask for special literature).

D – 2.4.1 Belt width

The belt width should be selected as far as possible from standardized or customary widths as the dimensions of the idlers and other constructional elements of the belt conveyor are coordinated with these widths.

Standard belt widths*

Belt width B in mm	300	400	500	600	650	800
	1000	1200	1400	1600	1800	
	2000	2200	2400	2600	2800	
	3000	3200				

In the case of troughed belts, the belt width must not fall short of certain dimensions, depending on the lump size (edge length) of the material to be handled, as the material can otherwise not be transported safely. With strongly eccentric pieces there is furthermore the risk of the belt mistracking and of idlers being damaged by material projecting beyond the belt.

Minimum belt widths

Size k of lumps in mm	Mind. belt width B in mm	
	100	400
	150	500
	200	650
	300	800
	400	1000
	500	1200
	550	1400
	650	1600
	700	1800
	800	2000

This data applies to approximately cubic lumps. Narrower widths are also admissible for oblong lumps (so-called “slabbies“) or when single pieces are imbedded in mainly fine material.

It is to be observed that the troughability too is influenced by the belt width. The troughability decreases with diminishing belt width. On final determination of the belt structure, the troughability (D – 4.3.2) is to be checked.

* ContiTech conveyor belts are currently available in widths of up to 6400 mm. Textile carcass belts are available from stock in widths of 400 to 2 100 mm.

D – 2.4.2 Belt speed

The selection of the belt speed is of decisive significance for the further designing of the belt conveyor and of the belt.

Application features	Conveying speed v in m/s
Special cases, process-related (e. g. cooling conveyors)	0.5
Small flows of material, protective transport (e. g. coke bench conveyors)	0.5 – 1.5
Standard application conditions and material handled (e. g. gravel conveyance)	1.5 – 3.5
Large flows of material, long conveying lengths (e. g. overburden conveyance)	3.5 – 6.5
Special cases (e. g. jet conveyors)	6.5 and more

In general a more economic design can be achieved with higher conveying speeds. The greater the conveying lengths and thus the belt lengths, the more significant this is, so that maximum conveying speeds will be selected especially in such cases. The limits imposed first and foremost by the type and nature of the material to be handled can be exceeded in many instances if, for example, additional measures are taken at the feeding points to eliminate or diminish the drawbacks of high conveying speeds.

Bulk material features	
strongly abrasive fine and light fragile coarse grained (sized) and heavy	low conveying speeds
slightly abrasive medium bulk weight medium granularity (unsized)	higher conveying speeds

Increased conveying speed results in an increased Conveying capacity with a constant belt width. It may thus be possible to select a narrower belt width or a simpler troughing design for a given flow of material. In addition, reduced drive tractions and consequently reduced dimensioning of all elements constituting the belt conveyor may result. Drawbacks are increased belt wear, to which special attention should be paid in short belt conveyors, an increased risk of damage to the material handled and increasing power requirements for large capacities.

Reduced conveying speed correspondingly results in a larger belt width or a higher-capacity troughing design with the given flow of material. The increased drive tractions are offset by reduced belt wear and a reduced risk of damage to the material handled.

Belt speeds – guide values from systems in operation

Belt speeds of the standard series:

0.42 0.52 0.66 0.84 1.05 1.31 1.68 2.09 2.62 3.35 4.19 5.2 6.6 8.4 in m/s

Standard conveying speeds

Possible conveying speeds

D – 2.4.3 Belt support

The cross section of the conveyor belt is formed by the arrangement of its supporting elements - idlers or slip planes - and thus adapted to the application conditions and the necessary Conveying capacity.

The supporting system is generally effected by idlers with largely standardized lengths and diameters. The lengths are prescribed so as to ensure reliable belt support even with the mistracking that is permitted within certain limits. The selection of the idler diameter is influenced by the conveying speed. Further data to help determine idler weights and spacings are given in the relevant chapters.

Belts for bulk material transport are supported almost exclusively by rigidly mounted idlers or suspended (garland) idlers.

Application features for belt support systems

Support	Bulk materials		Piece goods
	Rollers		Plane-borne or rollers
Flat belt	as hopper drawing device and conveyor type scales	laterally chuted or with corrugated sidewalls as skirting	belt width dependent on piece goods dimensions
	as belt with corrugated sidewalls	with corrugated side-walls to increase Conveying capacity	single piece should not project beyond sidewall
	as partitioned belt with corrugated sidewalls for steep angle trans-port	with transverse partitions or with corrugated sidewalls	
Troughed belt	for all bulk materials various trough designs	with fins or partitions or with patterned cover for steep angle transport	transport of billets or tree trunks, reels and barrels to prevent lateral rolling
	Trough belt	with turned-up sides	construction as for bulk materials

The design of multiple troughs is determined essentially by the troughing angle.

Troughing angle of multiple belt supports

Troughing	Top run	Return run
V-trough	for belt widths up to 800 mm. Troughing angle up to 30° depending on belt construction. Belt widths up to 1200 mm and troughing angle up to 45° in special cases. With stepped fabric plies at belt centre where appropriate	in any width required for better tracking; standard troughing angle 10° – 15°
3-part	Classical version for all belt widths; standard troughing angle: 20° – 30° – 35° – 40° – 45° “Deep trough” with shortened centre idler	with double strand, material and passenger transport with the conventional top run troughings
5-part	In top run, preferred as close-set garland idler assembly in the material feeding area. Troughing angle dependent on load distribution, belt rigidity and belt tension: 25°/55° or 30°/60°	

Troughing designs

(The clearance d must not exceed 10 mm; from 2000 mm belt width it must not exceed 15 mm.)

Standard idler tube lengths in mm

Troughing design		Belt width B in mm																	
		300	400	500	600	650	800	1000	1200	1400	1600	1800	2000	2200	2400	2600	2800	3000	3200
flat	l	380	500	600	700	750	950	1150	1400	1600	1800	2000	2200	2500	2800				
V-trough	l	200	250	315	340	380	465	600	700	800	900	1000	1100	1250	1400	1500	1600	1700	1800
3-part	l		160	200	250	250	315	380	465	530	600	670	750	800	900	950	1050	1120	1150
3-part	l_1						200	250	315	380	465	530	600	640	670	700	800	900	900
(deep trough)	l_2						380	465	550	600	670	700	800	900	1000	1100	1150	1150	1250
5-part	l						165	205	250	290	340	380	420	460	500	540	580	640	670

When selecting the idler diameter, care should be taken that no excessively high numbers of revolutions result from the belt speed. 600 – 700 r. p. m. should not be exceeded.

Standard idler diameter in mm

Carrier idlers				88.9	108	133	159	193.7
Impact idlers				156	180	215	250	290
Return run support discs	120	133	150	180	215	250	290	

Idler speed

$$n_R = \frac{v \cdot 60}{\pi \cdot D_R}$$

in min^{-1} D_R in m

The economic efficiency of a belt conveyor can be considerably influenced by the selection of an optimum idler spacing in the top and the return run. The capital expenses and the maintenance expenditure are also reduced by a smaller number of idlers. Approximate standard values of the idler spacing lo in the top run can be calculated from the empirical relationship

$$l_0 \leq 5 \cdot (k \cdot \rho)^{-0.2}$$

in m

max. grain size k in mm
bulk density ρ in t/m^3

The idler spacing in the return run should be about 2 or 3 X l_O.

A detailed investigation, e. g. towards graduating the idler spacings when centre distances are greater, may be advisable and can be undertaken at any time by the staff of the ContiTech Transportbandsysteme GmbH Application Technique Department.

* The following effects should be observed in particular: Loading capacity of the idlers (Service life of the idler depending on bearing load and axial deflection)
Troughing properties of the belt (Safe material intake and adequate belt support)
Drive traction and belt sag (cf. D – 4.1) (Prevention of shear vibrations of the belt and excessive motional resistances due to higher flexing stress)

D – 2.5 Conveying capacity

The Conveying capacity of the belt conveyor is determined by the filling cross-section area A and the conveying speed v. The theoretical volume capacity thus amounts to

$$Q_{V\,th} = A \cdot v \cdot 3600 \qquad \text{in } m^3/h$$

and the theoretical capacity to

$$Q_{m\,th} = A \cdot v \cdot 3600 \cdot \rho \qquad \text{in t/h}$$

$$A = A_1 + A_2$$

in m^2

The filling cross-section area A is based on the effective belt width

$$b = 0.9 \cdot B - 50 \text{ mm für } B \leq 2000 \text{ mm}$$
$$b = B - 250 \text{ mm für } B > 2000 \text{ mm}$$

For belts with corrugated sidewalls the maximum effective belt width amounts to

$$b_{eff} = B - 100 \text{ mm for a corrugated sidewall height } h_K \leq 80 \text{ mm}$$
$$b_{eff} = B - 120 \text{ mm for a corrugated sidewall height } h_K > 80 \text{ mm}$$

which is reduced by the corresponding amount for a plain skirting zone. The effective filling height h_n is approx. $0.9 \cdot h_K$.

With feeding rate φ_1 the effective Conveying capacity amounts to

$$Q_{V\,eff} = A \cdot v \cdot 3600 \cdot \varphi_1 \qquad \text{in } m^3/h$$

or

$$Q_{m\,eff} = A \cdot v \cdot 3600 \cdot \rho \cdot \varphi_1 \qquad \text{in t/h}$$

The deviation of the filling cross-section form from the straight-side fill assumed with angle β, which occurs above all in non-horizontal transport, can be taken into account with feeding rate φ_1.

Guide values for φ_1 in non-horizontal transport

Gradient in °	2	4	6	8	10	12	14	16	18	20
H/L	0.035	0.070	0.105	0.140	0.174	0.208	0.242	0.276	0.310	0.342
Feeding rate φ_1	1.0	0.99	0.98	0.97	0.95	0.93	0.91	0.89	0.85	0.81

The values stated apply only to strongly troughed conveyor belts and bulk material with high internal friction. For untroughed belts and V-troughs in particular, non-horizontal transport results in considerably reduced feeding rates φ_1 that have to be calculated by the approximation method* or determined empirically.

* A mathematical formulation is given for instance in DIN 22 101

Gradients exceeding 20° can generally be attained only with conveyor belts with a partitioned or patterned cover. The stated values of φ_1 apply with the same restrictions as above.

Guide values for φ_1 with gradients exceeding 20°

Gradient in °	21	22	23	24	25	26	27	28	29	30
H/L	0.358	0.375	0.391	0.407	0.423	0.438	0.454	0.469	0.485	0.500
Feeding rate φ_1	0.78	0.76	0.73	0.71	0.68	0.66	0.64	0.61	0.59	0.56

Short flights deviating in their gradient from the overall conveying angle of the belt conveyor need not be taken into account with reduction factor φ_1 unless the filling cross-section changes on a brief run-through.

The necessary filling cross-section results from the above equations with a given flow of material Q_m (mass flow in t/h) and taking into account where relevant a uniformity coefficient $\varphi_2 \geq 1$ in feeding, in

$$A = \frac{Q_m \cdot \varphi_2}{v \cdot 3600 \cdot \rho \cdot \varphi_1} \quad \text{in m}^2$$

and thus enables the trough design to be selected from the tables showing the values for A with the corresponding parameters

$$A = \frac{b^2}{4} \cdot \tan\beta \quad \text{in m}^2 \qquad A = \left(l + \frac{h_n}{\tan\lambda}\right) \cdot h_n + \left(\frac{l}{2} + \frac{h_n}{\tan\lambda}\right)^2 \cdot \tan\beta \quad \text{in m}^2$$

Flat

Belt width B in mm	Angle of repose β 5°	10°	15°	20°
300	0.0010	0.0021	0.0032	0.0044
400	0.0021	0.0042	0.0064	0.0087
500	0.0034	0.0070	0.0107	0.0145
650	0.0062	0.0126	0.0191	0.0260
800	0.0098	0.0197	0.0300	0.0408
1000	0.0158	0.0318	0.0483	0.0657
1200	0.0232	0.0467	0.0710	0.0965
1400	0.0320	0.0645	0.0980	0.1332
1600	0.0422	0.0851	0.1294	0.1758
1800	0.0539	0.1086	0.1651	0.2242
2000	0.0669	0.1350	0.2051	0.2786
2200	0.0831	0.1676	0.2547	0.3459
2400	0.1011	0.2037	0.3096	0.4206

Square trough (λ = 80°)

Belt width B in mm	h_k* = 100 mm, Angle of repose β 5°	10°	15°	20°
400	0.0171	0.0183	0.0194	0.0207
500	0.0258	0.0282	0.0306	0.0332
650	0.0397	0.0447	0.0499	0.0554
800	0.0546	0.0633	0.0723	0.0817
1000	0.0759	0.0911	0.1067	0.1231
1200	0.0990	0.1224	0.1466	0.1719
1400	0.1239	0.1573	0.1918	0.2279
1600	0.1505	0.1957	0.2423	0.2911

Belt width B in mm	h_k* = 150 mm, Angle of repose β 5°	10°	15°	20°
650	0.0499	0.0534	0.0569	0.0606
800	0.0717	0.0783	0.0851	0.0922
1000	0.1024	0.1147	0.1273	0.1406
1200	0.1347	0.1545	0.1750	0.1964
1400	0.1689	0.1980	0.2279	0.2594
1600	0.2048	0.2449	0.2863	0.3296

*** $h_n = h_k + 25$ mm**

$$A = b_{eff} \cdot h_{eff} + \frac{b^2_{eff}}{2} \cdot \tan \beta$$ in m^2

Corrugated sidewall

Belt width B in mm	Angle of repose β = 0°				Angle of repose β = 10°			
	Height of corrugated sidewalls h_k in mm				Height of corrugated sidewalls h_k in mm			
	60	80	125	160	60	80	125	160
300	0.0097	0.0129	0.0180	0.0230	0.0111	0.0143	0.0191	0.0241
400	0.0151	0.0201	0.0292	0.0374	0.0185	0.0236	0.0322	0.0404
500	0.0205	0.0273	0.0405	0.0518	0.0268	0.0337	0.0462	0.0575
650	0.0286	0.0381	0.0573	0.0734	0.0410	0.0505	0.0688	0.0849
800	0.0367	0.0489	0.0742	0.0950	0.0571	0.0693	0.0934	0.1142
1000	0.0475	0.0633	0.0967	0.1238	0.0816	0.0974	0.1293	0.1564
1200	0.0583	0.0777	0.1192	0.1526	0.1097	0.1291	0.1687	0.2021
1400	0.0691	0.0921	0.1417	0.1814	0.1413	0.1643	0.2117	0.2514
1600	0.0799	0.1065	0.1642	0.2102	0.1764	0.2031	0.2582	0.3042
1800	0.0907	0.1209	0.1867	0.2390	0.2151	0.2453	0.3082	0.3605
2000	0.1015	0.1353	0.2092	0.2678	0.2573	0.2911	0.3617	0.4203
2200	0.1123	0.1497	0.2317	0.2966	0.3030	0.3404	0.4188	0.4837
2400	0.1231	0.1641	0.2542	0.3254	0.3522	0.3933	0.4794	0.5505

Belt width B in mm	Angle of repose β = 15°				Angle of repose β = 20°			
	Height of corrugated sidewalls h_k in mm				Height of corrugated sidewalls h_k in mm			
	60	80	125	160	60	80	125	160
300	0.0118	0.0151	0.0197	0.0247	0.0126	0.0159	0.0203	0.0253
400	0.0203	0.0254	0.0337	0.0419	0.0222	0.0272	0.0354	0.0435
500	0.0301	0.0370	0.0491	0.0605	0.0336	0.0404	0.0522	0.0636
650	0.0474	0.0569	0.0747	0.0908	0.0541	0.0637	0.0810	0.0971
800	0.0676	0.0799	0.1034	0.1242	0.0787	0.0910	0.1138	0.1346
1000	0.0993	0.1152	0.1462	0.1733	0.1179	0.1338	0.1640	0.1911
1200	0.1364	0.1558	0.1945	0.2279	0.1644	0.1838	0.2214	0.2548
1400	0.1788	0.2019	0.2480	0.2877	0.2182	0.2412	0.2862	0.3258
1600	0.2266	0.2532	0.3070	0.3530	0.2792	0.3058	0.3582	0.4041
1800	0.2797	0.3100	0.3713	0.4236	0.3475	0.3777	0.4374	0.4897
2000	0.3382	0.3721	0.4409	0.4995	0.4231	0.4569	0.5240	0.5826
2200	0.4021	0.4395	0.5160	0.5809	0.5059	0.5434	0.6178	0.6827
2400	0.4713	0.5123	0.5963	0.6675	0.5961	0.6371	0.7190	0.7901

$$A = \frac{b^2}{4} \cdot \cos\lambda \cdot \sin\lambda + \left(\frac{b}{2} \cdot \cos\lambda\right)^2 \cdot \tan\beta$$ in m²

V-trough

Belt width B in mm	Angle of repose β = 0°					Angle of repose β = 10°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
300	0.0038	0.0052	0.0056	0.0059	0.0060	0.0057	0.0068	0.0071	0.0072	0.0071
400	0.0077	0.0104	0.0112	0.0118	0.0120	0.0114	0.0135	0.0141	0.0143	0.0141
500	0.0128	0.0173	0.0187	0.0196	0.0199	0.0190	0.0226	0.0235	0.0238	0.0235
650	0.0229	0.0309	0.0336	0.0352	0.0357	0.0341	0.0404	0.0420	0.0426	0.0420
800	0.0360	0.0485	0.0527	0.0552	0.0561	0.0535	0.0634	0.0660	0.0668	0.0660
1000	0.0580	0.0782	0.0848	0.0889	0.0903	0.0861	0.1020	0.1062	0.1076	0.1062
1200	0.0852	0.1148	0.1246	0.1305	0.1326	0.1265	0.1499	0.1559	0.1580	0.1559
1400	0.1176	0.1584	0.1719	0.1802	0.1830	0.1746	0.2068	0.2152	0.2181	0.2152
1600	0.1552	0.2091	0.2269	0.2378	0.2415	0.2304	0.2730	0.2840	0.2878	0.2840
1800	0.1980	0.2668	0.2895	0.3034	0.3081	0.2939	0.3483	0.3624	0.3671	0.3624
2000	0.2460	0.3315	0.3597	0.3769	0.3828	0.3652	0.4327	0.4503	0.4562	0.4503
2200	0.3055	0.4116	0.4466	0.4680	0.4753	0.4535	0.5373	0.5591	0.5664	0.5591
2400	0.3714	0.5004	0.5429	0.5690	0.5778	0.5513	0.6532	0.6796	0.6886	0.6796

Belt width B in mm	Angle of repose β = 15°					Angle of repose β = 20°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
300	0.0067	0.0076	0.0078	0.0078	0.0076	0.0077	0.0085	0.0086	0.0085	0.0082
400	0.0134	0.0152	0.0156	0.0156	0.0152	0.0154	0.0169	0.0171	0.0169	0.0163
500	0.0223	0.0253	0.0259	0.0259	0.0253	0.0257	0.0282	0.0285	0.0282	0.0272
650	0.0399	0.0453	0.0464	0.0464	0.0453	0.0459	0.0505	0.0510	0.0505	0.0488
800	0.0626	0.0711	0.0729	0.0729	0.0711	0.0721	0.0792	0.0801	0.0792	0.0765
1000	0.1007	0.1145	0.1173	0.1173	0.1145	0.1161	0.1275	0.1289	0.1275	0.1231
1200	0.1479	0.1681	0.1723	0.1723	0.1681	0.1704	0.1872	0.1893	0.1872	0.1808
1400	0.2042	0.2320	0.2377	0.2377	0.2320	0.2352	0.2584	0.2613	0.2584	0.2496
1600	0.2695	0.3062	0.3137	0.3137	0.3062	0.3104	0.3410	0.3449	0.3410	0.3294
1800	0.3438	0.3906	0.4003	0.4003	0.3906	0.3961	0.4350	0.4400	0.4350	0.4202
2000	0.4272	0.4853	0.4973	0.4973	0.4853	0.4921	0.5405	0.5467	0.5405	0.5221
2200	0.5304	0.6026	0.6175	0.6175	0.6026	0.6110	0.6711	0.6788	0.6711	0.6483
2400	0.6448	0.7326	0.7507	0.7507	0.7326	0.7428	0.8156	0.8252	0.8158	0.7881

$$A = \left(l + \frac{b-l}{2} \cdot \cos\lambda\right) \cdot \frac{b-l}{2} \cdot \sin\lambda + \left(\frac{l + (b-l) \cdot \cos\lambda}{2}\right)^2 \cdot \tan\beta \qquad \text{in } m^2$$

3-part

Belt width B in mm	Angle of repose β = 0°					Angle of repose β = 10°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
400	0.0059	0.0084	0.0095	0.0104	0.0112	0.0099	0.0121	0.0130	0.0138	0.0144
500	0.0100	0.0143	0.0161	0.0177	0.0191	0.0166	0.0204	0.0220	0.0232	0.0242
650	0.0187	0.0266	0.0299	0.0328	0.0353	0.0305	0.0374	0.0402	0.0425	0.0443
800	0.0292	0.0415	0.0468	0.0514	0.0552	0.0477	0.0586	0.0630	0.0666	0.0694
1000	0.0482	0.0685	0.0771	0.0845	0.0907	0.0780	0.0958	0.1029	0.1087	0.1131
1200	0.0705	0.1002	0.1128	0.1237	0.1327	0.1143	0.1403	0.1507	0.1592	0.1657
1400	0.0987	0.1401	0.1576	0.1727	0.1852	0.1590	0.1953	0.2097	0.2214	0.2302
1600	0.1312	0.1860	0.2092	0.2291	0.2455	0.2106	0.2587	0.2778	0.2931	0.3047
1800	0.1682	0.2384	0.2680	0.2935	0.3144	0.2694	0.3310	0.3553	0.3749	0.3896
2000	0.2086	0.2957	0.3325	0.3641	0.3901	0.3344	0.4108	0.4410	0.4654	0.4837
2200	0.2635	0.3731	0.4191	0.4584	0.4905	0.4194	0.5153	0.5529	0.5830	0.6052
2400	0.3179	0.4503	0.5061	0.5539	0.5930	0.5076	0.6236	0.6693	0.7060	0.7333
2600	0.3849	0.5446	0.6116	0.6687	0.7152	0.6111	0.7508	0.8054	0.8490	0.8811
2800	0.4501	0.6373	0.7159	0.7831	0.8380	0.7167	0.8805	0.9448	0.9963	1.0344
3000	0.5256	0.7440	0.8356	0.9138	0.9775	0.8356	1.0265	1.1013	1.1611	1.2052
3200	0.6143	0.8682	0.9742	1.0641	1.1368	0.9702	1.1917	1.2778	1.3460	1.3956

Belt width B in mm	Angle of repose β = 15°					Angle of repose β = 20°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
400	0.0119	0.0140	0.0148	0.0155	0.0160	0.0141	0.0160	0.0168	0.0173	0.0177
500	0.0201	0.0236	0.0250	0.0261	0.0269	0.0237	0.0270	0.0282	0.0291	0.0297
650	0.0366	0.0431	0.0456	0.0475	0.0490	0.0431	0.0490	0.0512	0.0528	0.0538
800	0.0574	0.0675	0.0714	0.0745	0.0767	0.0675	0.0768	0.0802	0.0828	0.0844
1000	0.0935	0.1100	0.1163	0.1212	0.1247	0.1097	0.1249	0.1304	0.1344	0.1369
1200	0.1370	0.1612	0.1705	0.1777	0.1828	0.1608	0.1831	0.1911	0.1970	0.2007
1400	0.1903	0.2240	0.2368	0.2467	0.2536	0.2231	0.2540	0.2651	0.2732	0.2781
1600	0.2519	0.2965	0.3134	0.3264	0.3355	0.2951	0.3361	0.3507	0.3613	0.3677
1800	0.3220	0.3791	0.4007	0.4173	0.4287	0.3772	0.4295	0.4482	0.4616	0.4697
2000	0.3998	0.4706	0.4974	0.5181	0.5323	0.4683	0.5333	0.5565	0.5732	0.5833
2200	0.5005	0.5892	0.6224	0.6477	0.6649	0.5854	0.6666	0.6953	0.7155	0.7273
2400	0.6062	0.7136	0.7541	0.7850	0.8062	0.7095	0.8080	0.8429	08678	0.8826
2600	0.7287	0.8579	0.9061	0.9427	0.9673	0.8519	0.9701	1.0117	1.0409	1.0576
2800	0.8553	1.0069	1.0638	1.1071	1.1365	1.0005	1.1394	1.1884	1.2232	1.2434
3000	0.9966	1.1733	1.2394	1.2896	1.3235	1.1654	1.3271	1.3841	1.4243	1.4475
3200	1.1551	1.3597	1.4356	1.4925	1.5300	1.3489	1.5359	1.6009	1.6460	1.6709

$$A = \left(l_1 + \frac{b - l_1}{2} \cdot \cos\lambda\right) \cdot \frac{b - l_1}{2} \cdot \sin\lambda + \left(\frac{l_1 + (b - l_1) \cdot \cos\lambda}{2}\right)^2 \cdot \tan\beta \qquad \text{in m}^2$$

3-part (deep trough)

Belt width B in mm	Angle of repose β = 0°					Angle of repose β = 10°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
1000	0.0545	0.0764	0.0853	0.0925	0.0980	0.0837	0.1025	0.1095	0.1147	0.1180
1200	0.0795	0.1116	0.1246	0.1353	0.1435	0.1225	0.1501	0.1604	0.1681	0.1732
1400	0.1092	0.1534	0.1713	0.1861	0.1976	0.1685	0.2066	0.2208	0.2316	0.2388
1600	0.1423	0.2001	0.2238	0.2435	0.2590	0.2207	0.2708	0.2897	0.3042	0.3142
1800	0.1811	0.2548	0.2851	0.3102	0.3300	0.2813	0.3451	0.3693	0.3878	0.4006
2000	0.2242	0.3156	0.3532	0.3845	0.4092	0.3487	0.4279	0.4580	0.4812	0.4972
2200	0.2812	0.3953	0.4420	0.4807	0.5109	0.4355	0.5341	0.5713	0.5997	0.6190
2400	0.3455	0.4850	0.5416	0.5883	0.6243	0.5327	0.6529	0.6978	0.7317	0.7542
2600	0.4162	0.5834	0.6510	0.7063	0.7486	0.6395	0.7832	0.8365	0.8763	0.9022
2800	0.4854	0.6815	0.7612	0.8269	0.8777	0.7488	0.9178	0.9811	1.0289	1.0607
3000	0.5597	0.7867	0.8795	0.9564	1.0164	0.8665	1.0627	1.1367	1.1931	1.2314
3200	0.6531	0.9161	1.0227	1.1103	1.1776	1.0053	1.2316	1.3160	1.3793	1.4209

Belt width B in mm	Angle of repose β = 15°					Angle of repose β = 20°				
	Troughing angle λ					Troughing angle λ				
	20°	30°	35°	40°	45°	20°	30°	35°	40°	45°
1000	0.0989	0.1161	0.1221	0.1262	0.1284	0.1148	0.1303	0.1353	0.1383	0.1394
1200	0.1448	0.1701	0.1789	0.1851	0.1886	0.1682	0.1910	0.1984	0.2030	0.2048
1400	0.1994	0.2343	0.2466	0.2552	0.2602	0.2317	0.2632	0.2735	0.2800	0.2826
1600	0.2615	0.3075	0.3240	0.3358	0.3429	0.3042	0.3460	0.3598	0.3688	0.3729
1800	0.3333	0.3919	0.4130	0.4282	0.4373	0.3878	0.4411	0.4588	0.4704	0.4757
2000	0.4134	0.4862	0.5125	0.5314	0.5430	0.4812	0.5474	0.5695	0.5841	0.5909
2200	0.5157	0.6063	0.6386	0.6616	0.6752	0.5997	0.6818	0.7090	0.7264	0.7341
2400	0.6300	0.7401	0.7790	0.8062	0.8217	0.7319	0.8316	0.8640	0.8843	0.8924
2600	0.7555	0.8870	0.9329	0.9647	0.9820	0.8771	0.9958	1.0340	1.0573	1.0657
2800	0.8857	1.0406	1.0954	1.1338	1.1558	1.0291	1.1694	1.2151	1.2438	1.2555
3000	1.0260	1.2061	1.2703	1.3161	1.3431	1.1931	1.3564	1.4103	1.4450	1.4601
3200	1.1882	1.3956	1.4683	1.5191	1.5474	1.3800	1.5674	1.6280	1.6656	1.6799

$$A = \left(l_1 + l_2 \cdot \cos\lambda_1\right) \cdot l_2 \cdot \sin\lambda_1$$

$$+ \left[l_1 + 2l_2 \cdot \cos\lambda_1 + \left(\frac{b - l_1 - 2l_2}{2} \right) \cdot \cos\lambda_2 \right] \cdot \frac{(b - l_1 - 2l_2) \cdot \sin\lambda_2}{2}$$

$$+ \left[\frac{l_1}{2} + l_2 \cdot \cos\lambda_1 + \left(\frac{b - l_1 - 2l_2}{2} \right) \cdot \cos\lambda_2 \right]^2 \cdot \tan\beta \qquad \text{in m}^2$$

5-part

Belt width B in mm	Troughing angle λ_1 / λ_2 25°/55°			
	Angle of repose β			
	0°	10°	15°	20°
1000	0.0958	0.1181	0.1297	0.1418
1200	0.1402	0.1730	0.1901	0.2080
1400	0.1949	0.2399	0.2633	02878
1600	0.2541	0.3143	0.3455	0.3783
1800	0.3262	0.4025	0.4421	0.4837
2000	0.4073	0.5016	0.5506	0.6019
2200	0.5106	0.6264	0.6866	0.7497
2400	0.6254	0.7649	0.8374	0.9134
2600	0.7516	0.9171	1.0031	1.0932
2800	0.8892	1.0892	1.1835	1.2890
3000	1.0228	1.2512	1.3698	1.4942
3200	1.1909	1.4498	1.5844	1.7254

Belt width B in mm	Troughing angle λ_1 / λ_2 30°/60°			
	Angle of repose β			
	0°	10°	15°	20°
1000	0.1021	0.1224	0.1329	0.1439
1200	0.1496	0.1794	0.1949	0.2112
1400	0.2076	0.2484	0.2696	0.2918
1600	0.2714	0.3261	0.3546	0.3843
1800	0.3480	0.4172	0.4532	0.4909
2000	0.4340	0.5195	0.5639	0.6104
2200	0.5427	0.6475	0.7020	0.7590
2400	0.6634	0.7895	0.8550	0.9236
2600	0.7960	0.9453	1.0229	1.1042
2800	0.9406	1.1151	1.2058	1.3008
3000	1.0851	1.2914	1.3987	1.5110
3200	1.2595	1.4928	1.6140	1.7410

D – 3 Calculation of belt conveyor

➲ **Masses 3.1**
Idlers 3.1.1
Conveyor belt 3.1.2
Drive elements ..... 3.1.3 – 3.5.1

➲ **Coefficients 3.2**

➲ **Motional resistance 3.3**
Resistive force components 3.3.1
Motional resistances of sections . 3.3.2

➲ **Required power 3.4**

➲ **Peripheral forces 3.5**
Starting and stopping 3.5.1
Multi-pulley drive 3.5.2

➲ **Example of calculation 3.6**

When designing conveyor belts for low-powered conveyor systems, a rough calculation of the belt conveyor may be adequate in many cases. For this purpose it is quite sufficient to work with the standard formulae and guide values given in the main chapters of this section. The guide values listed there can also be used to achieve an initial rough assessment of the belt structure, the data of which can then be applied in the more precise calculation. It is therefore advisable to observe the main chapters of Sections D – 4 (tension members) and D – 5 (covers) prior to the more detailed investigations in the sub chapters.

The technical staff of the ContiTech Transportbandsysteme GmbH guarantee to undertake a meticulous investigation of both simple and extremely intricate applications at any time. The prompt preparation of a quotation can be facilitated considerably by use of the questionnaire attached in the appendix (D – 9).

An essential step in conveyor belt calculation is the determination of the power requirements from the overall motional resistances, taking into account where relevant temporary operating conditions such as starting, stopping or section loading in systems with varying gradients.

Motional resistances of a belt conveyor

1 Rolling resistance of idlers
2 Belt flexing resistance
3 Flexing resistance of material conveyed
4 Feeding resistance
5 Skirting friction resistance
6 Scraper friction resistance
7 Belt deflection resistance at pulleys
8 Gradient resistance

The different components of the motional resistance can in general be calculated comprehensively by means of empirical coefficients. In special cases, however, it may be advisable to ascertain the essential components of the motional resistance individually.

The size of the drives and their quantity, if the power is to be generated by more than one drive, can be determined from the required power. A statement on the optimum arrangement too can be made as soon as the motional resistances have been ascertained. When the number of drives and their arrangement have been stipulated, the peripheral forces acting on the drive pulleys and having a decisive effect on the extent of the belt tensions can be determined.

D–3.1 Masses*

In order to calculate the motional resistance, the sum total of all masses moved on the flight must be determined. These include not only the load mass but also the belt mass and the rotating idler parts. A more precise investigation of the intermittent operating conditions in starting and stopping furthermore involves reducing all rotating masses, including the masses of rotating drive parts, to the pulley circumference.

The mass m_L^I of the material handled, burdening one meter of the flight, amounts from the capacity and the conveying speed to

$$m_L^I = \frac{Q_m}{3.6 \cdot v} \qquad \text{in kg/m}$$

If transport is effected on the top and return runs simultaneously, the mass m_L^I is to be ascertained from the sum total of the two capacities.

Mass flow Q_m in t/h

In cases where the belt structure has not yet been assessed or further data on the intended design of the conveying system are not yet available, guide values for the mass m_G^I of the belt and m_R^I of the rotating idler parts can be inserted according to the loading of the system.

Guide values for the mass ($2\ m_G^I + m_R^I$) of the moving parts of a belt conveyor in kg/m

B in mm	300	400	500	650	800	1000	1200	1400	1600	1800	2000	2200	2400	2600	3000	3200
light systems	8.5	15.0	18.5	22.5	30	37	54	63	81	106	118	128	140	161	180	200
standard system	10	16.5	21	25.5	35	52	77	89	130	154	192	207	242	256	302	345
heavy-duty system	13.0	18	24	28.5	40	67	100	115	179	202	266	287	344	371	425	490

* In general usage the term “weight” can also be used for “mass”. In all instances, however, it Is only the unit “kilogram” (kg) that is admissible.

D – 3.1.1 Idlers

In order to ascertain with precision the mass of all moving idler parts, related to 1 m of the centre distance, the mass of one idler or idler trough set is to be taken from the table below and to be divided by the respective idler spacing l_o or l_u. If the idler spacings are graduated, the average idler spacing is to be taken as a basis.

$$m_R^I = \frac{m_{Ro}}{l_o} + \frac{m_{Ru}}{l_u} \qquad \text{in kg/m}$$

Guide values for masses of rotating parts of idlers (trough assemblies)

Belt width B in mm	Idler diameter in mm	Mass in kg			
		1-part	V-trough	3-part	5-part
300	88.9	3.2	4.1		
400	88.9	3.9	4.7	5.4	
	108	5.9	6.6	7.3	
	133	7.6	8.7	9.6	
500	88.9	4.5	5.5	6.1	
	108	6.6	7.8	8.4	
	133	8.9	10.4	11.1	
650	88.9	5.5	6.3	7.0	
	108	8.0	9.0	9.8	
	133	10.8	12.1	13.1	
800	88.9	6.7	7.4	8.3	9.0
	108	9.8	10.6	11.6	12.4
	133	13.3	14.2	15.6	16.3
1000	108	11.7	13.2	13.6	14.2
	133	15.9	17.8	18.2	18.9
	159	21.9	24.7	26.3	28.0
1200	108	14.2	15.0	16.3	16.3
	133	19.3	20.5	22.3	21.7
	159	26.1	28.0	29.8	31.9
1400	133	21.8	23.3	25.0	24.3
	159	29.3	31.6	35.5	35.0
1600	133	25.1	26.5	28.0	28.5
	159	33.4	35.0	38.7	39.3
1800	133	27.6	29.1	30.7	31.5
	159	37.8	39.5	42.4	42.5
2000	133	30.2	31.8	33.3	33.5
	159	40.2	43.3	47.0	46.5
	193.7	69.1	76.4	80.1	89.5
2200	159	46.5	49.0	50.1	49.5
	192.2	77.8	82.6	93.2	95.5
2400	159	50.7	51.5	53.5	53.0
	193.7	86.6	91.4	93.2	100.5
2600	159		55.1	57.5	56.5
	193.7		97.2	97.6	107.0
2800	159		58.5	59.1	60.0
	193.7		103.0	106.4	113.0
3000	159		63.0	65.5	65.0
	193.7		109.0	112.5	121.5
3200	159		70.0	71.5	68.0
	193.7		120.0	123.0	126.5

Idlers with supporting discs are frequently used in the return run in order to keep the system clean.

Masses of rotating parts of return run idlers with supporting discs in kg

Belt width B in mm	Tube diameter in mm	Disc diameter in mm	1-part	2-part
400	51	120	4.0	5.0
500	57	133	5.7	6.8
650	51	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	180 215	31.9 42.0	33.9 44.5
1800	108	180 215	34.3 44.9	36.3 47.3
2000	198	100 215	31.3 48.8	39.3 51.8
2200	133	215 250	59.8 73.8	62.8 76.8
2400	133	215 250	62.4 77.5	67.2 82.3
2600	133	215 250	68.7 84.9	71.7 87.9
2800	159	290	130.6	138.2
3000	159	290	138.4	146.3
3200	159	290	146.2	154.4

In the case of heavy-duty systems with centre distances below 80 m, the masses of the feeding idlers that may be fitted with impact rings are to be given special consideration where applicable.

Masses of rotating parts of feeding idlers with impact rings

Belt width B in mm	Tube diameter in mm	Impact idler diameter in mm	Mass in kg	
			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
2400	159	250	117.5	127.5
2600	159	250	127.3	137.5
2800	193.7	290	201.0	221.0
3000	193.7	290	214.0	234.0
3200	193.7	290	230.0	252.0

As the mass of the rotating idler parts is concentrated mainly in the idler tube, the idler mass reduced to the belt speed amounts approx. to

$$m^{I}_{R\,red} = 0.9 \cdot m^{I}_{R}$$ in kg/m

and

$$m_{R\,red} = m^{I}_{R\,red} \cdot L$$ in kg

D–3.1.2 Conveyor belt

The mass of the conveyor belt results from the mass $\mathbf{m_Z^{II}}$ of the tension member and the mass mo of the covers. If the type and structure of the conveyor belt are not stated, the belt construction must first be assessed. For this purpose an estimated calculation can be executed as described earlier, using the standard formulae and guide values stated in the main chapters.

Guide values to determine the tension member mass
Strength grade of belt in **N/mm**

Mass $\mathbf{m_Z^{II}}$ of tension member (belt core) in $\mathbf{kg/m^2}$

The mass $\mathbf{m_D^{II}}$ depends on the density of the cover compound and on the thickness of the top and bottom covers. For standard applications, the mass with cover thickness s_2 or s_3 in mm amounts approx. to

$$\mathbf{m_D^{II} = \rho_D \cdot (s_2 + s_3)} \qquad \text{in kg/m}^2$$

A density of ρ_D **= 1.1 kg/dm³** can be assumed in cover rubber compounds for standard application.

materials designed for the handling of material containing oil, for hot materials or for application in underground mining. The precise values are stated in Section D–5.1.

The belt weight m_G^I with B in m consequently amounts to

$$\mathbf{m_G^I = (m_Z^{II} + m_D^{II}) \cdot B} \qquad \text{in kg/m}$$

For conveyor belts with a special structure and for special applications, the precise data are to be obtained from the manufacturer. If it becomes apparent in the course of continued calculation or designing that a deviating belt structure has to be selected, the value of m_G^I is to be corrected where applicable.

D–3.1.3 Drive elements

The masses of the drive elements such as motor armature, clutch, brakes, gear mechanism and pulleys are not taken into account in the calculation of the motional resistance of a belt conveyor.
The operating conditions on starting and stopping (D–3.5.1) may, however, be considerably influenced by the masses as these also have to be accelerated or decelerated by the starting or braking torque. The mass moments of inertia must be reduced to the pulley circumference and, if they attain a significant quantity in comparison with the other moving masses, must be taken into account in the calculation of the peripheral forces.

The mass of a drive element rotating at speed n in min^{-1} reduced to the pulley circumference rotating at belt speed v is

$$m_{red} = J \cdot \left(\frac{n}{v}\right)^2 \cdot \left(\frac{2\pi}{60}\right)^2 \qquad \text{in kg}$$

with the mass moment of inertia $J = GD^2/4g$ in kgm^2. In obsolescent catalogues GD^2 is frequently still stated as a so-called “flywheel effect” in kpm^2.

If the masses of the individual drive elements cannot be reduced separately, the mass $\mathbf{m_{E\,red}}$ of a drive pulley with clutches and possibly with a brake results approximately from the mass moment of inertia of the motor $\mathbf{J_M}$, the motor speed $\mathbf{n_M}$ and a standard factor $\mathbf{K_M}$

$$m_{E\,red} = K_M \cdot J_M \cdot \left(\frac{n_M}{v}\right)^2 \cdot \left(\frac{2\pi}{60}\right)^2 \qquad \text{in kg}$$

Rotational inertia coefficient $\mathbf{K_M}$

	Slipring motor	Squirrel-cage motor
Drive with brake	0.58	1.2
Drive without brake	0.28	0.65

Guide values of mass moments of inertia for electromotors with a rated speed of 1000 min^{-1}

D – 3.2 Coefficients

The resistance coefficient f, resulting from a correlation between the weights and the motional resistances, yields the primary resistances F_H of the belt conveyor. The f-values take into account the sum of the motional resistances in the top run and the return run and are applicable approximately in the range from 70% to 110% rated load.

Guide values for f^+

Conveying systems ascending, horizontal or slightly descending transport (motor-driven)	f with v in m/s					
	1	2	3	4	5	6
Standard version Handling of material with average internal friction	0.016	0.0165	0.017	0.018	0.02	0.022
Well laid-out and with smoothly rotating idlers. Handling of material with low internal friction	0.0135	0.014	0.015	0.016	0.017	0.019
Unfavourable operating conditions. Handling of material with high internal friction	0.023 to 0.027					

The guide values of f^+ include a safety margin for the design as the actual motional resistances are dependent on numerous factors, some of which cannot be precisely determined. A safety margin is also to be taken into account in the designing of braked systems for descending transport. In this case, however, the resistance coefficient is to be assumed to be not larger but correspondingly smaller.

Guide values for f^-

Conveyor systems for steeply descending transport (generator-induced braking)	f
Well laid-out conveying systems with normal operating conditions. Handling of material with low to medium internal friction	0.012 to 0.016

Special account can be taken of certain limiting quantities. The guide values stated are based on a minimum belt sag of approx. 1 % so that the f-values can be reduced where applicable when the sag: idler spacing ratio is low, i. e. when the belt tensions are high. Overdimensioned idler diameters also lead to lower f-values.

Quantatively the influence of the outdoor temperature can be taken more exactly into consideration, provided that f-values not deviating considerably from the guide values result from the multiplication of f by factor C_T.

Influence of outdoor temperature

Temperatur in °C	+20	+10	+0	-10	-20	-30
Factor c_T	1	1.01	1.04	1.10	1.16	1.27

The secondary resistances F_N are determined by the length coefficient

$$C = \frac{F_H + F_N}{F_H}$$

the accuracy of which, however, is adequate only for conveying systems with centre distances or conveying lengths exceeding 80 m.

Length coefficient C* depending on conveying length L

L in m	C	L in m	C	L in m	C	L in m	C
3	9.0	80	1.92	250	1.38	700	1.14
4	7.6	90	1.86	300	1.31	800	1.12
6	5.9	100	1.78	350	1.27	900	1.10
10	4.5	120	1.70	400	1.25	1000	1.09
16	3.6	140	1.63	450	1.22	1500	1.06
20	3.2	160	1.56	500	1.20	2000	1.05
25	2.9	180	1.50	550	1.18	2500	1.04
32	2.6	200	1.45	600	1.17	5000	1.03
40	2.4						
50	2.2						
63	2.0						

Coefficient **C***

* For conveying lengths below 80 m and In individual cases for relatively high secondary resistances, it is advisable to check and, where relevant, to correct coefficient C in the light of the secondary resistance components.

D – 3.3 Motional resistance

The motional resistance F of a belt conveyor is calculated with the coefficients C and f.

$$F = C \cdot f \cdot L \cdot \left[m'_R + \left(2 \cdot m'_G + m'_L\right) \cdot \cos\delta \right] \cdot g + H \cdot m'_L \cdot g \qquad \text{in N}$$

In the case of conveying systems with slighter gradients ($\delta < 18°$) $\cos\delta$ can be equated with I as an approximation.

The motional resistance F consists of various components.

Primary resistance F_H

$$F_H = f \cdot L \cdot \left[m'_R + \left(2 \cdot m'_G + m'_L\right) \cdot \cos\delta \right] \cdot g \qquad \text{in N}$$

The primary resistance is that resistive force component that occurs in the top run and in the normally unloaded return run, irrespective of conveying height H. It is equivalent to a frictional force.

Secondary resistance F_N*

$$F_N = \left(C - 1\right) \cdot F_H \qquad \text{in N}$$

The secondary resistance is due mainly to frictional and acceleration forces in the feeding area and can generally be expressed with sufficient accuracy by the coefficient C as a fraction of the primary resistance for belt conveyors with a conveying length in excess of 80 m.

* When ascertaining the secondary resistances precisely, it is above all the following components that have to be taken into account:

Acceleration resistance FNa at the feeding point

$$F_{Na} = \frac{Q_m}{3.6} \cdot \left(v - v_o\right) \qquad \text{in N}$$

In this equation V_o is the relative speed of the material handled on feeding.

Frictional resistance F_{Nsch} at the feeding chute

$$F_{Nsch} = 2000 \cdot v^2 \cdot h^2_{sch} \cdot \rho \qquad \text{in N}$$

In this equation h_{sch} is the height in m of the material filling between the chute sidewalls.

The frictional resistance of belt cleaning equipment can be taken into account by

$$F_{Nr} \approx 1500 \cdot B \qquad \text{in N}$$

with belt width B in m. This component, however, does not normally occur in concentrated form at the feeding point.

The precise determination of F_N is thus based on

$$F_N = F_{Na} + F_{Nsch} + F_{Nr}$$

Gradient resistance F_{St}

$$F_{St} = H \cdot m_L^I \cdot g \qquad \text{in N}$$

Considering the belt conveyor as a whole, the gradient resistance results solely from the drop forces of the load masses, as the forces from the downward pressure of the conveyor belt (top run against return run) are counterbalanced.

As a further component of motional resistance F, special resistances F_S may occur; these are due, for example, to tilted idlers* skirtboards located on the flight or to scrapers for discharging the material conveyed. Their magnitude can be ascertained in individual cases from data in technical literature.

With the stated components, the overall motional resistance becomes

$$F = F_H + F_N + F_{St} + F_S$$

and must be overcome in operation by a peripheral force F_U generated at one or more than one drive pulley.

The conveying height H is to be inserted with a positive sign for ascending transport and with a negative sign for descending transport. When the gradient of the conveying system is slight, a prediction cannot always be made in the latter case as to whether a positive or a negative overall motional resistance will result. The calculation must then be executed with the selected values $f^{(+)}$ and $f^{(-)}$ respectively. The largest numerical value then states the decisive (positive or negative) motional resistance for further calculation.

In a borderline case, operation of the belt conveyor with no load may lead to the maximum motional resistance. A calculation of the motional resistances in no-load operation too is therefore essential for low-gradient downhill conveying systems.

The layout of the belt conveyor in accordance with this schedule can be executed by means of the table used in the calculation example in Section D–3.6 (cf. also D–9.3). If a precise investigation of the belt tension is to be made in the further course of the calculation, possibly with starting and stopping taken into account, the motional resistance can no longer be considered totally but must be allotted to top run and return run according to its origins.

It is also in the investigation of belt conveyors with an alternating flight gradient or varying section loads that the motional resistances have to be determined separately and allocated locally to the individual sections of the conveying system in order to ascertain the exact belt tensions (D–3.3.2).

* According to Vierling, the tilt resistance for 3-part (roughing sets In the top run and 2-part (roughing sets in the return run can be determined as follows (with μ = 0 3 to 0.4):

$$F_{So} = L \cdot c \cdot \mu \left(m_G^I + m_L^I \right) \cdot g \cdot \cos\delta \cdot \sin\varepsilon \qquad \text{in N} \qquad \text{and}$$

$$F_{Su} = L \cdot \mu \cdot m_G^I \cdot g \cdot \cos\lambda \cdot \cos\delta \cdot \sin\varepsilon \qquad \text{in N}$$

In these equations, the trouhing factor is c = 0.4 for 30°-troughing, c = 0.3 for 45°-troughing and ε the angle of tilt.

D – 3.3.1 Resistive force components

Components of the primary resistance $\mathbf{F_H}$ occur in the top run and in the normally unloaded return run of the belt conveyor. The resistance coefficients $\mathbf{f_o}$ or $\mathbf{f_U}$ to be inserted for this purpose can be set at approximately equal to f.

$$F_{Ho} = f_o \cdot L \cdot \left[m_{Ro}^{\prime} + \left(m_G^{\prime} + m_L^{\prime}\right) \cdot \cos\delta\right] \cdot g \qquad \text{in N}$$

$$F_{Hu} = f_u \cdot L \cdot \left(m_{Ru}^{\prime} + m_G^{\prime} \cdot \cos\delta\right) \cdot g \qquad \text{in N}$$

with

$$f_o \approx f_u \approx f$$

For loading states deviating considerably from the rated load or from noload operation of the belt, the values of f can be corrected in accordance with an empirically determined correlation.

If the return run is also used for transport, the load mass $\mathbf{m_{LU}}$ is to be taken correspondingly into account in component $\mathbf{F_{HU}}$. The secondary resistance $\mathbf{F_N}$ is determined separately from the individual components or calculated with coefficient C from the **primary resistance FH**

$$F_N = \left(C - 1\right) \cdot F_H$$

and can be allocated locally to the feeding point with sufficient accuracy. The gradient resistances are also to be considered separately for top run and return run, so that

$$F_{Sto} = H \cdot \left(m_G^{\prime} + m_L^{\prime}\right) \cdot g \qquad \text{in N}$$

and

$$F_{Stu} = -H \cdot m_G^{\prime} \cdot g \qquad \text{in N}$$

now results with regard to belt mass $\mathbf{m_G^{\prime}}$. In this case too the load mass $\mathbf{m_{LU}^{\prime}}$ is to be taken into account where relevant in component $\mathbf{F_{Stu}}$ if the return run is used for transport.

The tabular schedule shown in the calculation example can be used in allocating the motional resistances. F_{HO} and F_{Sto} or F_{HU} and F_{Stu} are combined in the schedule.

$$F_o = f_o \cdot L \cdot \left[m_{Ro}^{\prime} + \left(m_G^{\prime} + m_L^{\prime}\right) \cdot \cos\delta\right] \cdot g + H \cdot \left(m_G^{\prime} + m_L^{\prime}\right) \cdot g \qquad \text{in N}$$

and

$$F_u = f_u \cdot L \cdot \left(m_{Ru}^{\prime} + m_G^{\prime} \cdot \cos\delta\right) \cdot g - H \cdot m_G^{\prime} \cdot g \qquad \text{in N}$$

D – 3.3.2 Motional resistances of sections

If the motional resistances, the required power and the belt tensions of a belt conveyor with a layout including sections with varying gradients or loads are to be determined, a division into section lengths L_N and corresponding section heights H_N is undertaken in order to obtain more accurate results. Those sections too that result from variable feeding or discharge points are also to be distinguished in this process.

In order to facilitate recognition of critical operating conditions, those section lengths that have an approximately constant gradient or at least run at a continuous upward or downward inclination are selected. Initial feeding of the belt conveyor or interruptions in transport may result in loading states that lead to higher or lower motional resistances in conveying systems with such layouts than when loading is continuous. In addition, the motional resistances at vertices or saddle points of the system layout may lead to critical belt tensions that do not emerge from a calculation with the overall conveying height H. This calculation, however, must nevertheless be executed in order to determine whether the conveying system is to be regarded as braked or motor-driven in normal operation with continuous loading. In accordance with this, either the large or the small f-value is to be inserted uniformly in the subsequent calculation of the motional resistances by sections.

The motional resistance in the top run or return run of a section n results with primary resistance F_{HN} and gradient resistance F_{Stn} as

$$F_n = f \cdot L_n \left[m_{R\,n}' + \left(m_G' + m_{Ln}' \right) \cdot \cos \delta_n \right] \cdot g + H_n^* \cdot \left(m_G' + m_{Ln}' \right) \cdot g \quad \text{in N}$$

Any special resistances present must be taken into account additionally. The same f-value can be assumed here too for the loaded or unloaded belt if no data of greater accuracy are available.

The secondary resistances F_N amounting from the calculation executed above to

$$F_N = \left(C - 1 \right) \cdot F_H$$

or needing to be determined in special cases from the individual components, can be added with sufficient accuracy to the section at the beginning of which the material is loaded onto the belt.

The chart shown in chapter D – 3.6 can be used for the systematic recording of all calculated values. It is advisable to calculate first the values of the sections for the normal loading state and for unloaded operation, beginning with the return run in the belt driving direction, as the critical operating instance can frequently not be recognized from the outset. For this reason the moving masses required for calculation of intermittent operating states are to be determined at this point.

$$m_n = \left(m_{R\,n}' + m_G' + m_{L\,n}' \right) \cdot L_n \quad \text{in kg}$$

If the reduced masses m_{Ered} of the drive elements or pulleys are not negligible in relation to the moving masses of the flight, they are to be recorded at the point at which they are effective.

* For sections of the return run, H_n is to be inserted with an inverse sign (negative for ascending transport).

Apart from the normal case, in which the conveying flights are continuously loaded, further operating states arise from the section loads that may be critical for design and that lead to the largest or smallest (i. e. negative) motional resistances. The respective loading states are identified as follows.

a	continuously loaded
b	continuously non-loaded
c	transport starting
d	transport ending

Whilst these loading states are to be taken into account in all instances, there is a slight probability of extreme operating conditions occurring through accidental interruptions in the feeding of the conveyor. Such cases, that might lead to considerably increased dimensioning, can be prevented with appropriate control systems.

e	loaded on uphill gradient
f	loaded on downhill gradient

Low-gradient sections with a positive motional resistance even when loaded are to be regarded as loaded in case e and as unloaded in case f.

The maximum motional resistance to be taken into account in the design results from the sum total of the motional resistances.

$$F_{max} = F_N + F_n + F_{(n+1)} + F_{(n+2)} + \ldots$$

$\mathbf{F_N = 0}$ is to be set in the consideration of the critical case with the smallest, i. e. negative motional resistances (mainly loading of downhill sections).

D – 3.4 Required power

The power required by a belt conveyor that has to be transmitted to the conveyor belt by one or simultaneously by more than one drive pulley amounts from the motional resistance F to

$$P_{Tr} = \frac{F}{1000} \cdot v \qquad \text{in kW}$$

The power capacity P_M to be installed for driven conveying systems (with positive F) results from the power required P_{Tr} with the mechanical efficiency η

$$P_M = P_{Tr} / \eta^+$$

for braked conveying systems (with negative F)

$$P_M = P_{Tr} \cdot \eta^-$$

Guide values for mechanical efficiencies

Efficiency	Driven conveying system $\eta+$ Single-pulley drive	Driven conveying system $\eta+$ Multiple pulley drive	Braked conveying system η^-
Drive with pulley motor	0.96		
Drive via secondary transmission	0.94	0.92	
via secondary transmission and fluid clutch	0.9	0.85	0.95 – 1.0
with pump/hydraulic motor	0.86	0.80	

Whilst the mechanical efficiency leads to the power to be installed P_M having to be greater in the case of a positive power P_{Tr} on the pulley, braked conveying systems can possibly be provided with a drive whose output P_M is equal to or slightly lower than the required power P_{Tr} on the drive pulley. For safety reasons a somewhat higher efficiency factor is assumed on braking.

The power capacity of the motors to be provided for the drive is furthermore influenced by the ambient temperature (coolant temperature) and the altitude above mean sea level (coolant density). The instructions of the specific manufacturer should be observed. For air-cooled motors, standard conditions imply a coolant temperature of between 30 and 40°C and installation altitudes of up to 1000 m above mean sea level. Where relevant, special requirements concerning starting-up of the system or extreme power peaks due to section loading are to be taken into account in selection of the motor construction.

Selection from the rated output values (acc. to DIN 42 973 standards)

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

In order to attain favourable starting and operating properties, the type of motor (squirrel cage rotor or slipring rotor) and of clutch are to be adapted to the specific application. The decisive factor here is starting factor K_A, by which the ratio of maximum starting torque to rated torque is determined, so that the maximum torque on starting amounts to

$$M_{max} = K_A \cdot M_{nom}$$

The magnitude of the rated driving torque is determined by the engine size selected or - when a fluid clutch is used - by the clutch size selected. In order not to generate any unnecessarily high belt tensions on starting, it is advisable to coordinate the motor or clutch size as precisely as possible with the required power P_M.

Guide values for starting factors K_A

Drive	Application	Starting factor
Squirrel cage motor with special-purpose fluid clutches	Large-scale conveying systems	1.2
Slipring motor or thyristor controlled start-up	Medium to large-scale conveying systems	1.2 – 1.4
Squirrel cage motor with fluid clutch	Standard conveying systems (from approx. 30 kW per motor)	1.5
Squirrel cage motor with Y-delta connection	Only for conveying systems starting up unloaded	1.6
Squirrel cage motor with full-voltage starting	Small-scale conveying systems (up to approx. 30 kW)	2.0 – 3.0

The torque required for the breakaway and acceleration of a belt conveyor on start-up, which must normally be somewhat greater than the load torque in operation, can be designated by the so-called breakaway factor.

Guide values for breakaway factor K_I

Downhill conveying systems, long-distance systems, systems starting up unloaded	below 1,3
Standard systems	1.3 – 1.5
Heavy-running systems with high friction	above 1.5

For conventionally designed conveying systems $K_A > K_I$ is to be selected. If, however, a higher power capacity is installed ($P_{M\,nom} > P_M$) for reasons of drive uniformity or owing to only one specific size of drive being available, the starting factor can be determined with the equation

$$K_A \geq K_I \cdot \frac{P_M}{P_{M\,nom}}$$

and can be reduced if necessary by the selection of an appropriate fluid clutch or starting circuit.

In the case of drives with more than one motor, the start-up can be influenced by switching on the individual motors with time delay The start-up of conveying systems is dealt with in detail in Chapter D – 3.5.1.

When considering sections, the rating of the drives to be installed is to be laid out in the light of the maximum required power, i.e. in accordance with the criteria set out in Chapter D – 3.3.2.

$$P_{Tr\,max} = \frac{F_{max}}{1000} \cdot v \qquad \text{in kW}$$

and when F_{max} is positive

$$P_{M\,max} = P_{Tr\,max} / \eta^{+}$$

or when F_{max} is negative

$$P_{M\,max} = P_{Tr\,max} \cdot \eta^{-}$$

In the least favourable case a start-up process may coincide with the motional resistances F_{max}, resulting in the starting factor

$$K_A \geq \frac{(K_I - 1) \cdot P_M + P_{M\,max}}{P_{M\,nom}}$$

D – 3.5 Peripheral force

The peripheral forces occur at the driven or braked drive pulleys and are transmitted there to the belt to overcome all motional resistances acting on the conveyor belt. The peripheral forces can act on the belt at one pulley or distributed over a number of pulleys.

$$F_U = F = \frac{P_{Tr}}{v} \cdot 1000 \qquad \text{in N}$$

The belt tensions on the entry side and the leaving side of the drive pulleys are of decisive significance for further calculation. On the one hand the greater belt tension, generally T_1 on the entry side, affects the tensile strength calculation of the conveyor belt; on the other hand the height of the smaller belt tension, generally T_2 on the leaving side, determines whether power transmission is feasible at all. The target of the calculation is thus to determine the necessary magnitude of the belt tension on the entry side and the leaving side. These belt tensions are the outcome of this chapter.

Regardless of whether it is a driving or braking drive mechanism that is being investigated, whether the operating state concerned is starting or stopping, or whether it is a Multiple pulley drive that is under consideration, the following designation should be retained under all circumstances.

Belt tension at the leaving side of the belt from the last pulley at the head of the conveying system (start of return run)	in operation	T_2
	On Starting	T_{2A}
	On braking	T_{2B}

Setting out from this belt tension that is designated with T_2, T_{2A} oder T_{2B} the level of the further belt tensions is to be determined in the following section D – 4. A design of the tension member in the conveyor belt based on belt tension T_1 or T_{1A} sufficiently accurate only for the standard case of a horizontal or constant-gradient belt conveyor, ignoring stopping and any additional strains in the case of exclusive head drive.

In the case of a driven belt, the peripheral force F_U is positive; thus T_1 is greater than T_2. In the case of negative peripheral forces, i. e. of a braked belt, T_2 is greater than T_1.

For the case of a driven conveyor belt, the belt tensions required for friction transmission of peripheral force F_U result from marginal condition

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

In this formula μ is the friction coefficient between the pulley circumference and the belt surface and the degree of wrap round the pulley in radian measure. When the transmission potential is fully exploited, the necessary belt tension on the leaving side of the belt becomes

$$T_2 = F_U \cdot c_2 \quad \text{with drive factor } c_2 = \frac{1}{e^{\mu\alpha} - 1}$$

The belt tension T_1 at the entry side of the belt becomes

$$T_1 = F_U \cdot c_1 \quad \text{with drive factor } c_1 = 1 + \frac{1}{e^{\mu\alpha} - 1}$$

The peripheral force ist accordingly

$$F_U = T_1 - T_2$$

In the case of a braked conveyor belt, the equation apply according to the retroacting peripheral force.

Values for c_1*

Angle of wrap	Friction coefficient μ								
α in °	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5
170	3.90	2.78	2.23	1.91	1.69	1.54	1.44	1.35	1.29
175	3.80	2.72	2.19	1.87	1.67	1.52	1.42	1.34	1.28
180	3.70	2.66	2.15	1.83	1.64	1.50	1.40	1.32	1.26
185	3.62	2.60	2.10	1.80	1.61	1.48	1.38	1.30	1.25
190	3.55	2.55	2.06	1.77	1.59	1.46	1.36	1.29	1.23
195	3.47	2.50	2.02	1.74	1.56	1.44	1.34	1.28	1.22
200	3.41	2.45	1.99	1.71	1.54	1.42	1.33	1.26	1.21
205	3.32	2.41	1.96	1.69	1.52	1.40	1.31	1.25	1.20
210	3.28	2.36	1.93	1.67	1.50	1.38	1.30	1.24	1.19

If the belt tensions in the conveying direction are designated continuously as T_n and $T_{(n+1)}$, general equations that are also valid for drives at the rear end of the conveying system can be drawn up for the case of full exploitation of the transmission potential.

	Entry side belt tension T_n	Leaving side belt tension $T_{(n+1)}$	Peripheral force	$e^{\mu\alpha}$
Belt driven, positive peripheral force	$T_n = F_U \cdot c_1$	$T_{(n+1)} = F_U \cdot c_2$	$F_U = T_n - T_{(n+1)}$	$\frac{T_n}{T_{(n+1)}}$
Belt braked, negative peripheral force	$T_n = F_U^{**} \cdot c_2$	$T_{(n+1)} = F_U^{**} \cdot c_1$	$F_U = T_n - T_{(n+1)}$	$\frac{T_{(n+1)}}{T_n}$

* Values for c_2 result from $c_2 = c_1 - 1$
** F_U is to be inserted as a positive factor

Friction coefficient μ between drive pulley and rubber conveyor belt (60 to 65 Shore A hardness) with Grimmer

Operating conditions	Steel pulley bare smooth rust-free	Rubber wrapped herringbone grooves hardness: 60 Shore A Lagging surface: smooth Lagging thickness: 8 mm	Polyurethane extruded herringbone grooves hardness: 79 Shore A Lagging surface: lightly corrugated Lagging thickness: 11 mm	Ceramic porous herringbone grooves Lagging thickness: 11 mm
Dry operation	0.35 – 0.4	0.4 – 0.45	0.35 – 0.4	0.4 – 0.45
Clean, wet operation (water)	0.1	0.35	0.35	0.35 – 0.4
Loam- or clay-soiled wet operation	0.05 – 0.1	0.25 – 0.3	0.2	0.35

The friction coefficient μ is also influenced by the surface pressure between the conveyor belt and the pulley. The values stated are valid for a mean surface pressure of 10 N/cm^2.

The law of friction transmission represents an extreme condition that is not precisely fulfilled in every instance. If T_1/T_2 is less than $e^{\mu\alpha}$, the transmission potential of the drive pulley is not fully exploited. Power is transmitted only on the service arc of the angle of wrap α, whereas the non-operation arc represents a reserve for the transmission of higher peripheral forces or for a possible drop in the friction coefficient μ.
The ratio T_1/T_2 produces specific service angles α_n with the value of μ varying.

D–3.5.1 Starting and stopping

Starting and stopping are temporary operating conditions in which the belt velocity changes as a result of a supposedly constant acceleration or deceleration. Apart from the motional resistances of the steady state condition, the inertia forces of the moving masses are to be supplied by peripheral forces F_{UA} tor starting or F_{UB} for stopping (braking). The breakaway resistance component is regarded as negligible in this case.

The peripheral force F_{UA} on starting is stipulated by the size of the motor or clutch selected when the drive is being designed and by starting factor k_A, provided that the starting process is effected by switching on the corresponding drives.

$$\mathbf{F_{UA} = \frac{P_{Mnom} \cdot \eta}{v} \cdot K_A \cdot 1000} \qquad \text{in N}$$

This is based on the fact that the drive is restricted by a control mechanism or slip clutch on acceleration to a torque corresponding to the multiple rated torque determined by starting factor $\mathbf{K_A}$. It is only when the design corresponds precisely to the required power $\mathbf{P_M}$ that $\mathbf{F_{UA} = F \cdot K_A}$.

The expression “sequence starting” can be used when the belt conveyor is driven by a number of motors that are switched on with a time lag. The optimum time sequence is generally found by trial and error. A calculation is possible only when the dynamic processes are taken into account. If start-up is regarded as a quasi-stationary process, on the other hand, the peripheral force $\mathbf{F_{UA}}$ and the resulting drive tractions are determined from the capacity of the motors that are switched on on start-up.

The necessary belt tension results from the marginal value for fractional force transmission with negligible slip

$$\mathbf{T_{2A} = F_{UA} \cdot c_2} \qquad \text{and} \qquad \mathbf{T_{1A} = T_{2A} + F_{UA}}$$

Determination of the friction coefficient μ is generally based on values that provide for a certain safety margin against belt slip, i. e. incomplete utilization of the angle of wrap α. In order to calculate T_{2A}, therefore, a somewhat greater value of μ may be inserted if an increased utilization of the angle of wrap or a temporarily increased slip is permitted.

The acceleration a_A of the belt conveyor amounts from motional resistance F and the sum of the accelerated masses to

$$a_A = \frac{F_{UA} - F}{\left(m_L^I + m_{R\,red}^I + 2 \cdot m_G^I\right) \cdot L} \qquad \text{in m/s}^2$$

Any relatively large reduced masses of rotating drive elements are to be taken into account too. The acceleration is approximately

$$a_A = \frac{F_{UA} - F}{\left(m_L^I + m_{R\,red}^I + 2 \cdot m_G^I\right) \cdot L + m_{E\,red} + m_{red}^*} \qquad \text{in m/s}^2$$

Experimental values for acceleration a_A in m/s²

Type of conveyor system	No load	Full load
Small-scale systems **(L < 50m; B < 800mm)**	0.75	0.5
Medium systems **(50m < L < 500m; 1000mm < B < 1600mm)**	0.6	0.1
Large-scale systems **(L > 500m; B > 1800mm)**	0.5	0.075

The peripheral force F_{UB} for stopping generally depends on the required run-down time t_B of the belt conveyor, which must not exceed a specific value in emergency cutouts or to prevent overfilling. The necessary deceleration follows from this with an assumed constant braking torque as

$$a_B = -\frac{v}{t_B} \qquad \text{in m/s}^2$$

with the leaving flight

$$L_B = \frac{v \cdot t_B}{2} \qquad \text{in m}$$

and the possibly resulting overfilling

$$V = \frac{m_L^I}{\rho} \cdot L_B \qquad \text{in m}^3$$

* In this equation m_{red} are the reduced masses of non-driven rotating parts of the belt conveyor, such as deflection rollers or pulleys.

The peripheral force occurring on stopping becomes

$$F_{UB} = a_B \cdot \left(m'_L + m'_{R\,red} + 2m'_G\right) \cdot L + F \qquad \text{in N}$$

If significant reduced masses are accelerated, these must be taken into

$$F_{UB} = a_B \cdot \left(m'_L + m'_{R\,red} + 2m'_G\right) \cdot L + F + a_B \cdot m_{red} \qquad \text{in N}$$

The necessary belt tension for transmission of the negative peripheral force F_{UB} occurs on the entry side of the belt and is, in the case of a braked pulley,

$$T_{1B} = F_{UB}^{*} \cdot c_2$$

The drive traction (or, where relevant, the tensioning force) on the leaving side of the braked pulley becomes

$$T_{2B} = T_{1B} - F_{UB}$$

For ascertainment of the belt tensions on starting or stopping, the mass flight L_B may result in very high belt tensions that would lead to unjustified overdimensioning of the belt. In this case the transmittable peripheral force can be determined with the belt tension $T_{2B} = T_{admissible}$ given (possibly for other operating conditions).

$$F_{UB} = - T_{2B} \cdot c_1$$

This yields the deceleration

$$a_B = \frac{F_{UB} - F}{\left(m'_L + m'_{R\,red} + 2m'_G\right) \cdot L} \qquad \text{in m/s}^2$$

an the run-down time

$$t_B = - \frac{v}{a_B} \qquad \text{in s}$$

The non-braked run-down results from $F_{UB} = 0$.

For ascertainment of the belt tensions on starting or stopping, the mass forces of inertia resulting from the acceleration a_A or the deceleration a_B are to be determined.

Mass forces of inertia in N

	Starting	Stopping
Top run	$F_{ao} = a_A \cdot (m_{R\,red\,o} + m_G + m_L)$	$F_{bo} = a_B \cdot (m_{R\,red\,o} + m_{Go} + m_L)$
Return run	$F_{au} = a_A \cdot (m_{R\,red\,u} + m_G)$	$F_{bu} = a_B \cdot (m_{R\,red\,u} + m_G)$
Peripheral force**	$F_{aU} = a_A \cdot m_{red}$	$F_{bU} = a_B \cdot m_{red}$

For the investigation of sections, the mass forces of inertia of a section n amount to

$$F_{a\,n} = a_A \cdot m_n \qquad \text{in N}$$

$$F_{b\,n} = a_B \cdot m_n \qquad \text{in N}$$

* F_{UB} is to be inserted as a positive factor.

** The peripheral forces resulting from the mass forces of inertia occur only at tail pulleys, i. e. on rotating parts of the belt conveyor that are not directly driven or braked.

D – 3.5.2 Multiple pulley drive

The magnitude of the belt tensions is decisive for the conveyor belt design. In general, very high belt tensions occur on the entry side of the drive pulley. This belt tension, that is designated T1 and depends on drive factor c1 with a given peripheral force, can be reduced by increasing the angle of wrap. Values exceeding 230° can be attained by using multiple pulley drives.

Drives that are arranged separately as a head and a tail drive and have only one drive pulley each are not to be regarded as multiple pulley drives, although in this case too the peripheral force is transmitted by an enlarged angle of wrap and the belt tensions can consequently be reduced.

The total transmitted peripheral force is

$$F_U = F_{U1} + F_{U2} \quad \text{with} \quad \frac{F_{U1}}{F_{U2}} \approx \frac{P_{Mnom1}}{P_{Mnom2}}$$

for a two-pulley drive.

Belt tension T_2 is necessary for the transmission of peripheral force F_{U2}. For the transmission of peripheral force F_{U1}, the belt tension on the leaving side of pulley 1 results from this as

$$T_{1/2} = T_2 + F_{U2}$$

The smallest belt tensions are attained with a given peripheral force F_U when the transmission potential of the drive pulleys is fully exploited and the peripheral forces of series mounted drive pulleys are in a specific ratio to one another.

$$T_1 = F_U \cdot c_1 \quad \text{and} \quad T_2 = F_U \cdot c_2 \quad \text{with} \quad \alpha = \alpha_1 + \alpha_2$$

The distribution of the peripheral forces, at which the lowest belt tensions are yielded, follows from the equation

$$\frac{F_{U1}}{F_{U2}} = \frac{e^{\mu\alpha_1} - 1}{e^{\mu\alpha_2} - 1} \cdot e^{\mu\alpha_2}$$

This optimum ratio c_F can be approximated by the provision of differing power outputs at pulleys 1 and 2 in the ratio of

$$\frac{P_{Mnom1}}{P_{Mnom2}} \approx c_F$$

For a friction coefficient μ = 0.25 and the angle of wrap α_1 =180° or α_2 =210°, the frequently executed drive arrangement, for instance, results in the ratio 2/1.

Ratio c_F

μ = 0.25		Angle of wrap α_1					
		160°	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.0	2.2	2.4	2.6	2.8	3.0
	170°	1.9	2.1	2.3	2.48	2.67	2.86
	180°	1.83	2.02	2.2	2.38	2.57	2.75
	190°	1.77	1.95	2.12	2.3	2.48	2.65
	200°	1.71	1.89	2.06	2.23	2.4	2.57
	210°	1.67	1.83	2.0	2.17	2.33	2.5
μ = 0.3		**160°**	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.31	2.53	2.76	3.0	3.26	3.53
	170°	2.22	2.43	2.66	2.89	3.14	3.4
	180°	2.15	2.35	2.57	2.79	3.03	3.28
	190°	2.08	2.28	2.48	2.7	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	3.0
μ = 0.35		**160°**	**170°**	**180°**	**190°**	**200°**	**210°**
α_2	160°	2.66	2.92	3.2	3.51	3.84	4.18
	170°	2.56	2.82	3.1	3.39	3.70	4.03
	180°	2.48	2.74	3.0	3.29	3.59	3.91
	190°	2.41	2.66	2.92	3.19	3.48	3.8
	200°	2.35	2.58	2.84	3.11	3.39	3.7
	210°	2.29	2.52	2.77	3.03	3.31	3.61

When the conveyor belt revolves round pulley 1, the belt tension and thus the service elongation of the belt declines in part. With equal pulley diameters, this results in a slightly lower speed at pulley 2 than at pulley 1, that may lead to a relatively high strain on this drive. Unless this is compensated by special measures, optimum power distribution $PM_{nom1}/p_{M\ nom2}$ is always slightly above the value c_F.

Necessary belt tension on the leaving side of the belt
(on transmission of positive peripheral forces)

Power distribution $P_{M\ nom1}/P_{M\ nom2}$	Belt tension T_2	Remarks
$> c_F$	$F_{U1} \cdot \left(\frac{1}{e^{\mu\alpha 1} - 1} \right)^{*} - F_{U2}$	Full exploitation of transmitting potential at pulley 1
$\leq c_F$	$F_{U2} \cdot \left(\frac{1}{e^{\mu\alpha 2} - 1} \right)^{*}$	Full exploitation of transmitting potential at pulley 2

$$T_1 = T_2 + F_U$$

Necessary belt tension on the entry side of the belt
(on transmission of negative peripheral forces)

Power distribution $P_{M\ nom1}/P_{M\ nom2}$	Belt tension T_1	Remarks
$\geq c_F$	$F_{U1}^{**} \cdot \left(\frac{1}{e^{\mu\alpha 1} - 1} \right)^{*}$	Full exploitation of transmitting potential at pulley 1 or at both pulleys
$< c_F$	$F_{U2}^{**} \cdot \left(\frac{1}{e^{\mu\alpha 2} - 1} \right)^{*} - F_{U1}^{**}$	Full exploitation of transmitting potential at pulley 2

$$T_2 = T_1 - F_U$$

In the case of belt conveyors with separately located drive pulleys (head and tail drive) the peripheral forces are distributed in approximately the ratio of the drive powers. The optimum distribution in these cases is yielded by a comparison of the motional resistances in the top run and the return run. A distribution in which the specific component is taken over by the head or the tail drive is of advantage.

* corresponding to drive factor c_2 of the respective pulley

** The peripheral forces are to be inserted as a positive factor.

D–3.6 Calculation example

An example is given below to elucidate the procedure on calculation of a belt conveyor. For this purpose a schedule is used which permits basic quantities and calculation results to be entered. The most important calculation formulae are also listed once again. The **calculation of the belt conveyor** extends to determination of the peripheral forces. The adjacent calculation sheet accordingly contains all steps required to reach this point. A calculation taking sections into account can be undertaken by means of the expanded schedule attached in the appendix.
The **determination of the belt tensions** is continued for the selected example in the corresponding sections. For this purpose too a tabular schedule can be used; in its expanded form attached in the appendix it can also serve for calculations taking sections into account.
All tabular schedules can be obtained as printed forms from ContiTech's Application Technique Advisory Service.

Calculation of belt conveyor **Form 1**

Sketch of conveying system:

Principal data

	Material handled	Raw salt		
D–2.1	Bulk density	ρ	1.2	t/m³
D–2.1	Angel of repose	β	15	°
D–2.2	Flow of material	Q_m	225	t/h
		Q_v	188	m³/h
D–2.4.1	Max. lump size	k	200	mm

Sections cf. Form 3
D–3.3.2

Conveying length	L	439	m
Conveying height	H	138	m
Gradient	δ	+18	°

D–2.4.1 Belt width	B	800	mm
D–2.4.2 Belt speed	v	2.09	m/s

D–2.4.3 Troughing angle	λ	20	°
Idler spacing	l_o	1.25	m
	l_u	2.5	m

Calculation data

D–2.5 Degree of fill	φ_1	0.85	–
Degree of uniformity	φ_2	1	–
Filling cross section	A	0.03	m²

D–2.5

$$A = \frac{Q_m \cdot \varphi_2}{v \cdot 3600 \cdot \varphi_1 \cdot \rho}$$

Load mass	m_L^l	30	kg/m
Idler weight	m_R^l	13.5	kg/m
Belt weight	m_G^l	8	kg/m

D–3.1

$$m_L^l \frac{Q_m}{3.6 \cdot v}$$

D–3.1.1

$$m_R^l \frac{m_{Ro}}{l_O} + \frac{m_{Ru}}{l_u}$$

Resistance coefficient	f	0.0165	–
Length coefficient	C	1.23	–

D–3.2

$$C = \frac{F_H + F_N}{F_H}$$

Motional resistance

D – 3.3

$$F = C \cdot f \cdot L\,[m_R^l + (2 \cdot m_G^l + m_L^l) \cdot \cos\delta] \cdot g + H \cdot m_L^l \cdot g$$

Distribution cf. Form 3
D – 3.3.1

Motional resistance	F	45617	N
Primary resistance	F_H	4068	N
Secondary resistance	F_N	936	N
Gradient resistance	F_{St}	40613	N
Special resistance	F_S	–	N

Power

D – 3.4

$$P_{Tr} = \frac{F \cdot v}{1000}$$

$$P_M = \frac{P_{Tr}}{\eta} \qquad (P_{Tr} > 0)$$

$$P_M = P_{Tr} \cdot \eta \qquad (P_{Tr} < 0)$$

Efficiency	η	0.85	–
Required power	P_{Tr}	95	kW
	P_M	112	kW
Installed power	P_{Mnom}		kW
Start-up factor	K_A	1.5	–

Peripheral force

D – 3.5

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

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

$$F_U = F$$

$$T_1 = F_U \cdot c_1$$
$$T_2 = F_U \cdot c_2$$

Friction coefficient	μ		–
Angle of wrap	α		°
Drive factor	c_1		–
	c_2		–
Peripheral force	F_U		N

Belt tension	T_1		N
	T_2		N

Starting and stopping

D – 3.5.1

$$F_{UA} = \frac{P_{MN} \cdot \eta \cdot K_A}{v} \; 1000$$

Multiple-pulley drive cf. Form 4
D – 3.5.2

$$a_A = \frac{F_{UA} - F}{(m_L^l + m_{R\,red}^l + 2 \cdot m_G^l) \cdot L}$$

$$a_B = \frac{F_{UB} - F}{(m_L^l + m_{R\,red}^l + 2 \cdot m_G^l) \cdot L}$$

$$m_{R\,red}^l \approx 0{,}9 \cdot m_R^l$$

Peripheral force	F_{UA}		N
Belt tension	T_{1A}		N
	T_{2A}		N
Peripheral force	F_{UB}		N
Belt tension	T_{1B}		N
	T_{2B}		N

Acceleration	a_A	1.798	m/s²
	a_B		m/s²

Calculation of belt conveyor (sections) **Form 3**

Section		L	H	δ	Top run			Return run		
Top run	Return run	in mm	in mm	in °	l_O in m	m_L^l in kg/m	m_{RO}^l in kg/m	l_U in m	m_{RU}^l in kg/m	m_G^l in kg/m
1	2	439	138	+ 18	1.25	30	9.3	2.5	4.2	8

Motional resistances

D–3.3.1

$$F_O = f_O \cdot L \cdot [m'_{Ro} + (m'_G + m'_L) \cdot \cos\delta] \cdot g + H(m'_G + m'_L) \cdot g$$

$$F_U = f_U \cdot L \cdot (m'_{Ru} + m'_G \cdot \cos\delta) \cdot g - H \cdot m'_G \cdot g$$

Section		Top run						Return run			
Top run	Top run	F_O	$m_{R\,red\,o}$	m_G	m_L	F_{ao}	F_{bo}	F_u	$m_{R\,red\,u}$	F_{au}	F_{bu}
		in N	in kg	in kg	in kg	in N	in N	in N	in kg	in N	in N
1	2	54673	3674	3512	13170	36592	–	-9991	1659	9295	–

Mass inertia forces

D–3.5.1

$$F_{ao} = a_A \cdot (m_{R\,red\,o} + m_G + m_L)$$
$$F_{bo} = a_B \cdot (m_{R\,red\,o} + m_G + m_L)$$
$$F_{au} = a_A \cdot (m_{R\,red\,u} + m_G)$$
$$F_{bu} = a_B \cdot (m_{R\,red\,u} + m_G)$$

Calculation of belt conveyor (Multiple pulley drive)

D–3.5.1/3.5.2

Form 4

Power Distribution			
	$c_{F\,head}$	1 : 1	–
	$c_{F\,tail}$	–	–

Operation

Drive pulley	1.	2.	3.	4.	
$P_{M\,nom}$	75	75			kW
F_U	22808	22808			N
μ	0.3	0.3			–
α	180	210			°
c_1	1.64	1.5			–
c_2	0.64	0.5			–

Belt tension (leaving side)			
	1.	14597	N
	2.	11404	N
	3.		N
	4.		N

Starting

D–4.1.2

$$F_{aU} = a_A \cdot m_{red}$$

Drive pulley	1.	2.	3.	4.	
F_{UA}	45753	45753			N
F_{aU}	–	–			N

Belt tensions (leaving side)			
	1.	29282	N
	2.	22877	N
	3.		N
	4.		N

Stopping

D–4.1.2

$$F_{bU} = a_B \cdot m_{red}$$

Drive pulley	1.	2.	3.	4.	
F_{UB}	–	–			N
F_{bU}	–	–			N

Belt tensions (leaving side)			
	1.	–	N
	2.	–	N
	3.		N
	4.		N

D – 4 Stipulation of tension member

➲ **Belt tensions..... 4.1**

Belt tensions in operation 4.1.1
Belt tensions on starting
and stopping..... 4.1.2
Calculation with sections 4.1.3

➲ **Additional strains 4.2**

Flat-or trough and
trough-to-flat transition 4.2.1
Transition curves..... 4.2.2
Turnover 4.2.3

➲ **Selection criteria 4.3**

Pulley diameter 4.3.1
Troughing properties 4.3.2
Centre distance and takeups..... 4.3.3
Effects of material handled 4.3.4
Feeding strain..... 4.3.5

➲ **Tension member data 4.4**

Textile tension members..... 4.4.1
Steel cable tension members 4.4.2
Splice dimensions..... 4.4.3

The tension member of a conveyor belt is designed in general so as to withstand with adequate certainty the maximum tensile strains that may occur on operation of the belt conveyor. The operational and construction-related effects must, however, also be taken into account on selection, as these may lead to a different dimensioning of the tension member or to an adaptation to operating conditions.

Properties and possible applications of textiles and steel cables as load carrying elements in conveyor belts

Material constituting load carrying element	Symbol	Properties and possible applications	Types most widely used
Cotton	**B**	**Natural fibre**, thus susceptible to moisture and less suitable for wet operation; relatively low strength and elongation, not suitable for heavy loads, and now rarely used (only in certain cases such as for bare-bottom-ply belts)	(B 18) B 60
Aramid (Kevlar)	**D**	**Fully synthetic**, highly aroma tic polyamide fibre with extra high strength and low elongation (similar to steel) and tested so far in special cases of application; shares most of the advantages of fully synthetic fibres, is noninflammable without special treatment and gains significance in high-strength textile belts*	
Polyester (Diolen, Trevira)	**E**	**Fully synthetic**, high strength with good elongation properties, large working capacity, largely acid and alkali resistant, unaffected by moisture; processed with polyamide, an ideal fabric for conveyor belts and suitable for handling hot materials	EP 63 EP 100 EP 125 EP 160 EP 200 EP 250 EP 315 EP 400 EP 500 EP 630 (E 100)
Polyamide (Nylon, Perlon)	**P**	**Fully synthetic**, resistant to moisture and chemicals, processed primarily as union thread or in the weft; less suitable as sole tension member material for fully loaded belts and large centre distances owing to excessive takeups	Pb 315 Pb 400 Pb 500 Pb 630
Rayon (artificial silk)	**R**	**Semi-synthetic**, high strength, but now rarely used owing to detrimental properties - loss of strength and shrinkage through moisture	
Steel cables	**St**	**Anticorrosive-treated**, in various types ideal for belt conveyor systems of all kinds; high strength and low elongation, of advantage with large centre distances in particular, good troughing properties and high load capacity	Special conveyor belt steel cables for St 630 St 800 St 1600 St 2500 St 3150 St 4000

* Available on request

Conveyor belts with textile plies and steel cable conveyor belts conform in certain versions to standard specifications and are marked correspondingly.

In the case of textile belts, the

Nominal belt strength k_{nom} in N/mm

is marked after the symbol representing the ply material or, in special-purpose versions, after the type designation.

Example **EP 500/4**
Polyester in warp
Polyamide in weft
Nominal belt strength in N/mm
Number of plies

For steel cable conveyor belts the nominal belt strength is also stated after the symbol representing the ply material.

Example **St 3150**
Steel cables in longitudinal direction
Nominal belt strength in N/mm

STAHLCORD®- Special versions not conforming to standard specifications are marked with a letter/number combination separated by a dash.

The different tension members have preferential applications depending on the strength grading and belt width.

The preselection of a tension member material currently in standard use can be made by means of a rough assessment of the properties in application.

Tension member	B	EP	P	St
Specific strength	4	2	3	1
Additional strains	4	2	1	3
Pulley diameter	4	3	2	1
Troughing properties	3	2	2	1
Elongation	4	2	4	1
Chemical resistance	4	2	3	3
Properties under load	4	2	2	2

1 – very good, 2 – good, 3 – adequate, 4 – poor

The

nominal strength of the splice $k_{nom\ V}$ in N/mm

must be taken into account in the design of a conveyor belt. The splice*, with which the ends of the initially open-ended belt are joined, may show a loss of strength as compared with nominal belt strength k_{nom}; this loss is yielded purely theoretically by the splice construction (cf. D–4.4.3).

Even if no drop in strength in the splice is yielded theoretically, a conveyor belt must not be strained to the nominal strength limit. Factors arising from splicing, time-dependent fatigue symptoms and defective friction transmission lead to further losses in strength that also depend in the case of steel cable belts, for instance, on the number of laps in the splice and cannot yet be determined mathematically.

The design of the conveyor belt tension member, with regard to the nominal strength of the splice, is therefore based on a safety factor S which takes into account these influences among other things.

A rough determination of the tension member needed can be made if the required or actually available power P_{Tr} on the drive pulley is known (cf. D–3.4).

The global determination of the conveyor belt type is based on

Safety factor $S = 8$

which is intended to cover not only the imprecisely determinable losses of strength in the splice but also the increased belt tensions that may occur on starting and stopping, together with all additional strains. This calculation is therefore reasonably reliable only for conventional belt conveyors with standard start-up and braking properties.

* Conveyor belt splices and splicing processes are described in detail in Section E of the ContiTech Conveyor Belt Service Manual.

The assumption is made that the greatest strain on the belt occurs at the entry side of the belt on the drive pulley located at the head of the system. As mentioned in Section D–3.5, however, T1 can be assumed to be the maximum belt tension only when the conveyor systems in question have a constant uphill gradient with a low conveying height.

Determination of belt construction

The graph applies to multiple-pulley drives too, when they are located at the head and the transmission potential is fully exploited.

D–4.1 Belt tensions

If the tension member design is based on the width-related belt tension (unit belt tension) amounting from the maximum belt tension T_{max} constantly prevalent in operation to

$$k_{max} = \frac{T_{max}}{B} \quad \text{in N/mm}$$

the safety factor S is to be taken into account; this covers the temporarily increased drive traction $T_{max\,t}$, that may occur in special operating states, together with additional strains that may occur at the flat-to-trough or trough-to-flat transition flights, at transition curves or at turnovers.

$$k_{nomV} = k_{max} \cdot S \quad \text{in N/mm}$$

In general, however, the starting-up process of a belt conveyor at least will have to be investigated besides the state of steady operation. If the unit belt tension from the temporarily occurring maximum drive traction $T_{max\,t}$ is ascertained as

$$k_{max\,t} = \frac{T_{max\,t}}{B} \quad \text{in N/mm}$$

whereby stopping or critical section loads may need to be taken into account, a reduced safety factor Si can be applied for layout.

$$k_{nomV} = k_{max\,t} \cdot S_t \quad \text{in N/mm}$$

Safety factors*

Operating conditions	S	St
Favourable (minimum load cycle strain and bending strain)	6.7	4.5
Normal	8.0	5.4
Unfavourable (high load cycle strains and bending strains)	9.6	6.4

Low numbers of load cycles can be assumed if the conveyor belt is being designed for a relatively short operating period. A low bending strain results, for instance, from over dimensioning of the drive and tail pulleys.

The conveyor system is normally constructed in such a way as to prevent extreme additional strains. In special cases, however, a limit design may be necessary in which the additional strains attain extremely high values or other strains of disadvantage for the conveyor belt arise. In these cases, both the maximum belt tension and the additional strains are to be taken into account (D–4.2).

Diverging from the global considerations described in Section D–4, the maximum belt tension may occur at another point, for example in braked belt conveyors or with different drive layouts. The magnitude of the belt tensions is also determined to a considerable extent by the point and method of the belt tensioning generation required for transmission of the peripheral forces in the permanent and in the temporary operating state. In these cases the belt tensions occurring at the individual points of the belt conveyor must be determined. It is advisable to designate the points at which the constant course of the belt tension changes with T_1, T_2, T_3, T_4 etc. consecutively and to retain this designation regardless of the drive layout and the operating method.

* In some cases lower nominal belt strengths can be used if a detailed calculation is carried out and the dynamic belt safety factor and the fatigue strength of the belt splice are checked.

Although the temporary operating states are decisive in certain cases for the permanent belt tensions too, it is the belt tension curve for operation that is first determined, since the data required for this purpose are known in any case.

The belt tension curve is determined by a **sequential calculation** in which the belt tensions are calculated in succession, starting at one point of the belt conveyor. In this process it is always advisable to proceed in accordance with firm rules:

1.	Starting the sequential calculation with T_2. The minimum belt tension T_{min} or - if a head drive is present - the belt tension on the leaving side is used as the initial value of T_2. The sequential calculation may, however, also be started as a general rule with the assumption $T_2 = 0$, the final values being yielded by subsequent adjustments..
2.	Addition of the motional resistances and subtraction of the peripheral forces, continuously in belt movement direction. All variations in belt tension are taken into account with the corresponding signs.

The belt tensions resulting from the motional resistances and peripheral forces at the individual points of the top run and the return run must be adjusted if, for instance, a certain minimum belt tension is not attained to restrict the sag.

$$T_{min} = \frac{(m_L^I + m_G^I) \cdot g \cdot l_O}{8 \cdot h_{rel}} \quad \text{in N}$$

The sag should not exceed the admissible value of $h_{rel} = 0.01$ at any point of the loaded top run, as an increased motional resistance must otherwise be anticipated. T_{min} can be selected as a smaller value by reduction of the idler spacing l_o, possibly at the critical point only.

The belt tensions may also require adjustment if the belt tension necessary for power transmission is not present at one drive pulley. All belt tensions must be increased by the same quantity until all demands have been fulfilled.

The type of belt tensioning must be known for further calculation of the temporary belt tensions. Three systems are to be distinguished in principle:

1. Rigid tensioning (with fixed takeup pulley) by takeups, straining pulleys

2. Movable tensioning (with constant tension force) by counterweights

3. Adapted tensioning (with adjustable tension force) by controllable straining pulleys etc.

In the case of **rigid tensioning**, the belt elongations are constant* in their sum total for all operating conditions. The consequence is belt tensioning forces that have to comply with the extreme operating states on starting (or stopping) and lead in operation to the highest belt tensions in comparison with other types of tensioning.

* For a conveyor belt the absolute elongation, approx.

$$\Delta l = \frac{T_2 + T_3}{2 \cdot B \cdot E} \cdot L + \frac{T_5 + T_1}{2 \cdot B \cdot E} \cdot L \qquad \text{in m}$$

and

$$\Delta l_A = \frac{T_{2A} + T_{3A}}{2 \cdot B \cdot E} \cdot L + \frac{T_{5A} + T_{1A}}{2 \cdot B \cdot E} \cdot L \qquad \text{in m}$$

It follows, with rigid tensioning and a constant takeup, that

$$\Delta l = \Delta l_A$$

and

$$\frac{T_2 + T_3}{2} \cdot L + \frac{T_5 + T_1}{2} \cdot L = \frac{T_{2A} + T_{3A}}{2} \cdot L + \frac{T_{5A} + T_{1A}}{2} \cdot L$$

or in simple terms

$$\sum T = \sum T_A$$

This condition is fulfilled by all belt tensions in operating being increased by the difference

$$\Delta T = \frac{\sum T_A - \sum T}{4}$$

when four span end tensions are present. When calculating sections (with more than four span end tensions) the section lengths are to be taken into account. “Span end” belt tension refers to forces acting on the belt which occur at the beginning of a belt section of length L or of part lengths measuring L_n .

A constant tensioning force through movable counterweight gear causes belt tensions that are constant at the takeup point in any operating state and must therefore comply with the most extreme operating conditions. The consequence of this in operation is higher belt tensions than would be required for the transmission of the peripheral forces.

The **adapted belt tensioning** leads to belt tensions complying with the specific operating instance, as they are adjusted by regulating devices to the value required for transmission of the peripheral forces. In operation, this more intricate type of belt tensioning yields the lowest belt tensions as compared with other systems.

The operating instances of starting and stopping are distinguished from permanent operation by the additional mass inertia forces and the changed peripheral forces. The belt tension curve therefore deviates from that ascertained for operation and is to be determined by a new sequential calculation. The belt tensions in operation are then to be checked and adjusted where

Adjustment of drive tractions in operation

Type and location of tensioning	Condition	Adjustment
rigid, tensioning location has no effect on the calculation	$\Sigma T = \Sigma T_{max}$	$\Delta T = \frac{\Sigma T_{max} - \Sigma T}{4}$
movable, belt tension Tv at takeup point is constant	$T_v = T_{v\,max}$	$\Delta T_v = T_{v\,max} - T_v$
adapted, tensioning location has no effect on the calculation	–	–

The decisive factor for the adjustment is always the largest sum of the belt tensions ΣT_{max} or the maximum belt tension $T_{v\,max}$ at the takeup point. The operating-state belt tensions deviating from this, that have to be adjusted, are designated as T or. T_V , i.e. when ΣT_{max} or $T_{v\,max}$ occurs on stopping, both the belt tensions in operation and those on starting are to be adjusted. Apart from starting and stopping, it is also critical section loads that may lead to an adjustment of the belt tensions.

The sequential calculation process permits the belt tensions of a belt conveyor to be determined in accordance with the ever-constant schedule. The conclusion of each sequential calculation is the adjustment of the belt tensions according to demands relating to tension for transmission of peripheral forces and sag restriction.

A distinction is made between three stages of the sequential calculation which provide greater precision of the results, depending on execution:

Stage 1 After conclusion of the adjustments, the belt tensions in operation (permanent state of operation at normal rating) are at hand without starting or stopping being taken into account. This procedure is sufficiently accurate only for horizontal or uphill belt conveyors with head drive, standard starting and braking properties being a prerequisite.

Stage 2 After conclusion of the sequential calculation for operation and adjustment, a second sequential calculation is executed based on the peripheral forces and motional resistances including the mass inertia forces on starting. After determination of the type and location of the takeup, the two sequential calculations are concluded by a further adjustment, i. e. the belt tensions in both operating states subsequently fulfil the demands relating to tension, sag and the condition set by the takeup gear (e. g. a constant sum of the span end tensions for rigid takeup). This procedure is sufficiently accurate for all belt conveyors if the stopping can be effected in such a way that no critical operating states occur.

Stage 3 A third sequential calculation with the stopping conditions is executed and adjusted. The question of which belt tensions must be adjusted in order to fulfil the resultant condition, again taking the takeup gear into account, is then investigated. In general the result is sufficiently accurate for any application.

Course of belt tension determination by sequential calculation

In the case of belt conveyors with varying system gradients or with critical load states, all three stages of the sequential calculation may be executed accordingly.

D–4.1.1 Belt tensions in operation*

As the belt tensions in operation are initially determined by a sequential calculation, starting with T_2, a uniform schedule can be used which is equally suitable both for random drive layouts and belt tensioning systems and for driven or braked conveying systems. The belt tensions on starting and stopping and on section loading can also be calculated according to the same schedule and be entered in the adjacent table. The varying factors are the respective peripheral forces to be entered at the head of the table and the resulting belt tensions at the drive pulleys. In the case of multiple pulley drives it should be noted that peripheral forces are transmitted at each drive pulley. In this instance the belt tension $T_{1/2}$ (or $T_{3/4}$) can be entered additionally between T_1 and T_2 (or T_3 and T_4).

The belt tension T_{min} required for restriction of the sag can also be determined in advance. The peripheral forces and motional resistances are now listed with their correct signs in Column 1 in the sequence determined by the moving direction of the belt (cf. table D–3.6).

The belt tension resulting after each section can be designated and entered in Column 2. The values of the belt tensions are determined in the sequential calculation.

Sequential calculation of drive tractions in operation

Tensioning force	$T_2 = T_{min}$
+ motional resistance (return run)	$T_3 = T_2 + F_u$
– peripheral force (if drive exists)	$T_4 = T_3 - F_{U2}$
+ motional resistance (feeding point)	$T_5 = T_4 + F_N$
+ motional resistance (top run)	$T_1 = T_5 + F_O$
– peripheral force (control)	$T_2 = T_1 - F_{U1}$

It must then be checked whether the belt tension required for restriction of the sag or for transmission of the peripheral forces is being maintained at the corresponding points.

Adjustment (operation)	Condition	Adjustment ΔT_C
Tensioning force, drive 1	$T_2 \geq F_{U1} \cdot c_2$	$F_{U1} \cdot c_2 - T_2$
Tensioning force, drive 2	$T_4 \geq F_{U2} \cdot c_2$	$F_{U2} \cdot c_2 - T_4$
Sag, feeding point	$T_5 \geq T_{min}$	$T_{min} - T_5$
Sag, discharge point	$T_1 \geq T_{min}$	$T_{min} - T_1$

The maximum value of ΔT_C (sequential calculation) is decisive for the adjustment. When each belt tension from Column 2 has been increased by the value of ΔT_C, the belt tensions for operation are yielded in Column 3, though regardless of starting and stopping. Any adjustments that may arise from this can be entered in the next columns after investigation of these cases.

* Operating states in which no significant changes in belt speed occur (permanent state) are given the abbreviated designation “Operation”. This term also covers part-load and no-load operation.

Calculation of drive tractions **Form 6**
(example)

D–4.1

Operating state	operation
Stage	2
Tensioning	manual

$$T_{min} = \frac{(m_L^I + m_G^I) \cdot g \cdot l_O}{8 \cdot h_{rel}}$$

Belt tension	T_{2A}	11404	N
	T_{min}	5824	N

Adjustment	ΔT_C	9055	N

Sequential calculation

Section	**1** Motional resistances / peripheral forces / mass inertia forces	**2** Sequential calculation		**3** after adjustment ΔT_C in **N**	after adjustment* with ΔT in **N**	after adjustment with ΔT in **N**
2	- 9991	T_2*	5824	14879	38996	
–	–	T_3*	-4167	4888	29005	
F_N	936	T_4	-4167	4888	29005	
1	54673	T_5*	-3231	5824	29941	
F_{U1}	22808	T_1*	51442	60497	84614	
F_{U2}	22808	$T_{1/2}$	28634	37689	61806	
		T_2	5826	14881	38998	
Sum total of span end tensions				86088	182557	

* Span end tension

* This correction is possible only after calculation of the belt tension on starting, and subsequent determination of the value ΔT .

D–4.1.2 Belt tensions on starting and stopping

The belt tensions on starting and stopping are also yielded by a sequential calculation, taking into account, however, the changed peripheral forces that must be subtracted again and the additional mass inertia forces (D–3.5.1) that are added.

Sequential calculation of belt tensions on starting	
Tensioning force	$T_{2A} = T_{min}$
+ motional resistance (return run) + mass inerta forces (return run)	$T_{3A} = T_{2A} + F_U + F_{au}$
– peripheral force*	$T_{4A} = T_{3A} - F_{UA2}$
+ motional resistance (feeding point)	$T_{5A} = T_{4A} + F_N$
+ motional resistance (top run) + mass inertia forces (top run)	$T_{1A} = T_{5A} + F_O + F_{ao}$
– peripheral force (control)	$T_{2A} = T_{1A} - F_{UA1}$

On stopping, the drive traction required for peripheral force transmission is yielded at the entry side of the braking pulley.

The sequential calculation may, however, be started at the leaving side of pulley 1 in this case too. It is then executed in accordance with the same schedule, all peripheral forces, motional resistances and mass inertia forces being inserted with the corresponding signs.

Sequential calculation of belt tensions on stopping	
Tensioning force	$T_{2B} = T_{min}$
+ motional resistance (return run) + mass inertia forces (return run)	$T_{3B} = T_{2B} + F_U + F_{bu}$
– peripheral force**	$T_{4B} = T_{3B} - F_{UB2}$
+ motional resistance (feeding point)	$T_{5B} = T_{4B} + F_N$
+ motional resistance (top run) + mass inertia forces (top run)	$T_{1b} = T_{5B} + F_O + F_{bo}$
– peripheral force (control)	$T_{2B} = T_{1B} - F_{UB1}$

The minimum belt tensions are to be checked and adjusted if necessary by raising all belt tensions. The sag may be ignored on starting and stopping, provided attention is paid that no negative belt tension values occur.

Adjustment (starting)	Condition	Adjustment
Tensioning force, drive 1	$T_{2A} \geq F_{UA1} \cdot c_2$	$\Delta T_C = F_{UA1} \cdot c_2 - T_{2A}$
Tensioning force, drive 2	$T_{4A} \geq F_{UA2} \cdot c_2$	$\Delta T_C = F_{UA2} \cdot c_2 - T_{4A}$
Sag	$T > O$	$\Delta T_C = - T$

The minimum belt tensions are to be checked for stopping too and adjusted where necessary by raising the belt tension.

Adjustment (stopping)	Condition	Adjustment
Tensioning force, drive 1 (braked)	$T_{2B} \geq F_{UB1} \cdot c_1$	$\Delta T_C = F_{UB1} \cdot c_1 - T_{2B}$
Tensioning force, drive 2 (braked)	$T_{4B} \geq F_{UB2} \cdot c_1$	$\Delta T_C = F_{UB2} \cdot c_1 - T_{4B}$
Sag	$T > O$	$\Delta T_C = - T$

* If a drive is not switched on on start-up, the inertia forces F_{aU} of the reduced masses of the rotating drive components are to be added at this point where applicable.

** If a drive is not combined with a brake, the inertia forces F_{bU} of the reduced masses of the rotating drive components are to be added at this point where applicable.

Calculation of belt tensions **Form 5**

D–4.1

Operating state	Starting
Stage	2
Tensioning	manual

$$T_{min} = \frac{(m_L^I + m_G^I) \cdot g \cdot l_O}{8 \cdot h_{rel}}$$

Belt tension	T_{2A}	22877	N
	T_{min}	–	N

Adjustment	ΔT_C	–	N

Sequential calculation

Section	1 Motional resistance / peripheral forces / mass inertia forces	2 Sequential calculation		3 after adjustment with ΔT_C in N	
		T_{2A}*	22877		
2	-696	T_{3A}*	22181		
–	–	T_{4A}	22181		
Fn	936	T_{5A}*	23117		
1	91265	T_{1A}*	114382		
FuA	45753	$T_{1/2A}$	68629		
FuA	45753	T_{2A}	22876		
Sum total of span end tensions			182557		
Adjustment			ΔT	24117	N

* Span end tensions

Starting:

$$k_{nom\,V} = k_{max\,t} \cdot S_t$$

$$= \frac{114382}{800} \cdot 5.4 = \underline{772\ N/mm}$$

Operation:

$$k_{nom\,V} = k_{max} \cdot S$$

$$= \frac{84614}{84614} \cdot 8 = \underline{846\ N/mm}$$

Selected nominal belt strength:

St 800	S = 7.56 S_t = 5.59

$$\Delta T = \frac{\sum T_{max} - \sum T}{4} = \frac{182557 - 86088}{4}$$

D – 4.1.3 Belt tensions with regard to sections

Conveying systems with sections at different gradients due to topographical factors or other constructional conditions may have increased motional resistances and thus greater belt tensions in special loading states. Even, however, if these critical operating instances are prevented by an appropriate control system, belt tensions may be present at certain points of the conveying system with continuous loading, and these must be taken into account in the design. Whereas increased belt tensions may occur at vertices of the system layout due to the weight of belt and load, the minimum belt tension may not be attained at saddle points.

The belt tension curve in operation, on starting, stopping and also with differently loaded sections can be executed in accordance with the sequential calculation schedule too, the belt tensions at the individual points of the system resulting from addition of the motional resistances or mass inertia forces of the sections and subtraction of the peripheral forces. Following the sequential calculation, the minimum belt tensions are checked and increased where necessary. The expanded tabular schedule attached in the appendix can be used for this purpose.

Starting and stopping, too, are to be checked as before by a sequential calculation. It should be observed, however, that the adjustment ΔT of the traction forces in operation results from the equation (for starting)

$$\sum \frac{T_n + T_{n+1}}{2} \cdot L_n = \sum \frac{T_{nA} + T_{n+1A}}{2} \cdot L_n$$

when tensioning is rigid, T_n and T_{n+1} representing the span end tensions of the sections with the length L_n. The adjustment is then

$$\Delta T = \frac{\sum (T_n + T_{n+1}) \cdot L_n - \sum (T_{nA} + T_{n+1A}) \cdot L_n}{2 \cdot \sum L_n}$$

D – 4.2 Additional strains

Determination of the belt strength is generally founded on the calculated belt tensions, taking as a basis the corresponding safety factors, the assumption being made that the belt tension is distributed uniformly across the belt cross-section.

$$S_x \frac{k_{nomV} \cdot B}{T_x}$$

These optimum conditions, however, apply to untroughed conveyor belts only. If the belt is led through horizontal or vertical curves, undergoes transitions from flat to trough or vice versa or, for instance, is turned, the uniform tension distribution is disturbed at these points and also at a greater distance from them. Local additional strains overlap.

If the belt undergoes gradual transitions from flat to trough or vice versa, i. e. in the course of a longer flight, or is led through gradual curves, i. e. those with a negligible bend, the additional strains occurring are covered globally by standard safety factors.

Taking certain marginal conditions into account, conveying systems featuring extremely short troughing transitions, relatively small transition radii or curves can also be equipped with ContiTech conveyor belts. The belt strength is exploited to such an extent that values remain below the global safety factors.

Minimum safety factors for additional strains (guide values)

Belt type	S_{min}	$S_{min\ t}$ (short-term)
STAHLCORD® belts with 1- or 2-lap splice	3.0	2.5
STAHLCORD® belts with 3- an more lap splice	3.3	2.8
Fabric carcass belts	3.6	3.0

The given safety factors are to be regarded as limit values. The constructional design must ensure that these values are always attained and that the stated minimum safety factors are maintained even in the extremely unfavourable operating state.

The additional strains are determined essentially by the elongation properties of the tension member. This is identified by the belt parameter K_G which can be read off from the chart below for ContiTech conveyor belts.

Excessive **traction reliefs** of specific belt zones may also occur in individual cases.

Under certain conditions the additional strains are reduced by an elongation compensation outside the conveyor belt section under consideration. This can be taken into account approximately by a compensation factor **C**ε.

Compensation factor Cε

	STAHLCORD® belts	Fabric belts
Standard conditions	150	60
Adjacent transition arcs* or troughing transitions	0 – 100	0 – 15
Turnover or inverted trough roll in return run	100 – 150	15 – 60
High friction on tail pulley	120 – 150	30 – 60

Extreme additional strains are to be avoided in cases of unfavourable operating conditions. Such a high degree of exploitation of the belt strength may reduce the service life of a belt, so that a check must be made in the individual case, taking all affecting factors into account, as to whether the selected belt is still suitable for the specific application. The staff of ContiTech are available for detailed consultation on these questions.

* Adjacent, **opposing** transition curves can lead to more favourable relationships.

D–4.2.1 Flat-to-trough and trough-to-flat transition

In order to prevent extensive additional strains in the flat-to-trough or trough-to-flat transition area, the transition lengths L_Y must have adequate dimensions.

Transition with increased pulley height h_{Tr}

The belt edges are stretched by converting the belt from the troughed form to the flat form on the tail pulley or vice versa. This yields an uneven tension distribution that may lead to excessive strain on the belt edges. When the belt tensions are relatively low, certain areas of the belt may also be buckled. This form of strain should also be avoided, or a certain degree at least not exceeded.

In the rough calculation of the transition lengths L_y, it is correspondingly essential to distinguish whether the decisive criterion is the edge elongation or the buckling of the belt centre.

In general, excessive edge elongation can occur if the mean belt tension of the conveyor is relatively large as a result of the belt tension T_x at the place concerned being high.

Minimum transition lengths for flat-to-trough or trough-to-flat transitions (guide values)

for	Transition length in m	Criterion
$S_x < \frac{S_{min}}{b_f/B}$	$L_Y \geq B \cdot \lambda \sqrt{\frac{S_{min} \cdot S_x}{S_x - S_{min}} \cdot K_G . K_{f1}}$	Edge elongation

The troughing angle λ is to be inserted here in radian measure and is to be estimated for 5-part idler assemblies, possibly by means of the connecting line from the ends of the centre idler to the belt edge. The troughing parameter $\mathbf{K_{f1}}$ can be determined from the ratio $\mathbf{b_f/B}$.

$$K_{f1} = \left(\frac{b_f}{B}\right)^2 - \left(\frac{b_f}{B}\right)^3$$

Buckling may occur when the drive traction T_x at the belt conveyor point under consideration is relatively small.

Minimum transition lengths for flat-to-trough or trough-to-flat transition

for	Transition length in m	Criterion
$S_x \geq \frac{S_{min}}{b_f/B}$	$L_Y \geq B \cdot \lambda \sqrt{S_x \cdot K_G \cdot K_{f2}}$	Buckling

The troughing parameter here is

$$K_{f2} = \left(\frac{b_f}{B}\right)^3$$

Troughing angle λ and parameter $\mathbf{K_G}$ are to be inserted as above.

In some cases the calculation procedure described here will lead to transition lengths that cannot be realized straight away in the case of conveyors, for example, because of the overall construction. An improvement in the tension distribution can be attained by raising the top edge of the tail pulley by the amount h_{Tr} above the lowest point of the troughing. The increased height of the pulley should not exceed

$$h_{Tr} = \frac{1 - \cos\lambda}{\sin\lambda} \cdot b_f$$

It should be noted that a so-called “springboard effect” may occur at the discharge pulley. Increased pulley heights in which the loaded belt rises from the centre idlers should also be avoided.

The reduced transition length results approximately with

$$\lambda_{red} = \lambda - \frac{h_{Tr}}{b_f} \qquad \text{in rad}$$

Investigations have shown that a compensation of the elongations caused by the troughing transition occurs outside the actual transition length too. If this is taken into account with compensation factor c_ε, the necessary transition length can possibly be reduced somewhat more.

$$L_{Y\,red} \approx L_Y^2 \,/\, \sqrt{L_Y^2 + c_\varepsilon \cdot \lambda^2 \cdot b_f^2}$$

In those cases where an extreme dimensioning of the transition lengths is of special importance, a verification of the effective tension distribution can be undertaken by our Application Technique Advisory Service.

D – 4.2.2 Transition curves

In general the belt conveyor will connect the feeding point with the discharge point of the material handled in the straightest possible line. The course can also, however, be adapted within certain limits to the given situation of ah optimum flight routing or the system construction by horizontal and vertical curves.

The minimum radius of horizontal curves is normally not given by the admissible belt strain but by the demand that the belt should maintain a steady position in the top run and in the return run even in the least favourable operating state. This occurs when a relatively high drive traction $\mathbf{T_X}$ is active in the curve with an unloaded belt or with low belt and material weight. The following formula can serve as a statement for estimating the minimum radius $\mathbf{R_h}$; this formula takes into account that the idler trough or idler is raised by the angle λ_R on the inside of the curve as compared with the horizontal.

$$R_h \geq \frac{T_x}{m_G' \cdot g\left[b_f/B \cdot \tan(\lambda+\lambda_R) + l/B \cdot \tan\lambda_R + b_f/B \cdot \tan(-\lambda + \lambda_R)\right]} \quad \text{in m}$$

Additional measures such as tilting single idlers may enable horizontal curves to be executed more narrowly too. A precise investigation is indispensable in this case.

In all instances the execution of the transition curves must ensure that there is no falling short of the minimum safety factors through the additional strains.

Minimum radii for horizontal curves (guide values)

for	Minimum radius in m	Criterion
$S_x \geq 2\ S_{min}$	$R_h \geq B \cdot S_x \cdot K_G$	Buckling
$S_x < 2\ S_{min}$	$R_h \geq B \cdot \frac{S_x \cdot S_{min}}{S_x - S_{min}} \cdot K_G$	Edge elongation

Vertical transition curves are frequently provided to attain an increased discharge for transfer to an adjoining belt conveyor. Both concave and convex curves may arise in this process; these have to be calculated in different ways. Transition curves are also necessary with underpasses and overpasses, and occur in an extreme form on conveying equipment with a articulated boom.

The radius R_k of concave conveyor belt curves must be sufficiently large to guarantee that the belt fits into the troughing assemblies in any operating state. When the takeup is rigid, the lifting of the belt may lead to an excessive drop in the necessary belt tension, especially on starting in the unloaded state. In the case of tensioning with controllable straining pulleys, too, however, the belt lengths being released may possibly not be taken up quickly enough by the takeup loop. Any lifting of the belt should be avoided, primarily in order to prevent discharge or scraping of material from the belt.

$$R_k \geq \frac{T_x}{m_G^{I} \cdot g \cdot \cos\delta} \qquad \text{in m}$$

Smaller radii are permissible if lifting in unloaded state is accepted. In this case, however, checking brackets or idlers must be provided.

The selected radius should also be checked under all circumstances when very low belt tensions are present or the belt in its troughed form is forced through concave curves.

Minimum radii for concave curves (guide values)

for	Radius in m	Criterion
$S_x \geq \frac{S_{min}}{(1-b_f/B)}$	$R_k \geq \frac{S_x \cdot S_{min}}{S_x - S_{min}} \cdot 2 \sin\lambda \cdot \frac{b_f^2}{B} \cdot K_G$	Buckling
$S_x < \frac{S_{min}}{(1-b_f/B)}$	$R_k \geq S_x \cdot 2 \sin\lambda \left(b_f - \frac{b_f^2}{B} \right) K_G$	Edge elongation

The radius R_V of convex conveyor belt curves is determined exclusively by the admissible additional strains or excessive tension reliefs.

Minimum radii for convex curves (guide values)

for	Radius in m	Criterion
$S_x \geq \frac{S_{min}}{b_f/B}$	$R_v \geq S_x \cdot 2 \sin\lambda \cdot \frac{b_f^2}{B} \cdot K_G$	Buckling
$S_x < \frac{S_{min}}{b_f/B}$	$R_v \geq \frac{S_x \cdot S_{min}}{S_x - S_{min}} \cdot 2 \sin\lambda \left(b_f - \frac{b_f^2}{B} \right) \cdot K_G$	Edge elongation

Taking the elongation compensation into account, the radii of convex curves can be reduced.

$$R_{v\,red} \approx \frac{R_v^2}{R_v + c\varepsilon \cdot \sin\lambda \cdot b_f}$$

If extreme radii, e. g. on bends, are to be realized and if buckling of the belt is possibly to be permitted, the ContiTech Application Technique Advisory Service should be consulted.

D – 4.2.3 Turnover

Turnovers are provided to prevent the material still adhering on the belt surface after discharge from being abraded by the idlers and leading to the conveying system being soiled. Wear on the idlers is furthermore reduced. Turnover as a measure is of interest primarily in the transport of bulk material where conventional belt cleaning fails to produce the required success.

There are three types of turnover to be distinguished; these are selected in accordance with the conveying system data. The turnover length L_W is to be dimensioned in such a way that no excessive additional strains occur.

Guide values for planning turnover lengths

			Min. turnover L_W for		
Typ of turnover	Belt width B in mm	Belt speed in m/s	cotton belts	EP belts	STAHLCORD® belts
Unguided	≤ 1200	1,6	8 x B	10 x B	–
Guided	≤ 1600	3.4	10 x B	12.5 x B	22 x B
Supported	≤ 2400	6,0	–	10 x B	15 x B

In order to prevent excessive strains on the belt, care should be taken that the belt wrap round any pulley does not exceed 15° on a length corresponding to approx. half the turnover in each instance before entry into the turnover and after leaving the turnover.

With the safety factor S_x in the turnover area, the required turnover length, in which neither buckling nor excessive edge elongation is to be expected, can be determined.

Minimum turnover lengths (guide values)

for	Turnover length in m	Criterion
$S_x \geq 3S_{min}$	$L_W \geq \pi \cdot B \sqrt{S_x \cdot \frac{K_G}{12}}$	Buckling
$S_x < 3S_{min}$	$L_W \geq \pi \cdot B \sqrt{\frac{S_x \cdot S_{min}}{S_x - S_{min}} \cdot \frac{K_G}{6}}$	Edge elongation

A check should be made as to whether the marginal condition below is adhered to with the selected turnover length L_W and the belt tension T_x.

$$\frac{L_W \cdot m'_G \cdot g}{8 \cdot T_x} \leq 0.015$$

D – 4.3 Selection criteria

When selecting the conveyor belt for a specific conveyor system, a number of other aspects are to be observed that can still be taken into account by the designer or that may lead to a reconsideration of the design already prepared. As is the case with most technical problems, there are frequently a number of solutions in belt conveyor designing. Operational reliability, expected service life and thus long-term economy can be improved by investing more in the designing or drafting of the system. The engineers of ContiTech's Application Technique Department will be pleased to give advice on individual questions.

D–4.3.1 Pulley diameter

The minimum admissible pulley diameter is linked with the demand for an adequate service life of the belt and in particular of its splice, the basic consideration being that the service life of the splice equals that of the belt. In the case of belts subject to wear and of belts with mechanical fasteners, there is therefore the possibility of using pulleys with smaller diameters.

Type of pulley

Type of pulley

Type of pulley

A distinction is made between the following types of pulley:

A – pulleys in the region of high drive tractions with angles of wrap > 30° (drive pulleys)
B – pulleys in the region of low drive tractions with angles of wrap > 30° (tail pulleys)
C – pulleys with angles of wrap ≤ 30° (snub or deflection pulleys)

Stipulation of the pulley diameter depends essentially on the thickness of the tension member in the conveyor belt selected.

The diameter of the pulley at which the highest drive traction $T_X = T_{max}$ is present (pulley type A) is determined first.

$$D_{Tr\,(A)} \geq S_Z \cdot C_I \qquad \text{in mm}$$

The thickness s_Z of the tension member can be drawn from the tension member data (D–4.4) and factor c1 is shown in the table below.

c_I	Tension member material in warp (longitudinal)
60	Cotton (B)
90	Polyamide (P)
105 **100** **80**	Polyester (E) CON-MONTEX® CON-BITEX®
145	STAHLCORD® (St)

The ascertained pulley diameter is to be rounded off upwards to the next standardized size if necessary. For pulleys of types B and C, the diameters are stated in the same stage respectively.

Standardized pulley diameters in mm

	Type of pulley		
Stage	**A**	**B**	**C**
1	100	–	–
2	125	100	–
3	160	125	100
4	200	160	125
5	250	200	160
6	315	250	200
7	400	315	250
8	500	400	315
9	630	500	400
10	800	630	500
11	1000	800	630
12	1250	1000	800
13	1400	1250	1000
14	1600	1250	1000
15	1800	1400	1250
16	2000	1600	1250

Practical experience permits the pulley diameters calculated in accordance with the global formula to be reduced slightly.

When the belt strength is utilized to a lesser degree, pulley diameters of lower stages can be selected.

Utilization	**Diameter stages Type of pulley**		
	A	**B**	**C**
$\frac{S}{S_x} < 0.6$		1 stage lower	
$\frac{S}{S_x} < 0.6$	2 stages lower		1 stage lower

In the case of conveying systems subject to extreme loads or high additional strains, it is advisable to check the surface pressure between the conveyor belt and the pulley circumference too (cf. paragraph D – 3.5).

D – 4.3.2 Troughing properties

Impeccable belt guidance in the idler trough demands good tracking properties and adequate troughability of the conveyor belt. ISO recommendation R 703 states that the conveyor belt is troughable when the D/B parameter determined in a stipulated test procedure attains a certain minimum value.

Troughability in accordance with ISO R 703

Troughing angle λ	20°	25°	30°	35°	40°	45°
Min. value D/B	0.05	0.06	0.07	0.09	0.12	0.17

Practice-orientated tests by ContiTech have shown that a conveyor belt can be regarded as troughable when more than 35 % of its width rests on the idlers in unloaded state, the centre idler being contacted too.

The troughability is determined essentially by the transverse rigidity of the plies and the dead weight of the belt. Troughability improves with increasing belt width. The troughability can be estimated by means of an approximation formula in the range of standard belt widths, troughing angles and cover gauges.

$$\tan\lambda \leq \left(\frac{B}{S_Z}\right)^3 \cdot m_G^{II} \cdot c_q$$

m_G^{II} – conveyor belt mass in kg/m² (cf. D – 3.1.2)
B – belt width in m
S_Z – tension member thickness in mm
c_q – transverse rigidity factor

c_q	Tension member material in weft (transverse)
12	Cotton (B)
16	Viscose (Z)
18	Polyamide (P)
42	Steel cable belts (St)

Apart from the generally good troughability of ContiTech conveyor belts, it is primarily their outstanding tracking properties that ensure reliable belt guidance even when operating conditions are unfavourable. STAHLCORD® conveyor belts with textile transverse reinforcements differ only insignificantly from standard steel cable belts in their troughability. In special cases the troughability can be determined in our laboratory by means of a rigidity test.

A minimum transverse rigidity is desirable in wider conveyor belts to enable the belt to form a stable troughing form and to withstand lateral compression stresses, too. The rigidity of fabric carcass belts is normally adequate provided that the belt is of a certain minimum thickness.

$s_1 \geq 8 \cdot B$ in mm

Belt width B in m
Belt thickness s_1 in mm

An increase in cover gauges contributes to an adequate ratio between belt rigidity and belt width, particularly in the case of STAHLCORD® conveyor belts.

D – 4.3.3 Centre distance and takeups

It is the elongation properties of the conveyor belt that determine the size of the takeup that has to be taken into account when designing the takeup gear. If the conveyor belt was tensioned on assembly to such an extent that the sag was largely eliminated, it is only the further takeup for absorption of the elastic strain and permanent elongation that is necessary for those belt tensions required for operation.

The elastic strain and those permanent elongations that may occur in the course of operation can be applied as guide values for the minimum takeup with a mean safety factor of S = 10.

Average elongations with 10% belt strength utilization

Belt type	reference elongation	growth	total reference elongation ε
Fabric carcass belts (B)	1.5 – 2 %	0.2 – 0.4 %	1.7 – 2.4 %
Fabric carcass belts (up to EP 200) (from EP 250)	 0.6 – 0.9 % 0.9 – 1.2 %	 0.5 – 0.8 % 0.8 – 1.6 %	 1.1 – 1.6 % 1.6 – 2.8 %
Steel cable belts (St)	0.1 – 0.3 %	–	0.1 – 0.3 %

This produces an ever-adequate takeup guide value for tensioning at the takeup pulley.

$$L_V = \frac{\varepsilon}{100} \cdot L$$

As the belt is generally strained with 10% of its strength on only a small part of its length, precise values of L can be determined by means of the average belt tension. In this instance a procedure is adopted similar to the one outlined in footnote D – 4.1. On request our department which advises on technical applications can carry out a calculation using the relevant characteristic values of the materials concerned.

When centre distances L are great, the use of fabric carcass belts may lead to considerable takeups. If the belt can be subsequently shortened, the permanent elongation component can be ignored. It is nevertheless usually advisable to use STAHLCORD® conveyor belts when centre distances exceed 500 m, unless other aspects are against this. Even when centre distances are below 500 m, the designer should consider whether a more favourable layout of the takeup gear can be achieved with STAHLCORD® conveyor belts.

D – 4.3.4 Effects of material handled

The tension member is protected from direct effects of the material handled by covers of corresponding quality and gauge and possibly by solid rubber edges. It is important, however, to use if possible a tension member material that offers sufficient resistance even when the cover is accidentally damaged.

Significant factors include:

Moisture
Temperature
Chemical properties

The basic properties of various tension member materials towards these influences can be seen in the tables below.

Effect of moisture

Tension member	Shrinkage %	Absorption of moisture %	Strength against normal state	Elongation against normal state	Sustained contact with water
Cotton (B)	2-3	7.5-8	10 – 15% higher	approx. 10% higher	Possible rotting
Polyester (E)	0	0	No change*	No change	No rotting
Polyamide (P)	0	4	approx. 10% lower	approx. 10% higher	No rotting
Steel (St)	0	0	No change	No change	Extensive corrosion prevention with galvanized cables (formation of rust and risk of deterioration when water is in direct contact with wires)
Viscose (Z)	approx. 4	13	approx. 30-35% lower	approx. 10% higher	Possible rotting

Chemical effects

Tension member	Reaction to acid	Reaction to alkali	Reaction to organic solutions
Cotton (B)	Destroyed by concentrated and hot diluted acids	Swelling but no destruction	Resistant
Polyester (E)	Highly resistant to most mineral acids	Dissolution in strong, hot alkalis	Generally resistant; soluble in certain phenolic compounds
Polyamide (P)	Destruction by hydrochloric acid, formic acid and acetic acid	Good resistance	
Steel (St)	Destruction by acids		Resistant

* See also footnote on page 84.

Effect of temperature

Tension member	Elongation and strength tendency on sustained subjection to high temperatures
Cotton (B)	Degradation from 120°C
Polyester (E)	Only slight degradation* up to 120°C; strong degradation from 180°C
Polyamide (P)	No degradation from 80-170°C; marked drop in strength from 170°C
Steel cable (St)	No effect on steel itself but drop in pull-out resistance of steel cable from core rubber

* If the tension member is simultaneously subjected to the effect of moisture, strong degradation can already occur at lower temperatures - from approx. 90°C.

D – 4.3.5 Feeding strain

The conveyor belt is subjected to considerable strain by the impact of the material handled at the feeding point. The frequency with which this strain occurs is particularly great in systems with a short centre distance. The covers normally provide adequate protection against damage to the tension member. In cases of extra coarse lumps and long falls, however, it may be necessary to select the load carrying elements not only in accordance with the drive tractions but also to take into account the impact strain due to the material handled. A measure of this is the drop energy that has to be absorbed by the conveyor belt.

$$E_F = m_k \cdot K_f \cdot g \cdot h_F \qquad \text{in Nm}$$

with m_K – mass of an approximately cubic lump with edge length k
K_f – shape factor for lumps of different shapes
h_F – height of free fall

Guide values for mass m_k of cubic lumps in kg

Material handled	Max. lump size k in mm							
	100	200	300	400	500	600	700	800
Lignite, Coke, coal	0.15	1.5	4.5	10	20	35	55	80
Salt, slag	0.25	2.0	6.5	15	30	50	80	120
Limestone, gravel	0.3	2.5	8.5	20	40	70	105	160
Overburden, refuse	0.4	3.0	10.5	25	50	85	135	200
Ore	0.55	4.5	14.5	35	70	120	185	280

With a specific edge length k*, lumps of different shapes also have different volumes. To determine the mass approximately, the shape factor K_f can be applied.

Shape factor K_f for different lump shapes

oblong	cubic	round
0.5	1	2.5

Apart from the drop energy, it is the surface texture of the lumps that is of decisive significance.

* Contrary to geometric bodies, the edge length k is used here to denote the greatest body diagonal of a lump.

Guide values for selection of the tension member according to height of feed strain

Drop energy E_F in Nm	Surface texture					
	Rounded, soft (e. g. lignite)		Irregular, angular (e. g. limestone)		Sharp-edged, hard (e. g. granite)	
	(EP)	(St)	(EP)	(St)	(EP)	(St)
< 10	< 315		< 400	< 630	< 630	< 1000
10 – 100	315 to 400		400 to 630	630 to 800	630 to 800	1000 to 3150
100 – 500	400 to 630	< 630	630 to 1000	800 to 1250	800 to 2000	3150 to 4000
500 – 1000	630 to 800	630 to 800	1000 to 1600	1250 to 2000	2000 to 2500	> 4000
1000 – 2000	800 to 1250	800 to 1250	1600 to 2500	2000 to 3150		
2000 – 4000	1250 to 2000	1250 to 2000		3150 to 4000		
4000 – 6000	2000 to 2500	2000 to 2500		> 4000		
6000 – 10 000		2500 to 3150				
> 10 000		> 3150				

Additional measures at the feeding point, such as yielding idlers* or a bedding layer of material help to reduce feeding strain. In all instances, covers of adequate thickness are to be provided on the top. STAHLCORD® conveyor belts give a better performance than fabric carcass belts, particularly when the cable diameter and pitch are selected correctly in relation to the nature of the material to be handled too. In cases of extreme strain, transverse reinforcements may offer extra protection (cf. Section D – 5.3).

* Impact idlers (cf. D – 3.1) are included in ContiTech's production range.

D – 4.4 Tension member data

The text below presents the essential features of the respective tension members in a selection from the ContiTech range of conveyor belts.

Fabric carcass belts, where standardized, comply with DIN standard 22 102 and equivalent standards. The tension members stated are primarily suitable for surface application.

Special-purpose versions of ContiTech's conveyor belts with textile plies, designed mainly for surface application, are marked with the registered trade marks.

CON-MONTEX® Single-ply conveyor belts with polyester/polyamide tension members e. g. CON-MONTEX® 200/1

or

CON-BITEX® Two-ply conveyor belts with polyester/polyamide tension members e. g. CON-BITEX® 315/2

ContiTech supplies special-purpose conveyor belts with textile plies for underground mining application, which have been approved by the regional mines inspectorate. These belts meet the special demands of coal mining and are marked as special versions with the registered trade mark

CON-MONTEX® Single-ply conveyor belts for underground mining application with polyester / polyamide tension members e. g. CON-MONTEX® EbPb 1250/1

or

CON-BITEX® Two-ply conveyor belts for underground mining application with polyamide / cotton tension members (union thread) e. g. CON-BITEX® Pb 800/2

DIN standard 22 109 (February 1988) has been edited as a standard specification for textile-ply conveyor belts used in underground coal mining. The belts are approved in self-extinguishing type "V"only and have rubber or PVC covers. Approval has been granted for all strengths between 500 and 1250 N/mm and belt widths up to 1200 mm. CON-MONTEX® conveyor belts have the LOBA approval for strengths between 630 and 1250 N/mm.

All ContiTech **steel cable conveyor belts** are marked with the registered trade mark

STAHLCORD® Steel cable conveyor belts in series R 10 strength grades (according to standard DIN 22 131) e. g. St 2000

and have a structure and properties complying with standard DIN 22 131 (November 1988).

The strength grades prescribed in standard DIN 22 131 by standard series R 10 are supplemented by

STAHLCORD®-special versions Steel cable conveyor belts in series R 20 strength grades e.g. St 2800.

The tension members comply with the DIN 22 131 standard specifications in their properties and in their structure.

By using more sophisticated calculation methods and layout methods of conveyor belts, a type can be selected from this ContiTech conveyor belt range for optimum fulfilment of the demands. The breaking strength of the cables in these Series R 20 belts likewise takes into account a tensile strength allowance, so that equal safety factors are guaranteed in the application of these belts.

Constructional features of STAHLCORD® conveyor belts in accordance with DIN standard 22 129, 22 131 and in special versions

Type	Minimum breaking strength in N/mm	Cable diameter in mm (max.)	Cable pitch in mm (approx.)	Remarks
St 500	500	3.0	12.5	Special version
St 630	630	3.0	10	Special version
St 800	800	3.8	12	Special version
St 1000	1000	4.1	12	DIN 22129, 22131
St 1120	1120	4.1	11	Special version
St 1250	1250	4.9	14	DIN 22129, 22131
St 1400	1400	4.1	9	Special version
St 1600	1600	5.6	15	DIN 22129, 22131
St 1800	1800	5.6	13.5	Special version
St 2000	2000	5.6	12	DIN 22129, 22131
St 2250	2250	5.6	11	Special version
St 2500	2500	7.2	15	DIN 22129, 22131
St 2800	2800	7.2	13.5	Special version
St 3150	3150	8.1	15	DIN 22129, 22131
St 3500	3500	8.6	15	DIN 22129, 22131
St 4000	4000	8.9	15	DIN 22129, 22131
St 4500	4500	9.7	16	DIN 22129, 22131
St 5000	5000	10.9	17	DIN 22129, 22131
St 5400	5400	11.3	18	DIN 22129, 22131
St 6300	6300	12.6	18	Special version
St 7100	7100	13.5	19	Special version

ContiTech supplies steel cable belts in another series as

STAHLCORD® special versions Steel cord conveyor belts e. g. St 3150 X-A

with mainly narrower cable pitches or in strengths deviating from DIN standard 22 131. The code letters X-A refer to the structure and the splicing system, to which special attention must be paid.

The code letters signify:

A – interlaced splice (as DIN standard 22 131)
B – interlaced splice with deflection of cables
C – graduated splice (e. g. the so-called “organ pipe” splice)
D – other systems

These belts are made to specification and may offer additional advantages.

ContiTech steel cable belts equipped with transverse reinforcements in the covers (cf. D – 5.3) are marked

CONTI-CROSS® Steel cable conveyor belts with textile transverse reinforcement on both sides e. g. St 2000 T/T

Any STAHLCORD® tension member can be used for steel cable conveyor belts with textile transverse reinforcements (STAHLCORD® tension members for steel cable transverse reinforcements available on request).

For underground coal mining in the Federal Republic of Germany, approval must be obtained from the regional mines inspectorate for steel cable belts too. The core of these belts complies with the structure stipulated in DIN standard 22 129, deviating versions, however, being permitted. All types of belt must be self-extinguishing (V). ContiTech has obtained the relevant approval for STAHLCORD® conveyor belts for strengths of 1000 to 5400 N/mm and for belt widths of up to 1600 mm with and without textile transverse reinforcements.

D–4.4.1 Textile tension member

Technical data (guide values)

Type	Nominal strength in splice $k_{nom\,V}$ in **N/mm**	Breaking elongation (longitudinal) in **%**	Thickness of tension member (belt core) s_Z in **mm**	Tension member mass (without cover **M** or **N**) m_Z^{II} in **kg/m²**	Standard elongation at 10% of min. breaking strength (longitudinal) in **%**	Min. pulley diameter (drive pulley) $D_{Tr(A)}$ in **mm**	Min. belt width at 30° troughing **B** in **mm**
1-ply belts							
CON-MONTEX®	160	12–16	2.0	2.2	0.6–1.2	200	400
	200		2.0	2.4		200	400
	250		2.5	2.5		250	400
	315		2.6	3.1		315	400
	400		2.8	3.3		315	400
	500		3.0	3.7		315	400
2-ply belts (with thick intermediate rubber layer)							
CON-BITEX®	200	12–16	3.5	3.9	0.6–1.2	250	400
	250		3.7	4.1		250	400
	315		3.9	4.3		250	400
	400		4.1	4.6		315	400
	500		4.7	4.8		400	500
	630		5.2	6.1		500	500
	800		6.3	6.4		630	650
3-ply belts							
EP	125	12–16	3.1	3.6	0.6–1.2	250	400
	200		3.0	3.2		315	400
	250		3.2	3.6		400	400
	315		3.5	3.8		400	500
	400		3.8	4.3		500	500
	500		4.7	4.6		500	650
	630		5.1	6.3		630	800
	800		6.9	7.0		800	800
4-ply belts							
EP	315	12–16	3.8	4.2	0.6–1.2	400	400
	400		4.2	4.7		500	500
	500		4.6	5.0		500	500
	630		5.0	5.7		630	650
	800		6.2	6.1		800	800
	1000		6.8	8.4		800	1000
	1250		9.1	9.3		1000	1000
5-ply belts							
EP	400	12–16	4.7	5.3	0.6–1.2	500	500
	500		5.2	5.9		630	500
	630		5.7	6.2		630	650
	800		6.2	7.1		800	800
	1000		7.7	7.6		1000	1000
	1250		8.5	10.5		1000	1200
	1600		11.4	11.6		1250	1200

Further belt constructions available on request

D – 4.4.2 Steel cable tension members

Technical data (guide values)

Type	Nominal strength in splice k_{nomV} in **N/mm**	Breaking elongation (longitudinal) in **%**	Thickness of tension member (belt core) in s_Z in **mm**	Tension member mass (without cover **M** or **N**) m_Z^{II} in **kg/m²**	Standard elongation at 10% of min. breaking strength (longitudinal) in **%**	Min. pulley diameter (drive pulley) $D_{Tr(A)}$ in **mm**	Min. cover gauge in **mm**
STAHLCORD® to DIN 22131 specifications and in special versions (Series R 20)							
St 500	500	2-3	2.9	6.3	0.1-0.3	500	3
St 630	630		2.9	6.7		500	3
St 800	800	2-3	3.6	8.2	0.1-0.3	630	3
St 1000	1000		4.0	9.6		630	3
St 1120	1120		4.3	10.1		630	3
St 1250	1250		4.8	10.6		630	3
St 1400	1400		4.0	11.2		630	3
St 1600	1600	2-3	5.5	13.0	0.1-0.3	800	4
St 1800	1800		5.5	15.2		800	4
St 2000	2000		5.5	15.8		800	4
St 2250	2250		5.5	16.5		800	4
St 2500	2500	2-3	7.1	18.6	0.1-0.3	1000	5
St 2800	2800		7.1	19.9		1000	5
St 3150	3150	2-3	7.9	22.5	0.1-0.3	1250	5.5
St 3500	3500		8.4	24.0		1250	5.5
St 4000	4000	2-3	8.9	27.0	0.1-0.3	1250	6.5
St 4500	4500	2-3	9.6	30.9	0.1-0.3	1400	7
St 5000	5000		10.7	33.6		1600	7.5
St 5600	5600		11.2	38.4		1600	8
St 6300	6300		12.3	40.9		1600	8.5
St 7100	7100		13.1	47.6		1800	9

STAHLCORD® special versions (Series X)							
St 1000 X	1000	2-3	3.6	8.7	0.1-0.3	500	3
St 1250 X	1250		4.0	10.6		630	3
St 1800 X	1800	2-3	4.8	13.2	0.1-0.3	800	3.5
St 2500 X	2500	2-3	5.5	16.9	0.1-0.3	800	4
St 2800 X	2800		6.1	18.4		1000	4.5
St 3150 X	3150		6.1	20.2		1000	4.5
St 3500 X	3500	2-3	7.9	22.7	0.1-0.3	1250	5
St 4000 X	4000		8.4	25.7		1250	6
St 4500 X	4500		8.9	27.9		1400	6.5
St 5000 X	5000		9.6	31.1		1400	7
St 5400 X	5400	2-3	10.7	33.7	0.1-0.3	1600	7.5

Further belt constructions available on request

D – 4.4.3 Splice dimensions

An open-ended belt is generally spliced and part lengths joined together by means of vulcanization; the belt tensions are then transmitted from one load carrying element to the other by shearing stresses in the rubber. The tension member is determined with regard to the durability of the splice (cf. D – 4). The nominal strength in the splice $\mathbf{k_{nomV}}$, assumed in the designing process, results from the splicing system, since single plies or cables of the tension member may be butt-jointed. The theoretical strength of any splice construction (nominal strength $\mathbf{k_{nomV}}$ in the splice) can be determined by this means.

Losses in strength, that may arise in the splice and cannot be precisely calculated, are taken into account in the safety factors applied in the design. These include, for instance, thermal effects dependent on operating time or fatigue of the rubber between the load carrying elements. Special core rubber compounds have therefore been developed by ContiTech for increased safety.

In the case of **fabric carcass belts** (standard version multiple ply belts) the following correlation is yielded between the minimum breaking strength of the belt $\mathbf{k_{nom}}$ and the nominal strength $\mathbf{k_{nom}}$ bof the splice, $\mathbf{k_{nom}}$ being rounded off to the next standardized strength grade upwards.

$$\mathbf{k_{nomV} \approx \frac{z}{z + 1} \cdot k_{nom}}$$

z - Number of splice laps

3-ply belt (Two-lap splice)

CON-BITEX® (Single-lap splice)

For **CON-MONTEX®** and **CON-BITEX®**

$$\mathbf{k_{nomV} = k_{nom}}$$

can be set in the case of overlapping or of an additional ply being inserted in the splice area.

CON-MONTEX® (Splice with overlap)

The strain on the splice is reduced on flexing across idlers and pulleys by means of a bias with a length of

$$L_A = 0.3 \cdot B$$

The additional belt length for the splice overlap is thus

$$L_{Vz} = z \cdot L_S + L_A$$

for fabric carcass belts with number of laps **z**.

The lap length L_S depends on the strength of the individual fabric plies.

Splice dimensions in fabric carcass belts

Type		Lap length L_S	Number of laps **z**		Additional belt length for overlap L_{Vz} in **mm**		
CON-MONTEX®	160/1	200		200			
	200/1	250		250			
	250/1	250	1	250	+ 0.3 · B		
	315/1	300		300			
	400/1	300		300			
	500/1	350		350			
CON-BITEX®				Single-lap or with fabric ply		Two-lap or interlaced	
	200/2	180		180		360	
	250/2	180		180		360	
	315/2	180		180		360	
	400/2	200		200	+ 0.3 · B	400	+ 0.3 · B
	500/2	200		200		400	
	630/2	250		250		500	
	800/2	250		250		500	
B EP	200/3	100		200			
	200/3	100		200			
	250/3	150		300			
	315/3	150		300			
	400/3	200	2	400	+ 0.3 · B		
	500/3	200		400			
	630/3	250		500			
	800/3	250		500			
	1000/3	300		600			
	1250/3	300		600			
B EP	250/4	100		300			
	400/4	150		450			
	500/4	200		600			
	630/4	200	3	600	+ 0.3 · B		
	800/4	250		750			
	1000/4	250		750			
	1250/4	300		900			
	1600/4	300		900			
B EP	315/5	100		400			
	500/5	150		600			
	630/5	200		800			
	800/5	200	4	800	+ 0.3 · B		
	1000/5	250		800			
	1250/5	250		1000			
	1600/5	300		1000			

Nominal strengths for mechanical fasteners, as used mainly in underground mining, are specified for example in DIN standard 22 109.

Steel cable conveyor belts are spliced by placing the steel cables side by side alternately. There is theoretically no loss of strength in standard versions of these splices as all cables of one side are joined with cables of the other side by vulcanization of the core rubber.

$k_{nomV} \approx k_{nom}$

Splices according to DIN 22 131

Splice with deflection of cables

The thickness of the residual core rubber between the cables is of decisive significance for the fatigue strength of the splice. The essential minimum thicknesses of the intermediate rubber are guaranteed in all events with ContiTech **STAHLCORD**® conveyor belts manufactured to DIN 22 131 standard specifications and in special versions.

STAHLCORD® conveyor belts too are normally spliced at a bias of

$$L_A = 0.3 \cdot B$$

Butt joints are permissible too, however, provided that appropriate equipment is available. DIN standard 22131 (Sheet 4) furthermore provides for short splicing.

Recommended splice dimensions for steel cable conveyor belts manufactured to DIN 22131 specifications and in special version (guide values)

Type	Lap length L_S in **mm**	Number of laps **z**	Additional belt length for overlap L_{Vz} in **mm**		Min. thickness of intermediate rubber in **mm**
St 500	550	1	550	+ 0.3 · B	3.2
St 630	550	1	550		2.0
St 800	300	1	600	+ 0.3 · B	2.3
St 1000	300	1	600		1.9
St 1120	350	1	650		1.7
St 1250	350	1	650		2.1
St 1400	350	2	1000		1.9
St 1600	450	1	750	+ 0.3 · B	1.9
St 1800	400	2	1150		3.5
St 2000	400	2	1150		2.4
St 2250	400	2	1150		1.8
St 2500	500	2	1350	+ 0.3 · B	2.5
St 2800	550	2	1450		2.0
St 3150	650	2	1650	+ 0.3 · B	1.9
St 3500	650	3	2350		2.6
St 4000	750	3	2650	+ 0.3 · B	2.3
St 4500	800	3	2800	+ 0.3 · B	2.3
St 5000	900	4	4050		2.7
St 5400	1000	4	4450		2.3
St 6300			Splice constructions are designed to meet the specific circumstances		
St 7100					

Other splicing systems are possible

Tests resulting in improved splices have been conducted by ContiTech in order to attain optimum durability of the splice while at the same time minimizing both belt length required as well as expenditure on operation and equipment. The craftsmanship of the execution is moreover of decisive importance. Detailed information is given in Section E of the conveyor belt service manual.

D – 5 Selection of covers

➲ **Cover material 5.1**

➲ **Cover gauge 5.2**

Abrasion resistance 5.2.1
Impact resistance..... 5.2.2

➲ **Special cover structures 5.3**

The main function of covers is to protect the imbedded tension member. To fulfil this requirement the Cover material must offer adequate resistance to any chemical and physical corrosive action. Rubber compounds and possibly synthetics can be adapted within broad limits to the specific demand by an appropriate composition or recipe. Cover materials accordingly have varying principal components.

Principal components of rubber materials

Standard international symbol	Type of rubber
NR	Natural rubber
SBR	Styrene butadiene rubber
CR	Polychloroprene rubber
NBR	Nitrile rubber
IIR	Butyl rubber
BR	Cis-Polybutadiene rubber
SIR	Silicone rubber
EPDM	Ethylene propylene terpolymer rubber

The covers of a conveyor belt must be sufficiently thick to withstand gradual wear even for long periods and to protect the tension member from mechanical injury. Moreover, the steel cable tension member in particular requires certain minimum cover thicknesses in order to ensure secure imbedding of the cable ends in the splice area. The covers can also be specially dimensioned to attain a certain transverse rigidity.

The conveyor belt can be adapted to special tasks, e. g. on uphill transport, by means of patterned or cleated cover surfaces. In cases of extreme strain, transverse fabric or steel cable reinforcements can be incorporated into the covers to increase impact resistance to a certain extent. ContiTech also supplies steel cable conveyor belts permitting an electronic slit monitoring system to be used by means of conductor loops imbedded in the covers.

Basic properties of elastomers

Property	Symbol					
	NR	SBR	CR	NBR	IIR	BR
Tensile strength (**N/mm²**)	> 21	>15	> 21	> 14	> 14	>5
Hardness (Shore A)	40-90	40-90	40-95	40-95	40-95	45-80
Tear resistance	good	moderate	good	moderate	moderate	good
Abrasion resistance	good	very good	good	good	moderate	excellent
Resilience	very high	medium	medium	medium	moderate	very good
Resistance to						
diluted acids	good	good	good	mod. to good	excellent	good
concentrated acids	mod. to good	mod. to good	good	moderate	excellent	moderate
aliphatic hydrocarbons	poor	poor	moderate	excellent	poor	very poor
aromatic hydrocarbons	poor	poor	poor	good	poor	very poor
ketones	good	good	poor	poor	good	good
lacquer solvents	poor	poor	poor	moderate	poor	poor
lubricating oils	poor	poor	moderate	very good	poor	poor
oil and petrol	poor	poor	moderate	excellent	poor	poor
animal and veg. oils	poor	poor	good	excellent	poor	poor
Resistance to						
water absorption	good	good to very good	good	mod. to good	very good	good
oxidation	moderate	good	excellent	good	excellent	moderate
ozone	moderate	moderate	excellent	moderate	excellent	moderate
sunlight	poor	poor	good	poor	very good	moderate
combustion	poor	poor	good	poor	poor	poor
high temperatures	moderate	good	very good	very good	excellent	good
low temperatures	excellent	good	moderate	good	good	excellent

The chemical and physical properties cannot usually be affected independently of each other, so that any rubber material has certain features due to the components used.

D – 5.1 Cover material

The basic qualities or essential properties are stipulated in corresponding standard specifications (DIN) and are accordingly identified by letters.

Code letters

Property	Steel cable belts (DIN 22 131)		Fabric carcass belts (DIN 22 102)
noninflammable		F	
antistatic		E	
noninflammable an antistatic	K		S
self-extinguishing and antistatic		V	
resistant to high temperatures		T	
resistant to low temperatures		R	
resistant to oil and grease		G	
belts for foodstuff			A
belts for chemical products			C
belts with fixed mechanical parameters	W, X, Y		W, X, Y, Z
special qualities		X	

The mechanical parameters stated in the standard specifications are far exceeded in part by corresponding ContiTech Cover materials.

Cover material	K	W	X	Y	Z
Tensile strength, longitudinal in **N/mm²**	≥ 20	≥ 18	≥ 25	≥ 20	≥ 15
Breaking elongation longitudinal in **%**	≥ 400	≥ 400	≥ 450	≥ 400	≥ 350
Abrasion in **mm³**	≤ 200	≤ 90	≤ 120	≤ 150	≤ 250

Properties of Continental Cover materials most widely used (guide values)

Code letter to DIN standard	ContiTech quality designation	Density ρ in kg/dm³	Suitability	Admissible temperature** in °C Sustained load min	Sustained load max	Peak load min	Peak load max	Resistant to oil and grease	Tension member	Colour	Polymer
X	ATRB	1.09	general conveying (peak requirements)	-50	+60	-55	+70	no	any type	black	NR
Y	Conti extra	1.13	general conveying	-30	+60	-35	+70	no	any type	black	NR/SBR
Y	NQ	1,1	general conveying	-30	+60	-35	+70	no	any type	black	SBR
***	ContiClean® (dirt repellent)	1.12	cases of heavy soiling (with high moisture content)	-50	+60	-55	+70	no	any type	black	BR/NR
Y	ContiClean®		Transport of desul-phurised gypsum	-40	+60	-45	+70	no	any type	black	SBR
S, K	FW (non-inflammable)	1.38	underground and surface	–	+60	–	+70	no	any type	black	SBR
V	V (self-extinguishing)	1.42 1.60	underground mining (most stringent safety regulations)	–	+100	–	+110	to some degree	fabric only	black	CR
S, T	VULKAN spezial (non-inflammable)	1.39	hot materials with glowing embers	-30	+110	-30	+130	to some degree	any type	black	CR
T	VULKAN T 190	1.13	hot materials	-30	+130	-40	+150	no	any type	black	SBR
T, C*	VULKAN T 200	1.05	hot materials, also resis-tant to acid and alkali	-30	+180	-40	+200	no	fabric only	black	EPDM
T	TCC	1.2	hot materials	-30	+180	-30	+200	no	any type	black	CR/IIR
G	OIL	1.19	oily or greasy materials	-15	+100	-20	+125	yes	any type	black	NBR
G	OIL GY	1.17	oily or greasy materials	-10	+70	-20	+90	yes	fabric only	black	NBR
G	ATBX	1.13	oil sand handling	-50	+60	-50	+70	yes	any type	black	NBR
A	OIL GA	1.21	foodstuffs (in oily, greasy or extreme temperature conditions)	-10	+120	-15	+140	yes	fabric only	white	NBR

Further Cover material and data on request.

* For application under chemical reactions we recommend detailed consultation with our Application Technique Department.
** The temperature specifications refer to the conveyor belt. The carrying capacity also depends on the nature of the temperature effect (see WdK guidelines, No. 339).
*** ContiClean® is a special quality with good operative properties.

D – 5.2 Cover gauge

The factor to be ascertained when determining the cover gauge is the **minimum thickness of the cover layers** on the top and bottom of the belt. This can be taken for STAHLCORD® conveyor belts from the relevant tables (cf. D – 4.4.2). In the case of fabric carcass belts the minimum thickness depends on the structure of the fabric and is between 1 and 2 mm. A rubber cover may be dispensed with completely in special applications, e. g. in belts for sliding bed operation.

An additional amount beyond the minimum thickness is to be provided as a **wearing cover**, depending on the operating conditions and the abrasive properties of the material conveyed.

Additional (wearing) cover thicknesses on the top of conveyor belts

Loading conditions	**Loading frequency**	Properties of bulk material conveyed		
		GRANULAR SIZE		
		fine	medium	coarse
unfavourable with high relative speeds, undamped drops and extreme capacities	frequent on short systems with low belt circulation time and primarily continuous operation or high estimated belt service life	DENSITY		
		light	medium	heavy
		AGGRESSIVENESS		
		low	moderate	high
unfavourable	frequent	3 - 6	6 - 10	> 10
moderate	moderate	1 - 3	3 - 6	6 - 10
favourable	infrequent	0 - 1	1 - 2	3 - 6

The cover gauges of STAHLCORD® conveyor belts can be selected asymmetrically. An asymmetric version results when a greater cover thickness has to be provided on the top to counteract the wear to be expected. If fabric carcass belts are to have asymmetrical cover gauges, the top: bottom ratio $\mathbf{s_2/s_3}$ should not exceed **3/1**. If the differences in the thicknesses of the covers are too great, then the natural shrinkage of the rubber after being vulcanized can lead to harmful tensions in the belt.

D – 5.2.1 Abrasion resistance

The cover abrasion caused by the material handled at the feeding point can be allowed for approximately if data on the abrasion occurring are available from first-hand knowledge or if values can be assessed with adequate accuracy.

Extensive investigations, in which numerous thickness gauging were made on ContiTech conveyor belts with highly abrasion-resistant covers, yielded empirical abrasion values related to 100 000 revolutions of the conveyor belt and based on the following classification of the material conveyed and the feeding conditions:

Material conveyed		**Feeding conditions**	
Grade 1:	grain size 0-50 mm; high proportion of fine material; low unit weight; minimum abrasion	Favourable:	up to 50 cm fall; low relative speed; cushioned bulk material feed; additional abrasion-reducing measures
Grade 2:	grain size 50-150 mm; standard grain size distribution; medium unit weight	Standard:	50 to 250 cm drop; medium relative speed; optimum bulk material control
Grade 3:	grain size 150-400 mm; sized material; high unit weight; sharp corners; aggressive	Unfavourable:	over 250 cm fall; high relative speed; no cushioning and steering measures

Applications are naturally conceivable in which single criteria from different groups or grades are applicable. In such cases the influences are to be weighed up.

Guide values s_W in mm abrasion, related to 100 000 belt revolutions

Feeding conditions	Material conveyed		
	Grade 1	Grade 2	Grade 3
favourable	0.4	0.6	1.4
standard	0.8	1.4	2.4
unfavourable	1.6	2.6	4.0

The necessary cover gauge s_2 on the top results with s_W from an approximation formula

$$s_2 \geq \text{min. cover gauge} + \frac{3.6 \cdot v \cdot s_W \cdot t}{200 \cdot L} \quad \text{in mm}$$

whereby

- v – conveying speed in m/s
- s_W – abrasion in mm (related to 100 000 revolutions)
- t – planned operating period in hours
- L – centre distance in m

The belts of conveying systems with very short centre distances and high conveying speeds (e.g. belts for heavy mining equipment) are subject to extra heavy wear and therefore attain in general a relatively short operating period, resulting from the cover gauge that is technically still feasible. ContiTech application engineers will be pleased to give advice on such matters.

D – 5.2.2 Impact resistance

A thicker cover makes an essential contribution to increased impact resistance in the range of minor drop energies and shorter edge lengths up to 400 mm. This can be confirmed by laboratory tests.

Greater cover gauges are recommended particularly for steel cable belts handling coarse-grained or sharp-edged material of dimensions approaching that of the cable pitch. Recommendations for cover gauge selection can be provided by test results and by practical experience.

Guide values of cover gauges for steel cable conveyor belts (top cover) in mm

D – 5.3 Special cover structures

Besides conveyor belts with smooth covers, ContiTech offers special-purpose versions for specific applications. These include conveyor belts with patterned or cleated surfaces.

Patterned or cleated covers

Type	Available belt width B in mm	Tension member	Suitability	Covers
Conveyor belt with herringbone pattern	100 to 1200	Fabric, at least 2 plies	Bulk material and piece goods; improvement in max. conveying angle by approx. 5°	All qualities apart from PVC top at least 2 mm
Belt with chevron cleats	400 to 1200	Fabric, at least 2 plies	3-roll troughing for steep angle transport of bulk materials and piece goods (up to approx. 30°) and grain sizes up to 50 mm	ATRB, Conti extra, ContiClean®, OIL GY, FW, VULKAN T 150, top at least 2 mm, bottom at least 1 mm
			2-roll troughing for steep angle transport of bulk materials (up to approx. 30°) and grain sizes up to 150 mm	
Conveyor belt with 1 wedge guide on top cover	to 1400	Fabric, at least 2 plies	Short, wide belts on conveying systems with problematic tracking properties	All qualities apart from PVC
Conveyor belt with 1 wedge guide on top cover	1400 to 2800	Fabric, at least 4 plies		

For further data and other types for steep angle conveying, elevators and piece goods conveyors, cf. D – 6, D – 7 and D – 8.

The covers can be provided with special plies to improve loading properties. Transverse reinforcements primarily increase impact resistance but also contribute to increased longitudinal slit resistance.

Transverse reinforcements of covers

Type	Transverse reinforcement	Arrangement	Tension member
CONTI-CROSS T	Textile cord (Polyamide)	on one side (top or bottom)	STAHLCORD (all types)
CONTI-CROSS T/T	Textile cord (Polyamide)	on both sides	STAHLCORD (all types)
CONTI-CROSS S/S	Steel strands	on both sides	STAHLCORD-special purpose types
Breaker	Fabric (Polyamide)	on one side (top) or on both	Fabric carcass

As an alternative ContiTech supplies STAHLCORD® conveyor belts with conductor loops vulcanized into the bottom cover and forming an electronic slit warning system in conjunction with a transmitter and a receiver. If a conductor loop is broken by any slitting, the conveying system is stopped due to the modified signal picked up by the receiver, so that the extent of the slit damage is restricted. Detailed information on the arrangement, function and maintenance of the conductor loops is obtainable from ContiTech's Application Technique Advisory Service.

D – 6 Steep angle conveyor belts

➲ **Chevron cleated belt 6.1**

➲ **Box section belt..... 6.2**

Conveying capacity..... 6.2.1
Constructions data 6.2.2

➲ **Fin-type belt and belt with bonded partitions 6.3**

Conveying systems with steep angle belts are calculated and designed in principle in the same way as belt conveyors equipped with conventional belts and thus suitable for maximum gradients of approx. 20°.

Variations arise, however, in the determination of the Conveying capacity of belts fitted with high cleats and consequently suitable for gradients of up to 70°.

The theoretical volume capacity $\mathbf{Q_{Vth}}$ is influenced by the section spacing $\mathbf{t_Q}$ and the height $\mathbf{h_Q}$ of the cleats. With the basic spacing

$$t_g = \frac{h_Q}{\tan(\delta - \beta)} \qquad \text{in mm}$$

the volume capacity results from the basic filling and the overfilling (cf. D – 6.2.1). The Conveying capacity is increased for $\mathbf{t_Q < t_g}$.

Conveying capacity increase in %

t_Q / t_g	0.5	0.75
increase by	approx. 50%	approx. 25%

For fine grained materials the section spacing $\mathbf{t_Q}$ should be selected between 200 and 500 mm. For coarse bulk materials a section spacing $\mathbf{t_Q}$ of at least 1.5 to 2 x the maximum edge length is recommended.

The belt velocities are to be stipulated with regard to feeding conditions, to the graining of the material conveyed and to the conveying angle. The feed should be effected centrically in the conveying direction and the material fall kept as low as possible.

Velocities for steep angle conveyor belts in m/s (guide values)

	Grain size (edge length)		
Gradient	**0-50 mm**	**50-150 mm**	**150-300 mm**
up to 35°	up to 2.0	up to 1.6	up to 1.3
35°-50°	up to 1.6	up to 1.3	up to 1.0
50°-70°	up to 1.0	up to 0.8	up to 0.6

Power requirements and drive tractions can be calculated with sufficient accuracy as for ascending conveyor systems. As the gradient is generally steeper, however, the primary resistance is reduced in contrast to a horizontal system. The majority of the motional resistances are usually formed by the gradient resistance $\mathbf{F_{St}}$.

One common feature of all steep angle conveyor belts is that their partitioning prevents them from being cleaned by scrapers. When viscous or heavily caking material is being handled, soiling of the system by the returning side has to be expected.

D – 6.1 Chevron cleated belt

The ContiTech chevron cleated belt is a steep angle belt with 15 mm high chevron cleats pressed out of the Cover material, sidewalls and 55 mm wide, plain borders permitting chuting at the feeding point.

The theoretical volume capacity can be determined from the table for conventional belt troughings or read off from the diagram below.

Technical data for chevron cleated belts *

Tension member	Fabric, at least 2 plies (STAHLCORD tension members on application)
Cover material (cleats are of the same material)	ATRB Conti extra ContiClean® OIL GY FW VULKAN T 150
Minimum cover gauge	Top cover 2.0 mm Bottom cover 1.0 mm
Cleat spacing Cleat length Cleat width	192 mm 80 mm 8 mm
Belt mass allowance for cleats	Corresponding to 1 mm additional cover thickness
Widths available	400 mm 500 mm 650 mm 800 mm 1000 mm 1200 mm up to 2400 mm on request
Minimum pulley diameter	250 mm (minimum pulley of selected tension member to be observed)

* A significantly expanded product range with a profile height of 10 – 35 mm is in part available ex stock. (Please ask for special documentation)

D – 6.2 Box Section Belt

This ContiTech steep angle conveyor belt is available under the designation

CONTIWELL® / MAXOFLEX®

in various dimensions and designs, thus offering optimum adaptation to any application.

The sidewall belt consists of an untroughed conveyor belt with corrugated sidwalls mounted longitudinally and lateral profiles mounted between the sidewalls with certain spacing.

The corrugated sidewall is distinguished by high ductility, permitting relatively small pulley diameters and deflection radii. The high lateral stability of the corrugated sidewall permits the return run to be supported on straight idlers.

Dimensions of the corrugated sidewall and technical data

Dimensions in mm	Type			
	W 63	W 80	W 125	W 160
Corrugated sidewall height h_K*	60	80	125	160
Base width b_F	60		80	
Corrugation depth b_W	37	46	52	62
Corrugation pitch t_W	60		70	
Minimum pulley diameter (to suit tension member)	200		315	
Sidewall mass in kg/m (approx.)	1.25	1.5	2.5	3.2

For gradients of up to approx. 20°, corrugated sidewalls can be used in conjunction with an untroughed belt, even without transverse partitions. This provides a considerable increase in conveying capacity as compared with a flat belt (cf. D – 2.5). The effective belt width beif results in the same way as for a CONTIWELL® box-section belt.

The corrugated sidewall can be supplied in various rubber qualities adapted to the special cover compounds and offering simultaneously optimum flexibility.

Corrugated sidewall types

Application	Maximum temperature	ContiTech cover	Tension member (with transverse rigidity)
general conveying (antistatic $< 3 \cdot 10^8 \Omega$)	60 °C	Conti extra	any type
handling of hot materials	90°C	VULKAN T 150	any type
handling of hot materials (oil-resistant)	90°C	VULKAN spezial	any type

Belts with fabric plies (at least 3 plies) or STAHLCORD® tension members can be used, depending on the required tensile strength.

B – cotton
EP – polyester/polyamide
St – STAHLCORD®

If the system is designed with bends in the flight routing (cf. D – 6.2.2), it should be borne in mind that the deflection and transmission of the belt tensions are effected first and foremost in the idler-supported or pulley supported plain edge zones, as the belt can be supported only partially by the corrugated sidewalls.

* MAXOFLEX® sidewalls with a profile height of 40 – 400 mm are in part available ex stock. (Please ask for special documentation)

D–6.2.1 Conveying capacity

The belt gradient and the coarsest grain size of the material conveyed are to be taken into account when selecting a CONTIWELL® box-section belt from the types available.

Guide values for the admissible grain size (max. edge length) in mm

Type	Gradient		
	up to 35°	35°–50°	50°–70°
WK 63	120	90	60
WK 80	150	120	80
WK 125	200	150	100
WK 160	300	220	120

The theoretical volume capacity $\mathbf{Q_{Vth}}$ can be calculated with transverse partition interval $\mathbf{t_Q}$ and basic spacing $\mathbf{t_g}$ (cf. D–6). For $\mathbf{t_Q \geq t_g}$ is

$$Q_{Vth} = \frac{h_Q \cdot b_{eff} \cdot t_g}{2\ t_Q} \cdot v \cdot 3600 \qquad \text{in } m^3/h$$

The Conveying capacity can be estimated by reading off the volume capacity from the diagrams below.

Theoretical volume flow for v = 1 m/s and 100 mm effective belt width

D – 6.2.2 Constructional data

CONTIWELL® box-section belts can be guided through extremely narrow transition curves and thus adapted to almost any routing.

Support in top run

Supporting methods in the return run

Supporting methods in the bend

Approximate values for construction data

Sidewall height	Idler diameter D_1 in mm		Idler diameter D_2 in mm		Idler width B_1	Idler width B_2 in mm		non-profiled border zone b_1 in mm		border zone b^2	dis-tance b^3
h_K [mm]	B≤650	B>650	B≤650	B>650	in mm	B≤650	B>650	B≤650	B>650	in mm	in mm
60	140	240	270	370	55	50	70	65	85	7,5	30
80	140	240	310	410	55	50	70	65	85	7,5	30
125	140	240	410	510	70	60	80	75	95	7,5	30
160	140	240	480	580	70	60	80	75	95	7,5	30

In this case $2 \cdot (b_1 - 10)$ mm is to be deducted from the service width b_{eff}.

The minimum admissible pulley diameters or deflection radii are normally determined by the tension member. For belt guidance through concave curves, CONTIWELL® box-section belts are available in special versions with plain borders.

Belt widths and partition layouts (without plain borders)

Belt width* B in mm	Effective belt width beff in mm		Type	Length and mass of transverse partitions (standard quality)			
	WK 63/80	WK 125/160		WK 63 (hQ = 50 mm)	WK 80 (hQ = 70 mm)	WK 125 (hQ = 110 mm)	WK 160 (hQ = 145 mm)
400	300	280	Single compartment	260 mm 0.25 kg	260 mm 0.35 kg	220 mm 0.7 kg	220 mm 0.8 kg
500	400	380		360 mm 0.3 kg	360 mm 0.5 kg	320 mm 1.0 kg	320 mm 1.2 kg
650	550	530		510 mm 0.45 kg	510 mm 0.7 kg	470 mm 1.45 kg	470 mm 1.75 kg

The top run and return run of CONTIWELL® box-section belts can normally be supported on straight idlers. The idler diameter and spacing can be selected for the top run as for conventional systems.

Guide values for idler selection

	Idler spacing in m			Idler diameter in mm
Top run	0.8 up to 1.25			89 up to 159
	Gradient			
Return run* ** 1-part	20° – 35°	35° – 50°	50° – 70°	≥ 89
	≤ 1.0	≤ 1.2	≤ 1. 5	

In the case of special types, it may be necessary to provide smaller idler intervals or to take special steps to support the belt, depending on the gradient of the system, the weight of the belt and the type of corrugated sidewalls used. Our department responsible for conveyor belt application will be pleased to advise you in this matter.

* Greater widths available on request
*** In the case of types combining a greater belt width the given sectioning (single compartment) or having an extremely heavy tension member, the idler spacings are to be checked.

D – 6.3 Fin-type belt and belt with bonded partitions

As an alternative to the Contiwell® box-section belt, conveying systems operating at steep angles can also be equipped with fin-type or partitioned belts. On the one hand their flexible dimensioning allows them to be used for wide range of applications in general, whereas the hard wearing transverse cleats fitted to these types of belt also make them highly suited to heavy-duty operating conditions in particular.

Fin-type belts are suitable for handling medium to coarse sized bulk materials at steep angles. The top run can similarly be troughed using dual fins. The return run of the belt is supported on disc idlers.

Theoretical volume flow at 1 m/s

Design data (guide values) in mm

Belt width mm	A	B	c	d	H
1000	330	225	50	60	133
1200	415	280	50	62,5	133
1400	480	350	50	60	133
1000	320	200	60	80	200
1200	405	255	60	82,5	200
1400	470	325	60	80	200

The width of the plain zones **c** and **d** can be enlarged if necessary.

Minimum pulley diameter

400 mm	for a fin height of 133 mm
630 mm	for a fin height of 200 mm

Partitioned belts operate in a non-troughed (flat) state and are guided on the top run between fixed skirtings provided with sealing strips of rubber.

Theoretical volume flow at 1 m/s and 100 mm effective belt width

Design data (guide values) in mm

	Partition height 133 mm						Partition height 200 mm					
	B	a	b	c	d	useful belt width	B	a	b	c	d	useful belt width
1-part	400	280	60	–	–	280	400	240	80	–	–	240
	500	380	60	–	–	380	500	340	80	–	–	340
2-part	650	240	60	50	–	530	650	215	80	60	–	490
	800	315	60	50	–	680	800	290	80	60	–	640
3-part	1000	330	60	50	225	880	1000	320	80	60	200	840
	1200	415	62.5	50	280	1080	1200	405	82,5	60	255	1035
	1400	480	60	50	350	1280	1400	470	80	60	325	1240

The width of the plain zones **b** and **c** can be enlarged if necessary.

Technical data

Height of partitions	mm 133	Height of partitions	mm 200
Spacing of partitions	mm 330	Spacing of partitions	mm 330
Minimum pulley diameter	mm 400	Minimum pulley diameter	mm 630
Maximum gradient of flight	60	Maximum gradient of flight	60
Suitable for bulk materials of mixed lump sizes with a diameter of up to	mm 250	Suitable for bulk materials of mixed lump sizes with a diameter of up to	mm 350

Further details on belt applications and system design can be taken from a special publication.

D – 7 Elevator belts

➲ **Conveying capacity 7.1**

➲ **Power requirement and drive tractions 7.2**

➲ **Selection of bucket elevator belt 7.3**

Tension member 7.3.1
Covers 7.3.2
Buckets and attachments 7.3.2

Belts are finding extensive application as traction elements for bucket elevators, and here to an increasing extent elevator belts with STAHLCORD® tension members. Belt-type bucket elevators have many advantages over chain bucket elevators, whose significance is diminishing. One essential factor is the quiet, low-vibration operation that permits higher conveying speeds and thus a considerably improved performance. The development of high-strength belts with fabric or cable cord tension members means that conveying heights of 150 m and more now present no insuperable problems.

Decisive advantages are reduced power requirements and a considerable reduction in maintenance expenditure. The following section is intended to help the designer and the user to ascertain the range of applications offered by ContiTech elevator belts for bucket elevators by means of a rough layout. As most instances are single conveying systems with special features, we recommend detailed consultation with ContiTech's Conveyor Belt Application Department.

D – 7.1 Conveying capacity

The theoretical capacity $\mathbf{Q_{mth}}$ results from

$$\mathbf{Q_{mth}} = \frac{\mathbf{3.6 \cdot V_{Be} \cdot \rho \cdot v}}{\mathbf{t_{Be}}} \cdot \varphi \qquad \text{in t/h}$$

$\mathbf{V_{Be}}$	bucket volume in litres
ρ	bulk density of material conveyed in in $\mathbf{t/m^3}$
v	belt velocity in m/s
$\mathbf{t_{Be}}$	bucket spacing on belt in m
φ	average degree of fill

Greater Conveying capacity can also be achieved by using a multiple-row arrangement for the buckets. In this case. the theoretical capacity is taken from the sum of bucket volumes.

Guide values

Type and properties material conveyed	Example of material conveyed	Shape and volume of buckets $\mathbf{V_{Be}}$ to DIN specifications	Average degree of fill	Belt velocity **v** in **m/s**	Bucket spacing $\mathbf{t_{Be}}$ ($\mathbf{h_1}$ = rear height of bucket) in m
fine, light material	flour, semolina, coarse meal	DIN 15 231 DIN 15 241 “flat”	0.75-0.9	0.7-1.3	(1.5-2.0) x $\mathbf{h_1}$
light granule material	cereals, oil seeds, pulses	DIN 15 232 DIN 15 242 “with flat bottom”	0.75-0.9	1.2-2.5 (3.0)	(1.3-1.5) x $\mathbf{h_1}$
viscous material	cane sugar, wet fine coal, damp soil	DIN 15 233 DIN 15 243 “medium deep”	1.0-1.2	1.0-1.5	2.0 x $\mathbf{h_1}$
heavy pulverized to coarse lumps	sand, cement, raw meal, coal	DIN 15 234 DIN 15 244 “deep”	0.6-1.0	1.0-1.6	(1.4-2.0) x $\mathbf{h_1}$
easy flowing or rolling material	light ashes, potatoes	DIN 15 235 DIN 15 245 “deep, back wall curved”	0.7-0.8	1.2-1.6	(1.3-1.8) x $\mathbf{h_1}$

The essential data are reproduced in Section D – 7.3.4 as an example of deep buckets complying with DIN standard 15 234 or of corresponding ContiTech rubber buckets. The data for other bucket designs can be inspected in the relevant standard specifications or obtained from the manufacturer.

D–7.2 Power requirement and drive tractions

The power requirement $\mathbf{P_{Tr}}$ of a bucket elevator at the drive pulley results approximately from the lifting capacity with the capacity $\mathbf{Q_m}$

$$P_{St} = \frac{Q_m \cdot g \cdot H}{3600} \qquad \text{in kW}$$

and the ratings

$\mathbf{P_S}$	for scooping
$\mathbf{P_{leer}}$	for resistances in no-load operatoin
$\mathbf{P_z}$	for unusual influences such as batched feeding

$$P_{Tr} = P_{St} + P_S + P_{leer} + P_z$$

Apart from the primary component of the lifting capacity, it is the scooping capacity, particularly with lumpy material, that represents a significant component. An empiric formula can be applied to determine this.

$$P_s = \frac{Q_m \cdot K_s \cdot E_s}{3600} \qquad \text{in kW}$$

$\mathbf{Q_m}$ in t/h

Guide values for K_S and E_S are to be taken from the diagrams below.

Scoop factor K_S

Specific scoop work done E_S in Nm/kg

Curve	Material handled	Bulk density in t/m³	Grain size in mm
1	Portland cement	1.2	ca 0.05
2	Cereals	0.74	2...5
3	Sand, gravel	1.5	2...10
4	Cement clinker	1.25	5...20
5	Hard coal, single nuts	0.75	10...30

1 to 5 kW are to be set for no-load rating $\mathbf{P_{leer}}$ depending on bucket width, whereas up to 7 kW must be allowed for additional rating $\mathbf{P_z}$ in unfavourable cases.

The calculation of the maximum drive traction must initially be based on an estimated mass $\mathbf{m_G^I}$, of the elevator belt; this may need to be adjusted later. The diagram in Section D–3.1.2 can serve as a basis for the estimate. The belt width can be presupposed as approx. 1.1 x bucket width.

Guide values for bucket masses $\mathbf{m_{Be}}$ are stated in Section 7.3.4 or can be taken from the relevant standard specifications, resulting in

$$T_1 = T_v + H \cdot g \cdot \left(m'_G + \frac{m_{Be}}{t_{Be}}\right) + \frac{1000 \cdot P_{Tr}}{v} \qquad \text{in N}$$

In order to transmit the peripheral force to the drive pulley, which is generally located at the top in the case of a bucket elevator, a belt tension $\mathbf{T_2}$ is required; this is generated by the dead weight of the leaving side, the weight of the tail pulley and any additional tensioning force $\mathbf{F_v}$ be applied.

If

$$H \cdot g \left(m'_G + \frac{m_{Be}}{t_{Be}}\right) + \frac{m_{Tr} \cdot g}{2} < \frac{1000 \cdot P_{Tr}}{v} \left(\frac{1}{e^{\mu\alpha} - 1}\right)$$

with the mass m_{Tr} of the tail pulley, a dead load or take up screw tension must be applied in addition.

$$\frac{F_v}{2} = \frac{1000 \cdot P_{Tr}}{v} \left(\frac{1}{e^{\mu\alpha} - 1}\right) - H \cdot g \left(m'_G + \frac{m_{Be}}{t_{Be}}\right) - \frac{m_{Tr} \cdot g}{2} \qquad \text{in N}$$

In this case the belt tension

$$T_v = \frac{m_{Tr} \cdot g}{2} + \frac{F_v}{2} \qquad \text{in N}$$

Guide values for the mass m_{Tr} of tail pulley in kg

Pulley diameter in mm	Pulley width in mm							
	120	250	350	450	550	650	750	850
315	16	25	38	42	48	54	60	66
400	21	33	45	61	69	77	85	93
500	29	47	69	75	83	93	103	113
630	35	57	90	115	132	148	164	180
800	46	79	124	160	190	210	230	250

Power transmission at the drive pulley of bucket elevators is effected in accordance with the same laws as with belt conveyors. Further details are therefore omitted at this point.

D – 7.3 Selection of bucket elevator belt

Apart from the maximum belt tension, there are numerous other factors determining the selection of the belt. The belt must permit the buckets to be fastened completely satisfactorily, so that a minimum of 4 plies is recommended in fabric carcass belts. The belt must furthermore permit durable splicing, a factor of special significance in bucket elevators for safety reasons. Apart from the vulcanized splice, there are reliable mechanical fasteners.

Application features for elevator belts

Belt	Type of splice	Operating conditions
St 1000 to St 2000	Vulcanized splice for high to belt tensions and normal operating temperatures; otherwise mechanical fastener	Ideal for all operating conditions and applications ranging from low capacities to heavy-duty bucket elevators
EP 500/4 to EP 1600/5	Vulcanized splice; mechanical fastener for very high operating temperatures	For low to medium capacities: fine material: EP 500/4 (drop charging) EP 630/4 (scooping)) coarse material: EP 630/4 (drop charging) EP 800/4 (scooping))

D – 7.3.1 Tension Member

The tension member of elevator belts is constructed in the same way as that of belts for conveying systems. The essential properties and main features are thus comparable and can be taken from the corresponding sections (D – 4.4).

The necessary nominal strength $\mathbf{k_{nomV}}$ of the tension member can be ascertained with reference to a standard safety factor **S** for bucket.

$$\mathbf{k_{nomV} = \frac{T_1}{B} \cdot S} \qquad \text{in N/mm}$$

The safety factor **S** also covers the starting process, the assumption being made that a hydraulic starting clutch is installed in cases of higher power requirements. Calculations for versions with bucket-attachment perforations must be based on a higher safety factor **S** due to the reduction in strength.

Safety factor S (guide values)

Tension member	Operating temperature				
	≤ 60°C	80°C	100°C	120°C	140°C
Fabric (with vulcanized bucket attachments)	8	9.5	–	–	–
Fabric (perforated version)	9-10	11.5	12	13.5	14
STAHLCORD® (with vulcanized bucket attachments)	8	8	–	–	–
STAHLCORD® (perforated version)			9-10		

Minimum pulley diameter for customary bucket elevator belts

Type		Drive pulley diameter in mm with belt strength utilization of 60 - 100%	Drive pulley diameter in mm with belt strength utilization of < 60%	Tail pulley diameter in mm
EP	500/4	500	400	400
EP	630/4	630	500	500
EP	800/4	630	500	500
EP	1000/4	800	630	630
EP	1250/4	800	630	630
EP	800/5	800	630	630
EP	1000/5	800	630	630
EP	1250/5	1000	800	800
EP	1600/5	1000	800	800
St	1000	630	500	500
St	1250	800	630	630
St	1600	1000	800	800
St	2000	1000	800	800

Other types available on application

The pulley diameters for bolted attachment generally correspond to those for conveyor belts in the respective fabric or STAHLCORD® construction. Elevator belt tracking may be improved by cambering the drive pulley.

Cambering* h for fabric carcass belts

Pulley width in mm	Pulley diameter in mm < 125	140/160	180/200	225/250	>280
400	1.0	1.2	1.2	1.2	1.2
500	1.0	1.5	1.5	1.5	1.5
630	1.0	1.5	2.0	2.0	2.0
800	1.0	1.5	2.0	2.5	2.5
1000	1.0	1.5	2.0	2.5	3.0

The camber should always be executed as an arc. A maximum camber of 2 mm is admissible for STAHLCORD® elevator belts. If screw strips or flat irons let into the cover are used for bucket attachment, these elements must be divided, owing to the camber of the pulley.

* According to provisional figures in ISO/DIS 5286, which applies to conveyor belts in general, the maximal crowning **h** should be less than $h < 0.005 \cdot D_{Tr}/2$.
Also no crowning is permissible for steel cable conveyor belts.

D – 7.3.2 Covers

Elevator belts are not subjected to direct wear by the material handled, so that fabric carcass belts may be used without covers for subordinate purposes. In general, adherence to certain minimum cover gauges is to be recommended, the bottom covers (pulley side) usually being thicker than those of the top (bucket side).

Cover material selection

ContiTech quality designation	Property	Tension member
Conti extra	Suitable for any standard requirements, antistatic **< 3 ·10^8 Ω**, temperature resistant up to 60°C	any type
VULKAN spezial	Temperature resistant up to 110°C conditionally oil-resistant and noninflammable	any type
VULKAN T 150	Temperature resistant up to 130°C	any type
VULKAN T 200	Temperature resistant up to 180°C	any type
TDLX	White; for handling foodstuffs; resistant to oil and grease and to temperatures of up to 120°C	fabric only
FW	Noninflammable and antistatic	any type
V	Self-extinguishing and antistatic	any type

Temperature data refer to the conveyor belt, the decisive factor being the effective temperature to be withstood by the belt in sustained operation. Short-term peak temperatures may be even higher. The material being conveyed in the buckets may possibly have considerably higher temperatures.

Minimum cover gauges (guide values)

	Operating conditions	Top (bucket side)	Bottom (pulley side)
Fabric	Handling of cereals etc. in closed rooms	Rubberized surface (without covers)	
	Handling of fine-grained and light materials (cereals, sugar, sand, salt, cement, potatoes, granulates)	1 mm	1 mm
	Handling of fine-grained and powdery materials at higher temperatures (cement, raw meal) and granulated material (gravel, coal)	2 mm	2 mm
	Handling of medium to coarse lump material (gravel, stones, concrete) under damp conditions	2 mm	3-5 mm
STAHL CORD®	For all operating conditions	4-5 mm	5-6 mm

The cover gauges are to be stipulated with regard to the type of bucket attachment. The top covers can be grooved or have flat irons vulcanized in on request. Elevator belts needing to be supported on the return side are available in transversely rigid versions on application. The covers can also be supplied with vulcanized brackets.

D – 7.3.3 Buckets and bucket attachments

The main types of buckets and attachments are described in detail in corresponding standard specifications. Bucket elevators are normally fitted with sheet metal buckets and - to a lesser extent - with cast iron buckets that can be attached to the belt with a standardized bolt attachment (DIN standard 15 236).

In addition, ContiTech's production range includes rubber elevator buckets with steel frames; these weigh up to 70 % less, enabling the belt strength to be exploited more favourably. Rubber elevator buckets display excellent properties in the handling of highly abrasive material, e. g. coke, and their natural distortion provides outstanding discharge properties when strongly adhesive material is being handled.

Bolt attachments with standard plate bolts are adequate for standardized bucket shapes for light and medium duty. Heavy duty service with scooping of coarse-grained material and heavy buckets with widths exceeding 500 mm necessitates additional measures to prevent the belt from being subjected to excessive strain. In this case it is advisable to use screw strips or other attachments that can be obtained from specialized mechanical engineering firms.

One measure contributing to belt protection is the bolt attachment with an additional strip of soft rubber inserted between the rear wall of the bucket and the belt, which seals the gap between belt and bucket even on rotation round the pulley.

This measure prevents the penetration of material and in some instances of moisture. The soft rubber compound can be adapted to the Cover material.

Similar advantages are offered by bucket attachment with bonded rubber brackets. Furthermore, perforation of the belt and the consequent reduction in strength are avoided.

Guide values for bucket attachments

Minimum pulley diameter	Type of attachment
	SCHWINGMETALL® (available from Beumer machine factory only)
	Rubber attachment (flexible)
	Rubber attachment (hinged)

Data and dimensions for deep buckets with flat rear wall to DIN 15 236 specifications (selection)

Belt width in mm	Buckets to DIN 15 234 specifications (dimensions in Belt width mm)								Number of bolts	Mass per bucket in kg (approx.)		Bucket capacity in liters
	Width **b**	Pro-jection **a**	**h_1**	**c**	**d**	**i**	**n**	**r**		Metal	Rubber	
200	160	140	180	4.0	9.5	67	80	32	2 x M 8	2.1	1.5	1.5
250	200	160	200	5.0	11.5	75	125	40	2 x M 10	3.8	2.1	2.4
300	250	180	224	5.0	11.5	85	80	40	3 x M 10	5.0	2.8	3.75
350	315	200	250	5.0	11.5	95	112	40	3 x M 10	8.6	4.1	6.0
450	400	224	280	5.0	11.5	106	100	40	4 x M 10	11.8	4.9	9.5
550	500	250	315	6.5	14.0	118	100	50	5 x M 12	19.4	5.9	15.0
680	630	280	355	6.5	14.0	132	100	50	6 x M 12	26.6	7.0	23.6
850	800	315	400	6.5	14.0	150	200	50	7 x M 12	36.9	9.4	37.5
1050	1000	355	450	6.5	14.0	170	200	50	9 x M 12	50.3	–	60.0

D–8 Piece goods handling and belts for sliding bed operation

➲ **Conveying capacity 8.1**

➲ **Calculation of required power ... 8.2**

➲ **Conveyor belt design 8.3**

Belts for handling piece goods are used essentially in the range of low to medium capacities. Whereas belt support on conventional idlers is occasionally provided for long conveying systems with a relatively high load, bare-bottom-ply belts for sliding bed operation are gaining economic significance in the range of conveying lengths below 70 m and light material or single loads at greater intervals. Light bulk material too is frequently transported on sliding-bed conveyor belts when capacities are low.

The economic efficiency, however, is not only determined by the operating expenses but is always closely linked with maintenance and upkeep costs, particularly in cases of small power requirements.

ContiTech belts for sliding bed operation have proved to be wear-resistant and reliable. Most cases of failure have been due not to surface abrasion but to poor tracking or outside influences. No significant wear is to be ascertained on PVC-coated belts after long-term operation either, although all parts supporting the belt are subject to greater strain.

ContiTech conveyor belts with special-purpose slide bearings also protect the supporting trough surfaces.

The calculation procedures for piece goods conveyors, in particular of belts for sliding bed operation, differ especially in the determination of the Conveying capacity and of the motional resistances from those used for conveyor systems for bulk materials. This point is therefore given special attention in the following pages. The basic procedure stated in Sections D–3 und D–4 can be applied to determine the drive tractions.

In special instances, the design can be executed by ContiTech‘s Application Technique Advisory Service. The questionnaire attached in the appendix can be used for this purpose.

D–8.1 Conveying capacity

The flow of piece goods is determined by a conversion from the average loading of the belt conveyor with single loads, the decisive factor being the largest number of lumps that can be transported simultaneously on the flight.

With the mass $\mathbf{m_p}$ of the single piece and the quantity $\mathbf{z_p}$, the capacity is

$$Q_m = 3.6 \cdot \frac{z_p \cdot m_p}{L} \cdot v \qquad \text{in t/h}$$

If charging takes place over a longer period (larger than **L/v**) regularly at specific intervals $\mathbf{t_p}$ (smaller than **L/v**) in seconds, the capacity results

$$Q_m = 3.6 \cdot \frac{m_p}{t_p} \qquad \text{in t/h}$$

or with the spacing between the single pieces

$$d_p = t_p \cdot v \qquad \text{in m}$$

to

$$Q_m = 3.6 \cdot \frac{m_p}{d_p} \cdot v \qquad \text{in t/h}$$

When charging is irregular, the largest edge length (body diagonal) must not exceed the belt width. If the piece goods are fed to the belt conveyor with absolute alignment, the belt width can be selected in accordance with the width of the piece goods.

The tables from Section D–2.5 are applicable to the handling of light bulk materials.

If the conveyor belt is supported on idlers for piece goods handling, an idler spacing deviating from the longitudinal dimension of the material is recommended. In order to prevent excessive disturbance or impacts, the piece goods should preferably rest on more than one idler.

D – 8.2 Calculation of required power

In piece goods transport and in transport on bare-bottom-ply belts, no flexing occurs in the material handled, which means that the calculation of the motional resistances and the power requirement differs from that for bulk material conveyors. In general the ascertainment of the power requirement from the resistive force components

- F_{ro} – friction resistance in top run
- F_{ru} – friction resistance in return run
- F_{St} – gradient resistance
- F_S – special resistance

is sufficiently accurate, provided that relevant values can be inserted for the friction coefficients μ_g of the belt on the sliding bed.

Guide values for friction coefficient μ_g

Bottom of belt	Material and temperature of supporting surfaces					
	Steel plate (bright)				Plastic	Hardwood
	-20°C	0°C	+18°C	+40°C	+18°C	+18°C
EP (raw)	0.3	0.3	0.3	0.25	0.28	0.25
B (raw)	0.75	0.7	0.45	0.35	0.4	0.35
B (rubberized)			0.7		0.6	0.6
Cover rubber			0.9			

The total motional resistance **F** is then

$$F = C_g \left(F_{ro} + F_{ru}\right) + F_{St} + F_S \qquad \text{in N}$$

Resistive force components of piece goods conveyors and bare bottom-ply belts in N

Designation	Belt support sliding	Belt support on idlers (z_R = number of idlers)
F_{ro}	$\mu_{go} \cdot \left(m_L^I + m_G^I\right) \cdot L \cdot g$	$2 \cdot z_R + 0.02 \cdot \left(m_L^I + m_G^I\right) \cdot L \cdot g$
F_{ru}	$\mu_{gu} \cdot m_G^I \cdot L \cdot g$	$2 \cdot z_R + 0.02 \cdot m_G^I \cdot L \cdot g$
F_{St}	$m_L^I \cdot g \cdot H$	
F_S	The special resistance is largely dependent on the scrapers, skirt boards etc. present and is to be determined if necessary from the individual resistive forces	

The material mass m_L related to the length unit results from the capacity Q_m to

$$m_L^I = \frac{Q_m}{3.6 \cdot v} \qquad \text{in kg/m}$$

and the related belt mass can be determined with the guide values for the belt weight m_G^{II} from the equation

$$m_G^I = m_G^{II} \cdot B \qquad \text{in kg/m}$$

The required power P_{Tr} at the drive pulley is calculated from the total resistive force **F**.

$$P_{Tr} = \frac{F \cdot v}{1000} \quad \text{in kW}$$

The efficiency of the drive is to be taken into account when determining the necessary engine output P_M.

D – 8.3 Conveyor belt design

The conveyor belt used for handling piece goods must have not only appropriate covers and adequate strength. Sufficient transverse rigidity must also be present, especially for belt conveyors with idler support. The design is based on this aspect, due to the frequently low drive tractions. Tension members with 2 to 3 cotton fabric plies have proved excellent in this respect.

The power requirements determined permit the application of a single pulley drive at the head or the tail of the conveyor in almost every instance, so that the greatest belt tension results with sufficient accuracy from the required power as

$$T_{max} \approx \frac{1000\ P_{Tr}}{v}\left(1 + \frac{1}{e^{\mu\alpha} - 1}\right) \qquad \text{in N}$$

The starting process can be covered globally by a safety factor. As on the one hand a generally higher non-operating friction coefficient μ_g between the belt and the sliding bed has to be observed and the starting factor on direct connection of the generally over dimensioned drive is at K_A = 2 to 2,6, and on the other hand no additional strains occur on the belt, the safety factor to be applied must be determined approximately.

$$S \simeq 4 \cdot \frac{P_{Mnom}}{P_M} \cdot K_A$$

The necessary nominal strength k_{nomV} of the belt at the splice thus

$$k_{nomV} \geq \frac{T_{max}}{B} \cdot S$$

The friction coefficient μ between the belt and the drive pulley surface, to be assumed for the maximum drive traction, can be approximately equated with the value of μ_g.

Conveyor belts for handling piece goods and for sliding bed operation

Type		Tension member	Covers	Suitability
Conveyor belts with cleated or rubberized covers		EP-Fabric B-Fabric E-Fabric	on one or both sides; cleated or patterned on the top; cover quality depending on type	Handling of piece goods up to approx. 40° belt gradient; wood working machines; bare-bottom ply belts
Sliding plate	B2/1 B2/2 B3/1 B3/2	B-Fabric	atop raw or rubberized; bottom with cotton sliding-bed ply	Handling of piece goods; bare-bottom-ply belts; special purpose belts for the rubber industry

Other types available

D – 9 Appendix

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D – 9.2 Symbols

Symbol	Unit	Formula quantity
A	**m²**	Filling cross-section area
B	**m, mm**	Belt width
C	**–**	Length coefficient
D	**mm**	Diameter
E_F	**Nm**	Drop energy
F	**N**	Force
H	**m**	Conveying height, height
J	**kg/m²**	Mass moment of inertia
K	**–**	Factor
L	**m, mm**	Conveying length, length
M	**Nm**	Torque
P	**kW**	Power
Q_m	**t/h**	Material capacity
Q_V	**m³/h**	Material volume capacity
R	**m**	Radius
S	**–**	Safety factor
T	**N**	Drive traction
V	**m³**	Volumes
a	**m/s²**	Acceleration
a	**mm**	Projection
b, b_{eff}	**mm, m**	Effective width, base width
b_F	**mm**	Sidewall width
c	**–**	Coefficient, factor
d	**mm**	Distance, spacing
e	**–**	Natural number
f	**–**	Resistance coefficient
g	**m/s²**	Acceleration due to gravity (9.81 m/s²)
h_F	**m**	Free fall
h_{rel}	**–**	Effective height
k	**N/mm**	Belt tension (length-related force)
k	**mm**	Edge length
l	**mm, m**	Idler length, idler spacing
m	**kg**	Mass, weight
m^l	**kg/m**	Length-related mass (linear load)
m^{ll}	**kg/m²**	Area-related mass (m²-weight)
n	**min^{-1}**	Number of revolutions
p_m	**N/cm²**	Mean surface pressure
r	**mm**	Radius
s	**mm**	Thickness, cover gauge
t	**h, s**	Time
t	**m, mm**	Pitch, section spacing
v	**m/s**	Belt velocity, speed
z	**–**	Amount, number
α	**°, rad**	Angle of wrap
β	**°**	Angle of incline of material handled
δ	**°**	Angle of slope, gradient
ε	**–**	Elongation
η	**–**	Efficiency
λ	**°, rad**	Troughing angle
μ	**–**	Friction coefficient
π	**–**	Digit π (3.1416)
ρ	**t/m³, kg/dm³**	Bulk density, density
φ	**–**	Degree of fill

Indices

A	Starting, bias
B	Braking, stopping
Be	Bucket
D	Cover
E	Drive element
F	Fall, base
G	Belt
H	Principal, main
K	Edge
L	Load
M	Motor, engine
N	Secondary
Q	Cross-section
R	Idler
S	Special, stage
St	Pitch
Tr	Pulley
U	Circumference
V	Volume, splice
W	Turn, shaft
Y	Troughing transition
Z	Tension member (belt core)
a	Mass inertia on starting
b	Mass inertia on stopping
c	Sequential calculation
eff	Effective
f	Shape
g	Slide, base
h	Horizontal
k	Cubic, concave
l	Breakaway, longitudinal
leer	No-load operation
m	Mass
max	Maximum
min	Minimum
n	Number
nom	Nominal
o	Top run
p	Piece
r	Friction
red	Reduction
s	Scooping
sch	Chute
t	Short-term, temporary
th	Theoretical
u	Return run
v	Tensioning, convex
w	Abrasion
x	Point, location
z	Supplement, allowance

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Industry

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