1 Technical Information

project and design criteria
for belt conveyors

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Summary

1.1 Introduction

During the project design stage for the transport of raw materials or finished products, the choice of the method must favour the most cost effective solution for the volume of material moved, the plant and its maintenance, its flexibility for adaptation and its ability to carry a variety of loads and even be overloaded at times.

The belt conveyor, increasingly used in the last 10 years, is a method of conveying that satisfies the above selection criteria. Compared with other systems it is in fact the most economic, especially when one considers its adaptability to the most diverse and the most difficult conditions.

Today, we are not concerned only with horizontal or inclined conveyors but also with curves, conveyors in descent and with speeds of increasing magnitude.

However, the consideration in this section is not meant to be presented as the "bible" on project design for belt conveyors.

We wish to provide you with certain criteria to guide you in the choice of the most important components and calculations to help with correct sizing.

The technical information contained in the following sections is intended to basically support the designer and be integrated into the technical fulfillment of the project.

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1.2 Technical symbols

a	pitch of troughing sets	m
A	length of roller spindle	mm
a_g	distance between the pulley flange and support	mm
a_i	pitch of impact sets	m
a_o	pitch of carrying sets	m
a_t	pitch of transition sets	m
a_u	pitch of return sets	m
B	length of roller shell	mm
C	distance between roller supports	mm
Ca	static load on the carrying set	daN
ca	load on central roller of the carrying set	daN
Ca_1	dynamic load on the carrying set	daN
cd	dynamic load on the bearing	daN
Cf	constant of elasticity of the frame/impact roller	Kg/m
ch	flats of roller shaft	mm
C_o	static load on bearing	daN
Cp	resulting load of associated forces on motorised drum shaft	daN
Cp_r	resulting load of associated forces on idler drum shaft	daN
Cq	coefficient of fixed resistance	—
Cr	static load on the return set	daN
c_r	load on the roller of return set	daN
Cr_1	dynamic load on the return set	daN
Ct	coefficient of passive resistance given by temperature	—
Cw	wrap factor	—
d	diameter of spindle/shaft	mm
D	diameter of roller/pulley	mm
E	modules of elasticity of steel	daN/mm^2
e	logarithmic natural base	2,718
f	coefficient of internal friction of material and of rotating parts	—
f_a	coefficient of friction between the belt and drum given an angle of wrap	—
f_r	deflection of belt between two consecutive troughing sets	m
ft	deflection of a symmetrical shaft	mm
Fa	tangential force to move the belt in the direction of movement	daN
Fd	factor of impact	—
Fm	environmental factor	—
Fp	contribution factor	—
Fp_r	contribution factor on the central roller of a troughing set	—
Fr	tangential force to move the belt in the return direction	daN
Fs	service factor	—
Fu	total tangential force	daN
Fv	speed factor	—
G	distance between support brackets	mm
G_m	weight of lump of material	Kg
H	height change of belt	m
Hc	corrected height of fall	m
Hf	height of fall of material belt-screen	m
Ht	height change between motorised drum and counterweight	m
Hv	height of fall of material screen - receiving belt	m
I_C	distance from centre of motorised drum to the centre of the counterweight connection	m
I_M	load volume	m^3/h
I_V	belt load (material flow)	t/h

I_{VM}	load volume corrected to 1 m/s in relation to the inclination and irregularity of the feed	m^3/h
I_{VT}	load volume theoretic to 1 m/s	m^3/h
J	moment of inertia of section of material	mm^4
K	inclination factor	—
K_1	correction factor	—
σ_{amm}	admissible stress	daN/mm^2
L	load centres	m
Lb	dimensions of material lump	m
Lt	transition distance	m
Mf	bending moment	daNm
Mif	ideal bending moment	daNm
Mt	torsion moment	daNm
N	belt width	mm
n	revolutions per minute	rpm
P	absorbed power	kW
p_d	dynamic falling force	Kg
p_i	impact force of falling material	Kg
p_{ic}	force impact on central roller	Kg
Ppri	weight of lower rotating parts	Kg
Pprs	weight of upper rotating parts	Kg
q_b	weight of belt per linear metre	Kg/m
q_{bn}	weight of belt density	Kg/m^2
q_G	weight of material per linear metre	Kg/m
q_{RO}	weight of the upper rotating parts referred to the troughing set pitch	Kg/m
q_{RU}	weight of the lower rotating parts referred to the troughing set pitch	Kg/m
qs	specific weight	t/m^3
q_T	weight of drum	daN
RL	length of motorised drum face	mm
S	section of belt material	m^2
T_0	minimum tension at end of load zone	daN
T_1	tension on input side	daN
T_2	tension on output side	daN
T_3	tension on idler drum	daN
Tg	tension on belt at the point of counterweight connection	daN
T_{max}	tension at point of highest belt stress	daN
Tu_{max}	unitary maximum tension of belt	daN/mm
Tx	tension of the belt at a considered point	daN
Ty	tension of the belt at a considered point	daN
v	belt speed	m/s
V	maximum rise of edge of belt	mm
W	module of resistance	mm^3
α	angle of wrap of belt on pulley	degree
α_t	inclination of rotating symmetrical shaft	rad
β	angle of overload	degree
γ	angle of screen inclination	degree
δ	inclination of conveyor	degree
λ	inclination of side roller of troughing set	degree
λ_1	inclination of intermediate side roller	degree
λ_2	inclination of external side roller	degree
η	efficiency	—
y	angle deflection of bearing	degree

The symbol for kilogram (Kg) is intended as a unit of force.

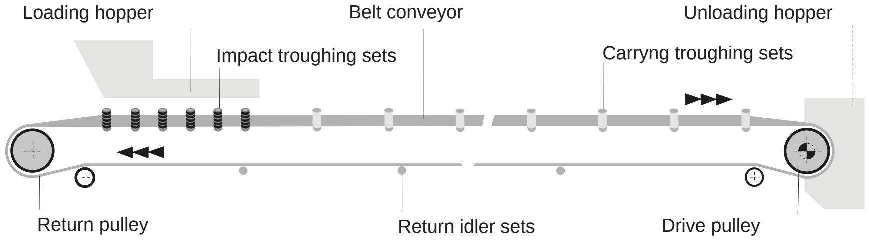


Fig.1 - Basic drawing of a belt conveyor

1.3 Technical characteristics of belt conveyors

The function of a belt conveyor is to continuously transport bulk materials of a mixed or homogeneous sort, a variable distance of some metres to tens of kilometres. One of the principal components of the conveyor is the elastomer belt which has a double function:
- to contain the conveyed material
- to transmit the force necessary to move the load.

The belt conveyor is designed to transport material in a continuous movement on the upper part of the belt.

The belt surfaces, upper on the carrying strand and lower on the return strand touch a series of rollers which are mounted from the conveyor structure itself in a group known as a troughing set. At either end of the conveyor the belt wraps around a pulley, one of which is coupled to a drive unit to transmit the motion.

The most competitive of other transport systems is certainly that of using lorries, With respect to the latter, the belt conveyor presents the following advantages:
- reduction in numbers of personnel
- reduction in energy consumption
- long periods between maintenance
- independence of the system to its surrounds
- reduced business costs

Based on the load, large belt conveyors are able to show cost add savings of up pass to 40-60% with respect to truck or lorry transport.

The electrical and mechanical components of the conveyor such as rollers, drums bearings, motors etc... are produced according to the highest standards. The quality level reached by major manufacturers guarantees function and long life.

The principal components of the conveyor, rollers and belt, need very little maintenance providing the design and the installation has been correctly performed. The elastomer belt needs only occasional or superficial repair and as the rollers are sealed for life they need no lubrication. The high quality and advanced technology of Rulmeca may reduce even further, or substitute, the need for ordinary maintenance.
Drum lagging has a life of at least two years.
The utilisation of adequate accessories to clean the belt at the feed and discharge points yields corresponding improvements to increase the life of the installation with minor maintenance.

All these factors combine to limit operational costs, especially where excavation work occurs, or underpasses below hills, roads or other obstacles. A smooth belt conveyor may travel up slopes up to 18° and there is always the possibility to recover energy on down hill sections. Projects have therefore been realised where conveyor system lengths may be up to 100 Km long with single sections of conveyor of 15 Km.

Utilising the characteristics of flexibility, strength and economy of purpose the belt conveyor is the practical solution to conveying bulk and other materials. Continuous developments is this field add to these existing advantages.

The following drawings show typical belt conveyor arrangements.

Fig.2.1- Conveyor with horizontal belt.

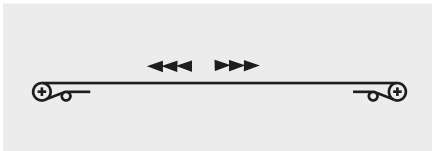


Fig.2.2 - Conveyor with horizontal belt with incline section, where the space permits a vertical curve and where the load requires the use of a single belt.

Fig.2.3 - Conveyor with incline belt and following horizontal section, when the load requires the use of a single belt and where space permits a vertical curve.

Fig.2.4 - Conveyor with horizontal and incline section where space does not allow a vertical curve and the load needs two belts to be employed.

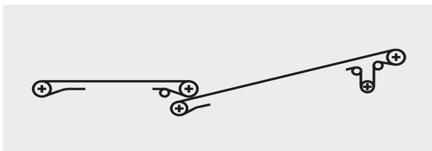


Fig.2.5- Conveyor belt with incline and horizontal where two belts are needed.

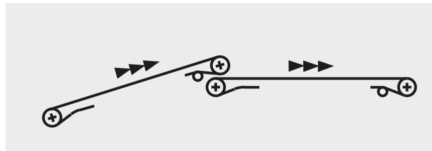


Fig.2.6 - Conveyor with horizontal and incline section where the space does not allow the vertical curve but the load may need the use of a single belt.

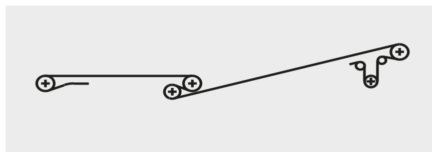


Fig.2.7 - Conveyor with a single belt comprising a horizontal section, an incline section and a decline section with vertical curves.

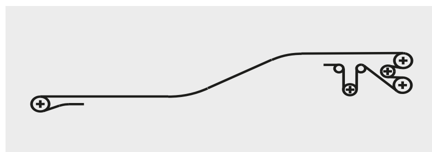
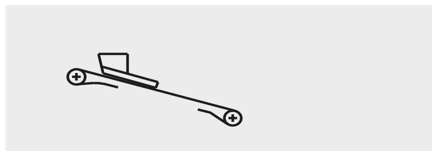


Fig.2.8 - Conveyor with belt loaded in decline or incline.



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1.4 Rulmeca key components for belt conveyors

Fig. **3** illustrates the basic components of a typical belt conveyor. In practice, according to the variety of uses, it is possible to have many other diverse combinations of load and unload areas, elevations, and other accessories.

Drive head
May be of traditional design or with motorised drum unit.
- Traditional
Comprises a drive group consisting of a drive drum of a diameter appropriately sized to the load on the belt, and an idler drum at the opposing end.
The power is supplied by a direct coupled motor gearbox or by a direct or parallel shaft drive driving the drive drum through a suitably sized couple.

- Motorised Pulleys
In this arrangement the motor, gearbox and bearings form a complete designed unit
inside and protected by the drum shell which directly powers the belt. This eliminates all the external complication of external drive, couples etc. as described above in the
traditional design. Today motorised pulleys are produced in diameters up to 1000 mm with a maximum power of 250 kW and with a drive efficiency which may reach 97%.

Drive pulley
The shell face of the conventional drive pulley or the motorised drum may be left as normal finish or clad in rubber of a thickness calculated knowing the power to be transmitted.

The cladding may be grooved as herringbone design, or horizontal grooves to the direction of travel, or diamond grooves; all designed to increase the coefficient of friction and to facilitate the release of water from the drum surface.

The drum diameter is dimensioned according to the class and type of belt and to the designed pressures on its surface.

Return pulleys
The shell face does not necessarily need to be clad except in certain cases, and the diameter is normally less than that designed for the drive pulley.

Deflection or snub pulleys
These are used to increase the angle of wrap of the belt and overall for all the necessary changes in belt direction in the areas of counterweight tensioner, mobile unloader etc..

Rollers
Support the belt and are guaranteed to rotate freely and easily under load. They are the most important components of the conveyor and represent a considerable value of the whole cost. The correct sizing of the roller is fundamental to the guarantee of the plant efficiency and economy in use.

Upper carrying troughing and return sets
The carrying rollers are in general positioned in brackets welded to a cross member or frame. The angle of the side roller varies from 20° to 45°. It is also possible to arrive at angles of up to 60° using the "garland" suspension design.
The return roller set may be designed incorporating one single width roller or two rollers operating in a "V" formation at angles of 10°.

Depending on various types of material being conveyed the upper carrying sets may be designed symmetrically or not, to suit.

Tension units
The force necessary to maintain the belt contact to the drive pulley is provided by a tension unit which may be a screw type unit, a counterweight or a motorised winch unit. The counterweight provides a constant tensional force to the belt independent of the conditions. Its weight designed according to the minimum limits necessary to guarantee the belt pull and to avoid unnecessary belt stretch.

The designed movement of the counterweight tension unit is derived from the elasticity of the belt during its various phases of operation as a conveyor.

The minimum movement of a tension unit must not be less than 2% of the distance between the centres of the conveyor using textile woven belts, or 0.5% of the conveyor using steel corded belts.

Hopper
The hopper is designed to allow easy loading and sliding of the material in a way to absorb the shocks of the load and avoids blockage and damage to the belt. It caters for instantaneous charging of load and its eventual accumulation.

The hopper slide should relate to the way the material falls and its trajectory and is designed according to the speed of the conveyor. Lump size and the specific gravity of the charge and its physical properties such as humidity, corrosiveness etc. are all very relevant to the design.

Cleaning devices
The system of cleaning the belt today must be considered with particular attention to reduce the need for frequent maintenance especially when the belt is conveying wet or sticky materials. Efficient cleaning allows the conveyor to obtain maximum productivity.

There are many types and designs of belt cleaners. The most straight forward simple design is that of a straight scraper blade mounted on rubber supports (chapter 5).

Conveyor covers
Covers over the conveyor are of fundamental importance when it is necessary to protect the conveyed material from the atmosphere and to guarantee efficient plant function (chapter 6).

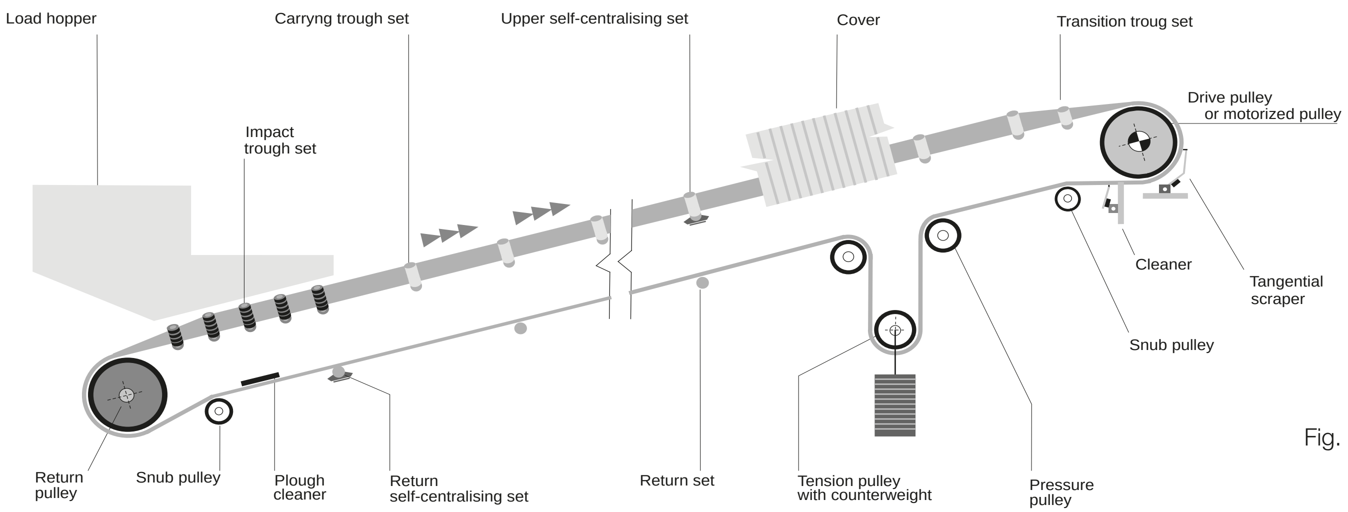

Fig. **3**

1.5 - Project criteria

The choice of the optimum conveyor system and its project design and rationalisation depends on full knowledge of the construction characteristics and the forces involved that apply themselves to all the system components.

The principal factors that influence the sizing of a belt conveyor are: the required load volume, the type of transported material and its characteristics such as grain or lump size, and chemical / physical properties. The route and height profile of the conveyor is also relevant.
In the following illustrations you may follow the criteria used for the calculation of the belt speed and width, the type and arrangement of troughing sets, the type of rollers to be used and finally the determination of the drum sizes.

1.5.1 - Conveyed material

The correct project design of the belt conveyor must begin with an evaluation of the characteristics of the conveyed material and in particular the angle of repose and the angle of surcharge.

The angle of repose of a material, also known as the “angle of natural friction” is the angle at which the material, when heaped freely onto a horizontal surface takes up to the horizontal plane. Fig. **4**.

The angle of surcharge is the angle measured with respect to the horizontal plane, of the surface of the material being conveyed by a moving belt. Fig. **5**.
This angle is normally between 5° and 15° (for a few materials up to 20°) and is much less than the angle of repose.

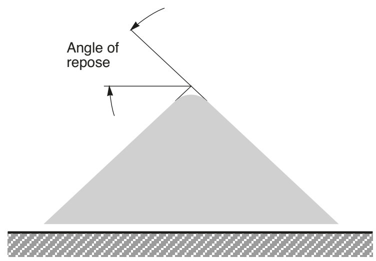

Fig.**4**

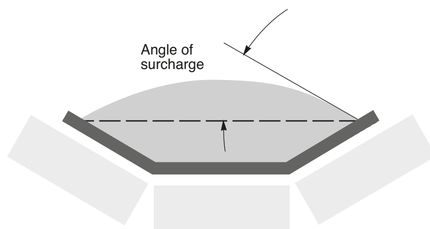

Fig.**5**

Tab.1 shows the correlation between the physical characteristics of materials and their relative angles of repose.

The conveyed material settles into a configuration as shown in sectional diagram Fig. **6**.
The area of the section "S" may be calculated geometrically adding the area of a circle A_1 to that of the trapezoid A_2.

$$S = A_1 + A_2$$

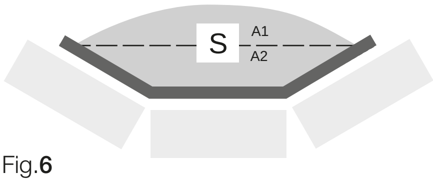

Fig.**6**

The value of the conveyed volume I_{VT} may be easily calculated using the formula:

$$S = \frac{I_{VT}}{3600} \quad [\,m^2\,]$$

where:

I_{VT} = conveyed volume at a conveyor speed of 1 m/s (see Tab.5a-b-c-d)

Tab. **1 - Angles of surcharge, repose and material fluency**

Fluency	very high	high	medium		low	Profile on a flat belt
Angle of surcharge β	**5°**	**10°**	**20°**	**25°**	**30°**	**ß**
						
Angle of repose	**0-19°**	**20-29°**	**30-34°**	**35-39°**	**40° and more**	**Others**
Characteristics of materials	Uniform dimensions, round particles, very small size. Very humid or very dry such as dry sand, silica, cement and wet limestone dust etc.	Partly rounded particles, dry and smooth. Average weight as for example cereal, grain and beans.	Irregular material, granular particles of average weight as for example anthracite coal, clay etc.	General everyday material as for example bitumi-nous coal and the majority of minerals.	Irregular viscous fibrous material which tends to get worse in handling, as for example wood shavings, sugar cane by product, foundry sand, etc.	Here may be included materials with a variety of characteristics as indicated in the following Tab.2.

*Tab.*2 - **Physical properties of materials**

Type	Average specific weight qs		Angle	Abrasive -	Corrosive -
	t/m³	lbs. / Cu.Ft	of repose	ness	ness
Alumina	0,80-1,04	50-65	22°	C	A
Aluminium chips	0,11-0,24	7-15	-	B	A
Aluminium oxide	1,12-1,92	70-120	-	C	A
Aluminium sulphate (granular)	0,864	54	32°	-	-
Ammonium nitrate	0,72	45	-	B	C
Ammonium sulphate	0,72-0,93	45-58	32°	B	C
Asbestos ore or rock	1,296	81	-	C	A
Ashes, coal, dry, up to 80 mm	0,56-0,64	35-40	40°	B	A
Ashes, coal, wet, up to 80 mm	0,72-0,80	45-50	50°	B	P
Asphalt, binder for paving	1,28-136	80-85	-	A	B
Asphalt, crushed up to13 mm	0,72	45	-	A	A
Bakelite, fine	0,48-0,64	30-40	-	A	A
Barite	2,88	180	-	A	A
Barium carbonate	1,152	72	-	A	A
Bauxite, mine run	1,28-1,44	80-90	31°	C	A
Bauxite, ground, dried	1,09	68	35°	C	A
Bentonite, up to 100 mesh	0,80-0,96	50-60	-	B	A
Borax, lump	0,96-1,04	60-65	-	B	A
Brick, hard	2	125	-	C	A
Calcium carbide	1,12-1,28	70-80	-	B	B
Carbon black pellets	0,32-0,40	20-25	-	A	A
Carbon black powder	0,06-0,11	4-7	-	A	A
Carborundum, up to 80 mm	1,60	100	-	C	A
Cast iron chips	2,08-3,20	130-200	-	B	A
Cement, rock (see limestone)	1,60-1,76	100-110	-	B	A
Cement, Portland, aerated	0,96-1,20	60-75	39°	B	A
Charcoal	0,29-0,40	18-25	35°	A	A
Chrome ore (cromite)	2-2,24	125-140	-	C	A
Clay, dry, fine	1,60-1,92	100-120	35°	C	A
Clay, dry, lumpy	0,96-1,20	60-75	35°	C	A
Clinker	1,20-1,52	75-95	30-40°	C	A
Coal, anthracite	0,96	60	27°	B	A
Coal, bituminous, 50 mesh	0,80-0,86	50-54	45°	A	B
Coal, bituminous, run of mine	0,72-0,88	45-55	38°	A	B
Coal, lignite	0,64-0,72	40-45	38°	A	B
Coke breeze, 6 mm	0,40-0,5	25-35	30-45°	C	B
Coke, loose	0,37-0,56	23-35	-	C	B
Coke petroleum calcined	0,56-0,72	35-45	-	A	A
Concrete, in place, stone	2,08-2,40	130-150	-	C	A
Concrete, cinder	1,44-1,76	90-110	-	C	A
Copper, ore	1,92-2,40	120-150	-	-	-
Copper sulphate	1,20-1,36	75-85	31°	A	-
Cork	0,19-0,24	12-15	-	-	-
Cryolite	1,76	110	-	A	A
Cryolite, dust	1,20-1,44	75-90	-	A	
A					
Diacalcium phosphate	0,688	43	-	-	-
Disodium phosphate	0,40-0,50	25-31	-		
Dolomite, lumpy	1,44-1,60	90-100	-	B	A

Table 2 states physical and chemical properties of materials that you have to take into consideration for the belt conveyor project.

*Tab.*2 - Physical properties of materials

Type	Average specific weight qs t/m³	lbs. / Cu.Ft	Angle of repose	Abrasive - ness	Corrosive - ness
Earth, wet, containing clay	1,60-1,76	100-110	45°	B	A
Feldspar, 13 mm screenings	1,12-1,36	70-85	38°	C	A
Feldspar, 40 mm to 80 mm lumps	1,44-1,76	90-110	34°	C	A
Ferrous sulphate	0,80-1,20	50-75	-	B	-
Foundry refuse	1,12-1,60	70-100	-	C	A
Gypsum, 13 mm to 80 mm lumps	1,12-1,28	70-80	30°	A	A
Gypsum, dust	0,96-1,12	60-70	42°	A	A
Graphite, flake	0,64	40	-	A	A
Granite,13 mm screening	1,28-1,44	80-90	-	C	A
Granite, 40 mm to 50 mm lumps	1,36-1,44	85-90	-	C	A
Gravel	1,44-1,60	90-100	40°	B	A
Gres	1,36-1,44	85-90	-	A	A
Guano, dry	1,12	70	-	B	-
Iron ore	1,60-3,20	100-200	35°	C	A
Iron ore, crushed	2,16-2,40	135-150	-	C	A
Kaolin clay, up to 80 mm	1,008	63	35°	A	A
Kaolin talc, 100 mesh	0,67-0,90	42-56	45°	A	A
Lead ores	3,20-4,32	200-270	30°	B	B
Lead oxides	0.96-2,04	60-150	-	A	-
Lime ground, up to 3 mm	0,96	60	43°	A	A
Lime hydrated, up to 3 mm	0,64	40	40°	A	A
Lime hydrated, pulverized	0,51-0,64	32-40	42°	A	A
Limestone, crushed	1,36-1,44	85-90	35°	B	A
Limestone, dust	1,28-1,36	80-85	-	B	A
Magnesite (fines)	1,04-1,20	65-75	35°	B	A
Magnesium chloride	0,528	33	-	B	-
Magnesium sulphates	1,12	70	--	-	
Manganese ore	2,00-2,24	125-140	39°	B	A
Manganese sulphate	1,12	70	-	C	A
Marble, crushed, up to 13 mm	1,44-1,52	90-95	-	B	A
Nickel ore	2,40	150	-	C	B
Phosphate, acid, fertilizer	0,96	60	26°	B	B
Phosphate, florida	1,488	93	27°	B	A
Phosphate rock, pulverized	0,96	60	40°	B	A
Phosphate, super ground	0,816	51	45°	B	B
Pyrite-iron, 50 to 80 mm lumps	2,16-2,32	135-145	-	B	B
Pyrite, pellets	1,92-2,08	120-130	-	B	B
Polystyrene beads	0,64	40	-	-	-
Potash salts, sylvite, etc.	1,28	80	-	A	B
Potassium cloride, pellets	1,92-2,08	120-130	-	B	B
Potassium nitrate (saltpeter)	1,216	76	-	B	B
Potassium sulphate	0,67-0,77	42-48	-	B	-

A non abrasive/non corrosive
B mildly abrasive/ mildly corrosive
C very abrasive/very corrosive

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Tab. 2 - Physical properties of materials

Type	Average specific weight qs		Angle	Abrasive -	Corrosive -
	t/m³	lbs. / Cu.Ft	of repose	ness	ness
Quartz 40 mm to 80 mm lumps	1,36-1,52	85-95	-	C	A
Quartz, dust	1,12-1,28	70-80	-	C	A
Quartz, 13 mm screening	1,28-1,44	80-90	-	C	A
Rubber, pelletized	0,80-0,88	50-55	35°	A	A
Rubber, reclaim	0,40-0,48	25-30	32°	A	A
Salt, common dry, coarse	0,64-0,88	40-55	-	B	B
Salt, common dry, fine	1,12-1,28	70-80	25°	B	B
Sand, damp	1,76-2,08	110-130	45°	C	A
Sand, dry	1,44-1,76	90-110	35°	C	A
Sand, foundry, shakeout	1,44-1,60	90-100	39°	C	A
Slag, blast furnace, crushed	1,28-1,44	80-90	25°	C	A
Slate, 40 mm to 80 mm lumps	1,36-1,52	85-95	-	B	A
Slate, dust	1,12-1,28	70-80	35°	B	A
Soap powder	0,32-0,40	20-25	-	A	A
Soapstone, talc, fine	0,64-0,80	40-50	-	A	A
Soda heavy asmes	0,88-1,04	55-65	32°	B	C
Sodium bicarbonate	0,656	41	42°	A	A
Sodium nitrate	1,12-1,28	70-80	24°	A	-
Steel shavings	1,60-2,40	100-150	-	C	A
Sugar beet, pulp (dry)	0,19-0,24	12-15	-	-	-
Sugar beet, pulp (wet)	0,40-0,72	25-45	-	A	B
Sugar, cane, knifed	0,24-0,29	15-18	50°	B	A
Sugar, powdered	0,80-0,96	50-60	-	A	B
Sugar, raw, cane	0,88-1,04	55-65	30°	B	B
Sugar, wet, beet	0,88-1,04	55-65	30°	B	B
Sulphur, crushed under 13 mm	0,80-0,96	50-60	-	A	C
Sulphur, up to 80 mm	1,28-1,36	80-85	-	A	C
Talc, powdered	0,80-0,96	50-60	-	A	A
Talc, 40 mm to 80 mm lumps	1,36-1,52	85-95	-	A	A
Titanium dioxide	0,40	25	-	B	A
Wheat	0,64-0,67	40-42	25°	A	A
Wood chips	0,16-0,48	10-30	-	A	A
Zinc concentrates	1,20-1,28	75-80	-	B	A
Zinc ore, roasted	1,60	100	38°	-	-
Zinc oxide, heavy	0,48-0,56	30-35	-	A	A

A non abrasive/non corrosive
B mildly abrasive/mildly corrosive
C very abrasive/very corrosive

1.5.2 - Belt speed

The maximum speed of a belt conveyor in this field has reached limits not thought possible some years ago.
Very high speeds have meant a large increase in the volumes conveyed. Compared with the load in total there is a reduction in the weight of conveyed material per linear metre of conveyor and therefore there is a reduction in the costs of the structure in the troughing set frames and in the belt itself.
The physical characteristics of the conveyed material is the determining factor in calculating the belt speed.
Light material, that of cereal, or mineral dust or fines, allow high speeds to be employed. Screened or sifted material may allow belt speeds of over 8 m/s.
With the increase of material lump size, or its abrasiveness, or that of its specific weight, it is necessary to reduce the conveyor belt speed.
It may be necessary to reduce conveyor speeds to a range in the order of 1.5/3.5 m/s to handle unbroken and unscreened rock of large lump size.
The quantity of material per linear metre loaded on the conveyor is given by the formula:

$$q_G = \frac{I_V}{3.6 \times v} \quad [\text{ Kg/m }]$$

where:

q_G = weight of material per linear metre

I_V = belt load t/h

v = belt speed m/s

q_G is used in determining the tangential force Fu.

With the increase of speed v it is possible to calculate the average belt load I_V with a narrower belt width (and therefore it follows a simpler conveyor structure) as well as a lower load per linear metre and therefore a reduction results in the design of rollers and troughing sets and in less belt tension.

Nevertheless larger belt widths, relative to the belt load, are used at high and low speeds where there is less danger of losing material, fewer breakdowns and less blockage in the hoppers.

From experimental data we show in Tab. **3** the maximum belt speeds advised considering the physical characteristics and lump size of the conveyed material and the width of the belt in use.

Tab. **3 - Maximum speeds advised**

Lump size max. dimensions uniform up to mm	mixed up to mm	**Belt** min. width mm	max. speed **A**	**B**	**C**	**D**
50	100	400	2.5	2.3	2	1.65
75	150	500				
125	200	650	3	2.75	2.38	2
170	300	800	3.5	3.2	2.75	2.35
250	400	1000	4	3.65	3.15	2.65
350	500	1200				
400	600	1400	4.5	4	3.5	3
450	650	1600				
500	700	1800	5	4.5	3.5	3
550	750	2000				
600	800	2200	6	5	4.5	4

A - Light sliding material non abrasive, specific weight from 0.5 ÷ 1,0 t/m^3
B - Material non abrasive, medium size, specific weight from 1.0 ÷ 1.5 t/m^3
C - Material moderately abrasive and heavy with specific weight from 1.5 ÷ 2 t/m^3
D - Abrasive material, heavy and sharp over 2 t/m^3 specific weight

Considering the factors that limit the maximum conveyor speed we may conclude:

When one considers the inclination of the belt leaving the load point: the greater the inclination, the increase in the amount of turbulence as the material rotates on the belt. This phenomena is a limiting factor in calculating the maximum belt speed in that its effect is to prematurely wear out the belt surface.

The repeated action of abrasion on the belt material, given by numerous loadings onto a particular section of the belt under the load hopper, is directly proportional to the belt speed and inversely proportional to its length.

1.5.3 - Belt width

Given, using Tab.**3**, the optimum belt speed, the determination of the belt width is largely a function of the quantity of conveyed material which is indicated by the project data.

In the following section, the conveyor capacity may be expressed as loaded volume I_{VT} [m^3/h] per v= 1 m/sec.
The inclination of the side rollers of a transom (from 20° to 45°) defines the angle of the troughing set Fig.**7**.

Troughing sets at 40°/45° are used in special cases, where because of this onerous position the belts must be able to adapt to such an accentuated trough.

In practice the choice and design of a troughing set is that which meets the required loaded volume, using a belt of minimum width and therefore the most economic.

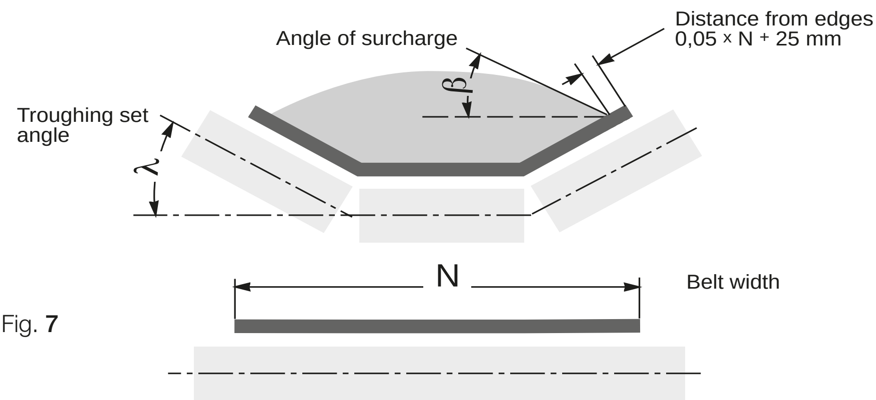


Fig. **7**

All things being equal the width of the belt at the greatest angle corresponds to an increase in the loaded volume I_{VT}.

The design of the loaded troughing set is decided also as a function of the capacity of the belt acting as a trough.

In the past the inclination of the side rollers of a troughing set has been 20°. Today the improvements in the structure and materials in the manufacture of conveyor belts allows the use of troughing sets with side rollers inclined at 30°/35°.

It may be observed however that the belt width must be sufficient to accept and contain the loading of material onto the belt whether it is of mixed large lump size or fine material.

In the calculation of belt dimensions one must take into account the minimum values of belt width as a function of the belt breaking load and the side roller inclination as shown in *Tab.***4** .

Tab. **4 - Minimum belt width**

in relation to belt breaking load and roller inclinations.

Breaking load	Belt width		
N/mm	λ= 20/25° mm	λ= 30/35°	λ= 45°
250	400		
315	400	400	450
400	400	400	450
500	450	450	500
630	500	500	600
800	500	600	650
1000	600	650	800
1250	600	800	1000
1600	600	800	1000

For belts with higher breaking loads than those indicated in the table, it is advisable to consult the actual belt manufacturer.

Loaded volume I_M

The volumetric load on the belt is given by the formula:

$$I_M = \frac{I_V}{qs} \quad [\, m^3/h \,]$$

where:

I_V = load capacity of the belt [t/h]

qs = specific weight of the material

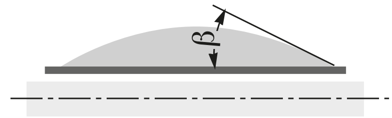
Also defined as:

$$I_{VT} = \frac{I_M}{v} \quad [\, m^3/h \,]$$

where the loaded volume is expressed relevant to the speed of 1 m/s.

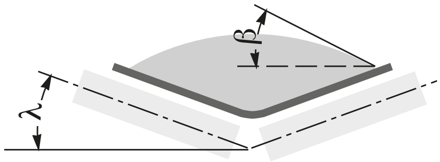
It may be determined from *Tab.* **5**a-b-c-d, that the chosen belt width satisfies the required loaded volume I_M as calculated from the project data, in relation to the design of the troughing sets, the roller inclination, the angle of material surcharge and to belt speed.

Tab. **5a - Loaded volume**
with flat roller sets **v = 1 m/s**



Belt width mm	Angle of surcharge β	I_{VT} m³/h λ = 0°	Belt width mm	Angle of surcharge β	I_{VT} m³/h λ = 0°
	5°	2.5		5°	101.3
	10°	5.1		10°	203.2
300	20°	10.3	1600	20°	411.3
	25°	13.0		25°	519.0
	30°	15.8		30°	630.1
	5°	5.0		5°	129.2
	10°	10.1		10°	259.2
400	20°	20.5	1800	20°	524.8
	25°	25.8		25°	662.1
	30°	31.3		30°	803.8
	5°	8.4		5°	160.5
	10°	16.8		10°	322.0
500	20°	34.1	2000	20°	652.0
	25°	43.0		25°	822.7
	30°	52.2		30°	998.7
	5°	15.0		5°	199.3
	10°	30.1		10°	399.8
650	20°	60.9	2200	20°	809.6
	25°	76.9		25°	1021.5
	30°	93.3		30°	1240.0
	5°	23.5		5°	242.3
	10°	47.2		10°	486.0
800	20°	95.6	2400	20°	984.1
	25°	120.6		25°	1241.7
	30°	146.4		30°	1507.4
	5°	37.9		5°	289.5
	10°	76.0		10°	580.7
1000	20°	153.8	2600	20°	1175.8
	25°	194.1		25°	1483.5
	30°	235.6		30°	1800.9
	5°	55.6		5°	340.8
	10°	111.6		10°	683.7
1200	20°	225.9	2800	20°	1384.4
	25°	285.0		25°	1746.8
	30°	346.0		30°	2120.5
	5°	76.7		5°	396.4
	10°	153.9		10°	795.2
1400	20°	311.7	3000	20°	1610.1
	25°	393.3		25°	2031.5
	30°	477.5		30°	2466.2

Tab. **5**b **- Loaded volume**
with 2 roll troughing sets **v = 1 m/s**



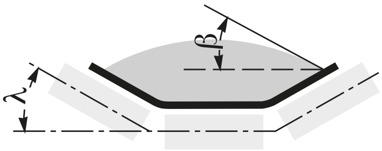
Belt width mm	Angle of surcharge β	I_{VT} m³/h λ = 20°
300	5°	16,2
	10°	18,5
	20°	23,1
	25°	25,5
	30°	27,9
400	5°	32,2
	10°	36,7
	20°	45,9
	25°	50,6
	30°	55,5
500	5°	53,7
	10°	61,1
	20°	76,4
	25°	84,2
	30°	92,4
650	5°	96,0
	10°	109,4
	20°	136,6
	25°	150,7
	30°	165,2
800	5°	150,6
	10°	171,5
	20°	214,2
	25°	236,3
	30°	259,1
1000	5°	242,4
	10°	276,1
	20°	344,8
	25°	380,4
	30°	417,0

To obtain the effective loaded volume I_M at the desired belt speed use:

$$I_M = I_{VT} \times V \quad [\,m^3/h\,]$$

Tab. **5c - Loaded volume**
with 3 roll troughing sets **v = 1 m/s**

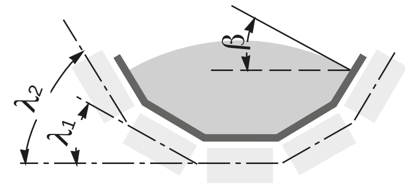
Belt width mm	Angle of surcharge β	I_{VT} m³/h λ = 20°	λ = 25°	λ = 30°	λ = 35°	λ = 45°
300	5°	12.5	14.7	16.7	18.4	21.3
	10°	14.9	17	18.9	20.6	23.3
	20°	19.8	21.8	23.5	25	27.2
	25°	22.4	24.3	25.9	27.3	29.3
	30°	25	26.8	28.4	29.7	31.4
400	5°	25.3	29.7	33.8	37.4	43.2
	10°	30.1	34	38	41.7	47
	20°	39.9	43.8	47.4	50.4	54.8
	25°	44.9	48.7	52	54.9	58.8
	30°	50.2	53.8	56.9	59.5	62.9
500	5°	43.2	50.7	57.7	63.8	73.6
	10°	51.1	58.4	65	70.8	79.8
	20°	67.4	74	80.1	85.2	92.6
	25°	75.8	82.3	87.9	92.7	99.2
	30°	84.4	90.7	96	100	106
650	5°	80.3	94.4	107.2	118.6	136.3
	10°	94.4	108	125	131	147.1
	20°	123	136	147	156.3	169.3
	25°	138	150	160	169	180
	30°	153	165	175	182	192.7
800	5°	125.9	148.1	168.2	186	213.8
	10°	148.1	169.5	188.7	205.4	230.8
	20°	193.5	213.3	230	245.1	265.6
	25°	217	235.9	252.2	265.7	283.6
	30°	241.2	259.3	274.6	286.9	302.2
1000	5°	207.5	244.1	277.1	306.1	351
	10°	243.2	278.4	309.8	337.1	377.9
	20°	316	348.5	376.7	400.4	433
	25°	353.7	384.8	411.4	433.1	461.4
	30°	392.5	422.2	447	466.9	490.8
1200	5°	304	357.5	405.9	448	514.3
	10°	356.3	407.9	454	494	554
	20°	463.3	510.9	552.3	587	634.9
	25°	518.6	564.2	603.2	635	676.8
	30°	575.7	619.2	655.7	684	720
1400	5°	424.9	499.7	547.1	626.3	717.2
	10°	497	569	633.3	688.8	771.3
	20°	644.4	710.8	768.4	816.5	881.9
	25°	720.6	784.1	838.8	882.5	939.1
	30°	799.2	859.8	910.4	950.6	998.1



Belt width mm	Angle of surcharge β	I_{VT} m³/h λ = 20°	λ = 25°	λ = 30°	λ = 35°	λ = 45°
1600	5°	564.1	663.4	752.8	831.2	951
	10°	659.2	754.8	839.9	913.4	1022.1
	20°	853.5	941.6	1017.9	1081.4	1167.3
	25°	954	1038.2	1110	1168.2	1242.4
	30°	1057.6	1137.9	1204.9	1257.9	1319.9
1800	5°	723	850.1	964.7	1064.9	1217.6
	10°	844.2	966.7	1075.6	1169.5	1307.9
	20°	1091.9	1204.7	1302.3	1383.3	1492.5
	25°	1220	1327.9	1419.6	1493.9	1587.9
	30°	1352.2	1454.9	1540.5	1608	1686.4
2000	5°	897.3	1055.2	1197.3	1321.7	1511.5
	10°	1047.9	1200	1335.2	1451.8	1623.8
	20°	1355.8	1495.8	1617	1717.6	1853.4
	25°	1515	1648.9	1762.7	1855.1	1972.1
	30°	1679.2	1806.7	1913	1996.9	2094.5
2200	5°	1130.8	1329.5	1508	1663.5	1898.1
	10°	1317.4	1508.7	1678.3	1823.8	2035.7
	20°	1698.7	1874.7	2026.2	2151.3	2317
	25°	1895.9	2064	2206.2	2320.7	2462.4
	30°	2099.3	2259.2	2391.8	2495.4	2612.4
2400	5°	1366.2	1606.4	1822.3	2010.9	2296.8
	10°	1599.2	1824.5	2029.8	2206.4	2465
	20°	2057.2	2270.1	2453.8	2605.9	2808.8
	25°	2297.2	2500.6	2673.1	2812.5	2986.6
	30°	2544.7	2738.3	2899.3	3029.5	3170
2600	5°	1650.6	1940.6	2200.6	2426.9	2767
	10°	1921.4	2200.4	2447.5	2659.1	2965.9
	20°	2474.7	2731.3	2951.9	3133.5	3372.4
	25°	2760.9	3005.8	3212.7	3378.9	3582.7
	30°	3056	3289	3481.8	3631.9	3799.5
2800	5°	1932.9	2272.7	2577.7	2843.6	3244.9
	10°	2252	2579	2868.9	3117.7	3480.3
	20°	2904.1	3205	3464.1	3678	3961.4
	25°	3241.4	3528.7	3771.9	3967.7	4210.3
	30°	3589.2	3862.6	4089.3	4266.6	4469.9
3000	5°	2256.1	2652.5	3008.1	3317.8	3783.9
	10°	2627	3008.4	3346.4	3636	4056.8
	20°	3384.9	3735.8	4037.6	4286.4	4614.5
	25°	3776.9	4111.9	4395.1	4627.7	4902.9
	30°	4181.3	4496.9	4763.8	4969.6	5200.3

To obtain the effective loaded volume I_M at the desired belt speed use:

$$I_M = I_{VT} \times V \quad [\,m^3/h\,]$$



Tab. **5d - Loaded volume**
with 5 roll troughing sets **v = 1 m/s**

Belt width mm	Angle of surcharge β	I_{VT} m³/h λ1 30° λ2 60°
800	5°	236.4
	10°	252.4
	20°	284.6
	25°	301.4
	30°	318.7
1000	5°	381.8
	10°	407.8
	20°	459
	25°	485.8
	30°	513.4
1200	5°	566.8
	10°	603.3
	20°	678.1
	25°	716.7
	30°	756.6
1400	5°	787.8
	10°	837.6
	20°	939.5
	25°	992.1
	30°	1046.4
1600	5°	1038.8
	10°	1104.6
	20°	1239.2
	25°	1308.8
	30°	1380.6
1800	5°	1324.4
	10°	1408.5
	20°	1580.4
	25°	1669.3
	30°	1761
2000	5°	1659
	10°	1762.6
	20°	1972.7
	25°	2081.3
	30°	2193.1
2200	5°	2058.2
	10°	2186.2
	20°	2447.7
	25°	2582.9
	30°	2722.4
2400	5°	2525.5
	10°	2678.1
	20°	2989.8
	25°	3151
	30°	3317.3
2600	5°	3030.5
	10°	3210.5
	20°	3579.4
	25°	3770.2
	30°	3966.9
2800	5°	3570.8
	10°	3782.9
	20°	4216.3
	25°	4440.5
	30°	4671.7
3000	5°	4165.6
	10°	4410.5
	20°	4910.9
	25°	5169.6
	30°	5436.6

To obtain the effective loaded volume I_M at desired belt speed use:

$$I_M = I_{VT} \times v \qquad [\, m^3/h \,]$$

Corrects loaded volume in relation to the factors of inclination and feed

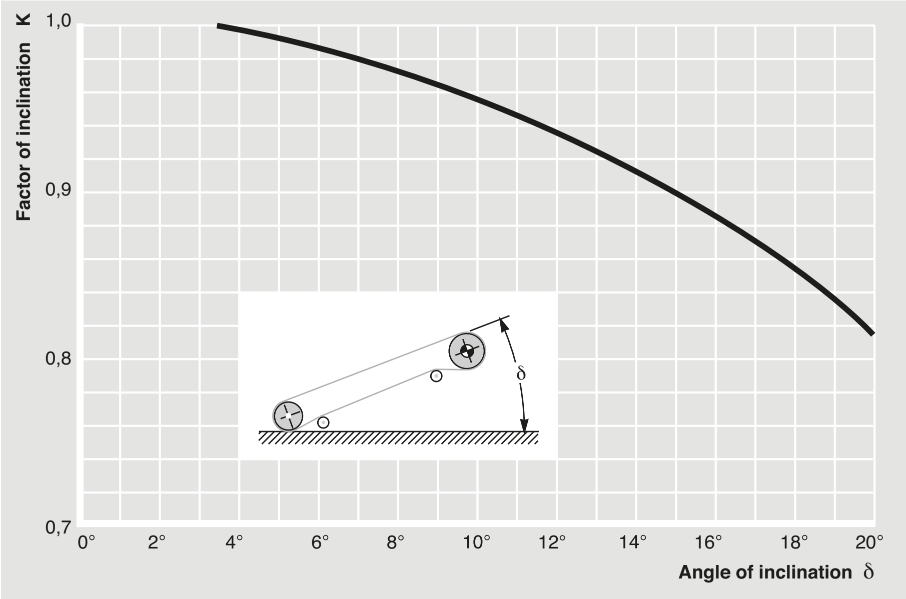
In the case of inclined belts, the values of loaded volume I_{VT} [m^3/h] are corrected according to the following:

$$I_{VM} = I_{VT} \times K \times K_1 \quad [m^3/h]$$

Where:

- I_{VM} is the loaded volume corrected in relation to the inclination and the irregularity of feeding the conveyor in m^3/h with $v = 1$ m/s
- I_{VT} is the theoretic load in volume for $v = 1$ m/s
- K is the factor of inclination
- K1 is the correction factor given by the feed irregularity

Fig.**8** - **Factor of inclination K**



The inclination factor K calculated in the design, must take into account the reduction in section for the conveyed material when it is on the incline.

Diagram Fig.**8** gives the factor K in function of the angle of conveyor inclination, but only for smooth belts that are flat with no profile.

In general it is necessary to take into account the nature of the feed to the conveyor, whether it is constant and regular, by introducing a correction factor K_1 its value being:

- $K_1 = 1$ regular feed
- $K_1 = 0.95$ irregular feed
- $K_1 = 0.90 \div 0.80$ most irregular feed.

If one considers that the load may be corrected by the above factors the effective loaded volume at the required speed is given by:

$$I_M = I_{VM} \times v \quad [m^3/h]$$

Given the belt width, one may verify the relationship between the belt width and the maximum lump size of material according to the following:

belt width ≥ max. lump size

1.5.4 - Type of troughing set, pitch and transition distance

Type

For each troughing set there is a combination of rollers positioned into a suitable fixed support frame Fig. **9**; the troughing sets may also be suspended as a “garland” Fig. **10**.

There are 2 basic types of troughing set base frame: the upper set, which carries the loaded belt on the upper strand, and the lower set, which supports the empty belt on the return strand.

• The upper carrying troughing set is generally designed as the following arrangement:
- one or two parallel rollers
- two, three or more rollers in a trough.

• The return set can be with:
- one or two flat rollers
- a trough of two rollers.

The roller frame with fixed supports, with three rollers of equal length, support the belt well with a uniform distribution of forces and load sharing.
The inclination of the side roller varies from 20° up to 45° for belts of 400 mm width up to 2200 mm and over.

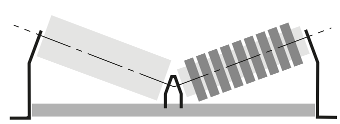
The suspended sets of “garland” design are used incorporating impact rollers to accept the impact under the load hopper, and also in use along the conveyor upper and lower strands where large loads may be carried or on very high performance conveyors.

The troughing sets are generally designed and manufactured according to international unified standards.

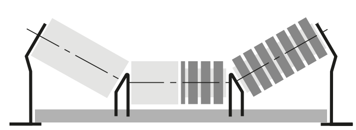
The drawings illustrate the more common arrangements.

Fig. **9** - Troughing sets upper strand

- parallel roller plain or impact



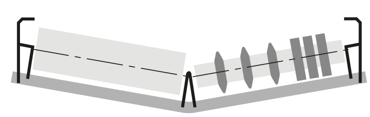
- 2 rollers plain or impact



- 3 rollers plain or impact

Return sets

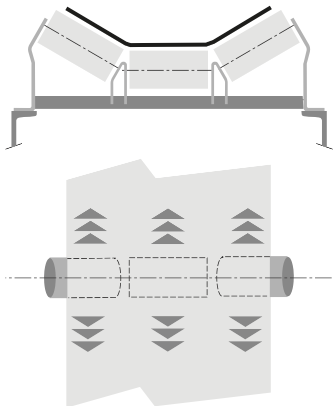
- roller plain or with rubber rings



- 2 rollers plain or with rings

The choice of the most appropriate and correct troughing set installation (one needs to calculate the frictional force between the rollers and the belt itself) is the guarantee for the smooth belt start up and movement.

The troughing sets on the upper strand of a reversible belt may have the rollers in line with each other and at right angles to the belt as in Fig. **11**; in the case of non-reversible belt the side rollers are inclined forward by 2° in the same sense of direction of the belt, as in Fig. **12**.



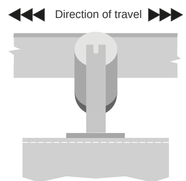
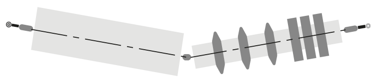
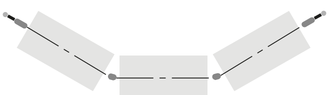


Fig. **11** - For reversible belts

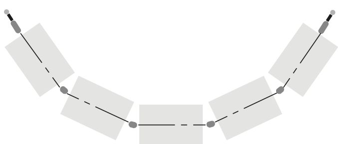
Fig. **10** - Suspension sets "garland"



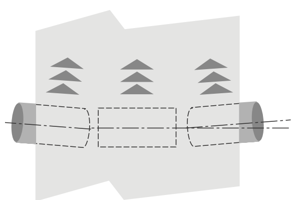
- 2 rollers plain or with rubber rings for return set



- 3 rollers plain for load carrying



- 5 rollers plain for load carrying



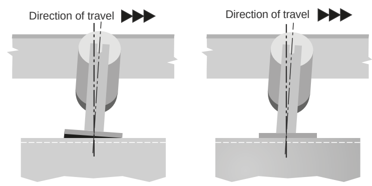


Fig. **12** - Only for single directional belts

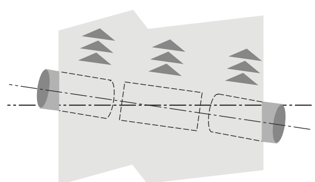


Fig.**13** - Misalignment of the troughing set may promote belt wandering.

Troughing set pitch

The trough set pitch **a_o** most commonly used for the upper strand of a belt conveyor is 1 metre, whilst for the return strand the sets are pitched normally at 3 metres (**a_u**).

indicated limits. Above all the pitch is also limited by the load capacity of the rollers themselves.

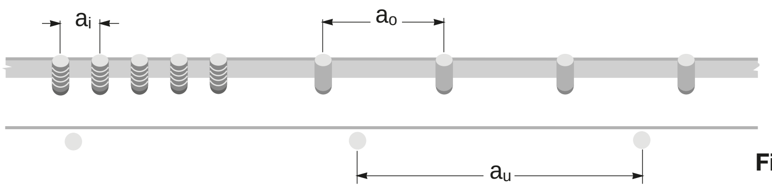


Fig.14

The deflection of the belt between 2 consecutive carrying troughing sets should not be more than 2% of the pitch itself.
A greater deflection causes the discharge of the material during the loading and promotes excessive frictional forces during the belt movement due to the manipulation of the material being conveyed. This not only the increases the horse power and work, but also increases forces on the rollers, and overall a premature belt surface wear occurs.

At the loading points the pitch is generally one half or less, that of the normal pitch of troughing sets so that any belt deflection is limited to the least possible, and also to reduce the load forces on the rollers.

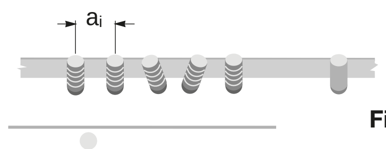


Fig.15

Tab.**6** advises the maximum pitch for troughing sets in relation to belt width and the specific weight of the conveyed material, to maintain a deflection of the belt within the

The calculation of the minimum pitch for suspension sets is calculated to avoid contact between adjoining "garlands" when the normal oscillation of the sets takes place during belt operation Fig.**15**.

Tab. **6 - Maximum advised pitch of troughing sets**

Belt width	Pitch of sets			
	upper			**lower**
	specific weight of conveyed material t/m^3			
	< 1.2	1.2 ÷ 2.0	> 2.0	
mm	m	m	m	m
300	**1.65**	**1.50**	**1.40**	**3.0**
400				
500				
650				
800	**1.50**	**1.35**	**1.25**	**3.0**
1000	**1.35**	**1.20**	**1.10**	**3.0**
1200	**1.20**	**1.00**	**0.80**	**3.0**
1400				
1600				
1800				
2000	**1.00**	**0.80**	**0.70**	**3.0**
2200				

Transition distance Lt

The distance between the last troughing set adjacent to the head or tail pulley of a conveyor and the pulleys themselves is known as the transition distance Fig.**16**.

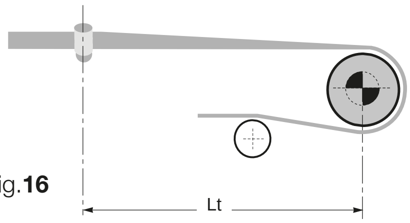
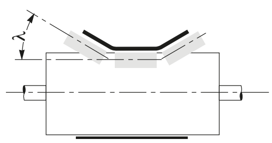


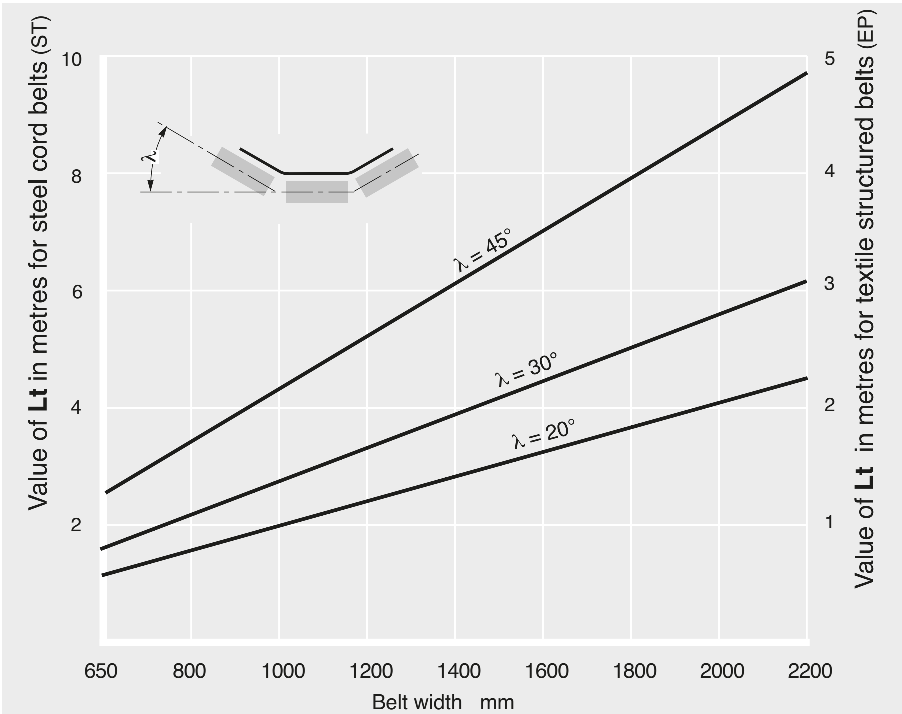
Fig.**16**

Along this section the belt changes from a trough configuration as determined by the inclination of the rollers of the carrying sets to a flat belt to match the flat pulley and vice versa.



The edges of the belt are in this area placed under an extra force which reacts on the side rollers. Generally the transition distance must not be less than the belt width to avoid excess pressures.

Fig.**19** - **Transition distance**



In the case where the transition distance Lt is larger than the pitch of the carrying troughing sets it is a good rule to introduce in this transition area troughing sets with inclined side rollers of gradual reduction in angle (known as transition troughing sets). In this way the belt may change gradually from trough to flat avoiding those damaging forces.

The graph Fig.**19** allows the determination of the transition distance Lt (in relation to the belt width and to the inclination of the side rollers of the troughing sets), for belts with textile structure EP (polyester) and for steel corded belts (ST).

Example:
For a belt (EP) 1400 mm width troughing sets at 45°, one may extract from the graph that the transition distance is about 3 metres.
It is advisable to position in this section Lt two troughing sets with respectively λ=15° and 30° at a pitch of 1 metre.

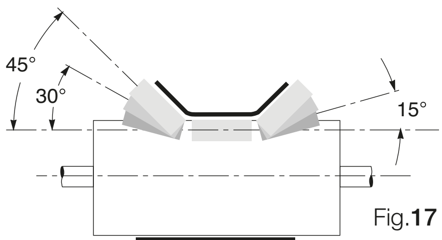


Fig.**17**

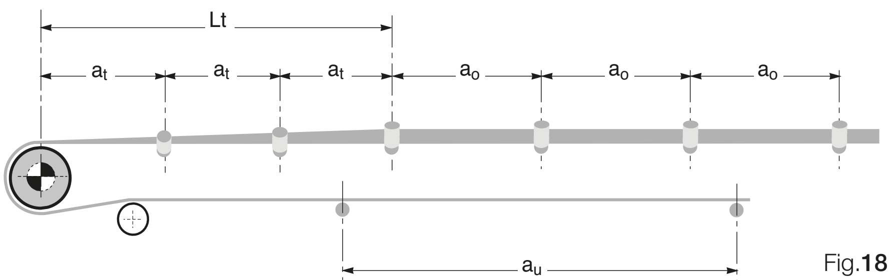


Fig.**18**

1.5.5 - Tangential force, driving power, passive resistance, belt weight, tensions and checks

The forces which act on a running conveyor vary along its length. To dimension and calculate the absorbed power of the conveyor it is necessary to find the existing tensions in the section under the most force and in particular for conveyors with the following characteristics:

- incline of more than 5°

- length of decline

- variable height profile Fig.**20**

Tangential force

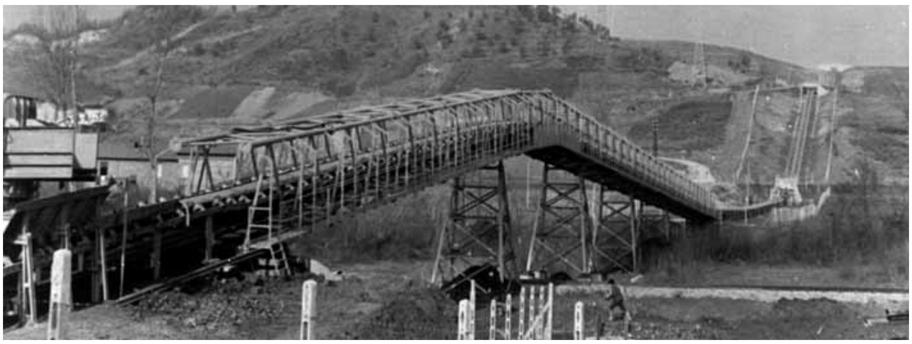
The first step is to calculate the total tangential force F_U at the periphery of the drive pulley. The total tangential force must overcome all the resistance that comes from motion and consists of the sum of the following forces:

- force necessary to move the loaded belt: must overcome the belt frictional forces from the carrying troughing sets upper and lower, the pulleys, return and snub etc.;

- force necessary to overcome the resistance as applied to the horizontal movement of the material;

- force necessary to raise the material to the required height (in the case of a decline, the force generated by the mass changes the resultant power);

- force necessary to overcome the secondary resistances where accessories are present (mobile unloaders, “Trippers”, cleaners, scrapers, rubber skirts, reversing units etc.).



The total tangential force Fu at the drive pulley periphery is given by:

$$F_U = [\, L \times C_q \times C_t \times f \,(\, 2\, q_b + q_G + q_{RU} + q_{RO}\,) \pm (\, q_G \times H \,)\,] \times 0.981 \quad [daN]$$

For decline belts a negative sign (-) is used in the formula

where:

- L = Centres of conveyor (m)
- C_q = Fixed coefficient of resistance (belt accessories), see Tab. **7**
- C_t = Passive coefficient of resistance see Tab. **8**
- f = Coefficient of friction internal rotating parts (troughing sets), see Tab. **9**
- q_b = Belt weight per linear metre in Kg/m, see Tab. **10** (sum of cover and core weight)
- q_G = Weight of conveyed material per linear metre Kg/m
- q_{RU} = Weight of lower rotating parts in Kg/m see Tab. **11**
- q_{RO} = Weight of upper rotating parts in Kg/m see Tab. **11**
- H = Height change of belt.

When it is necessary to calculate the forces on a variable altitude belt conveyor it may be seen that the total tangential force is made up from forces Fa (tangential force to move the belt, upper strand) and the lesser force Fr (tangential force on return strand) all necessary to move a single uniform section of the belt that comprises the conveyor (Fig.**20**) thus we have:

$$F_U = (Fa_1 + Fa_2 + Fa_3 ...) + (Fr_1 + Fr_2 + Fr_3 ...)$$

Where:

Fa = tangential force to move a single section of the belt upper strand

Fr = tangential force to move a single section of the belt lower strand

Therefore the tangential force Fa and Fr will be given by:

$$Fa = [\, L \times Cq \times Ct \times f \; (q_b + q_G + q_{RO}) \pm (q_G + q_b) \times H \,] \times 0.981 \quad [daN]$$

$$Fr = [\, L \times Cq \times Ct \times f \; (q_b + q_{RU}) \pm (q_b \times H) \,] \times 0.981 \quad [daN]$$

Using the indication (+) for belt sections that rise

(-) for sections that fall

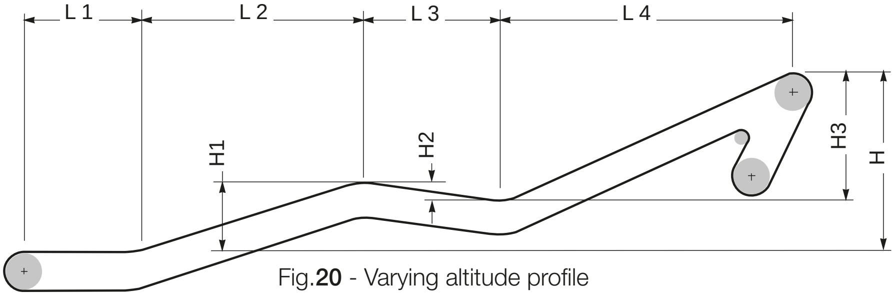


Fig.**20** - Varying altitude profile

Driving power

Noting the total tangential force at the periphery of the drive pulley, the belt speed and the efficiency "η" of the reduction gear, the minimum necessary driving power is:

$$P = \frac{F_U \times v}{100 \times \eta} \quad [kW]$$

1 Technical Information
project and design criteria for belt conveyors

Passive resistance

The passive resistance is expressed by a coefficient which is dependant on the length of the belt conveyor, ambient temperature, speed, type of maintenance, cleanliness and fluidity of movement, internal friction of the conveyed material, and to the conveyor inclinations.

Tab. **7 - Coefficient of fixed resistance**

Centres m	Cq
10	4.5
20	3.2
30	2.6
40	2.2
50	2.1
60	2.0
80	1.8
100	1.7
150	1.5
200	1.4
250	1.3
300	1.2
400	1.1
500	1.05
1000	1.03

Tab. **8 - Coefficient of passive resistance given by temperature**

Temperature °C	+ 20°	+ 10°	0	- 10°	- 20°	- 30°
Fattore Ct	**1**	**1,01**	**1,04**	**1,10**	**1,16**	**1,27**

Tab. **9 - Coefficient of internal friction f** of materials and of the rotating parts

Horizontal belt conveyor rising and gently falling	**speed m/s**					
	1	**2**	**3**	**4**	**5**	**6**
Rotating parts and material with standard internal friction	**0,0160**	**0,0165**	**0,0170**	**0,0180**	**0,0200**	**0,0220**
Rotating parts and material with high internal friction in difficult working conditions	**from 0,023 to 0,027**					
Rotating parts of a conveyor in descent with a brake motor	**from 0,012 to 0,016**					

Belt weight per linear metre q_b

The total belt weight qb may be determined adding the belt core weight, to that of the belt covers upper and lower allowing about 1.15 Kg/m^2 for each mm of thickness of the covers themselves.

*Tab.*10 - **Belt core weight q_{bn}**

Breaking force of belt N/mm	Belt with textile inserts (EP) Kg/m ^{2}	Belt with metal inserts Steel Cord (ST) Kg/m ^{2}
200	2.0	-
250	2.4	-
315	3.0	-
400	3.4	-
500	4.6	5.5
630	5.4	6.0
800	6.6	8.5
1000	7.6	9.5
1250	9.3	10.4
1600	-	13.5
2000	-	14.8
2500	-	18.6
3150	-	23.4

The weights are indicative of the belt core with textile or metallic inserts in relation to the class of resistance.

In *Tab.*11 the approximate weights of rotating parts of an upper transom troughing set and a lower flat return set are indicated.

The weight of the upper rotating parts q_{RO} and lower q_{RU} is given by:

$$q_{RO} = \frac{Pprs}{a_o} \quad [Kg/m]$$

where:

Pprs = weight of upper rotating parts

a_o = upper troughing set pitch

$$q_{RU} = \frac{Ppri}{a_u} \quad [Kg/m]$$

where:

Ppri = weight of lower rotating parts

a_u = return set roller pitch

*Tab.*11 - **Weight of rotating parts of the rollers (upper/lower)**

Belt width	Roller diameter mm									
	89		108		133		159		194	
mm	Pprs Kg	Ppri	Pprs	Ppri	Pprs	Ppri	Pprs	Ppri	Pprs	Ppri
400	—	—	—							
500	5.1	3.7	—							
650	9.1	6.5	—							
800	10.4	7.8	16.0	11.4	—					
1000	11.7	9.1	17.8	13.3	23.5	17.5				
1200			20.3	15.7	26.7	20.7	—			
1400					29.2	23.2	—			
1600					31.8	25.8	—			
1800							47.2	38.7	70.5	55.5
2000							50.8	42.2	75.3	60.1
2200							—	—	—	—

1 Technical Information

project and design criteria for belt conveyors

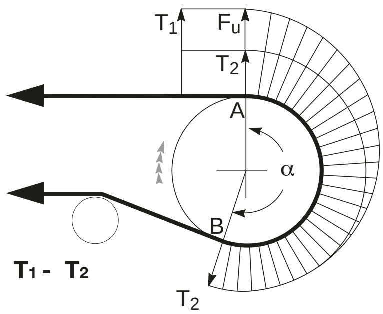
Belt tension

It is necessary to consider the different tensions that must be verified in a conveyor with a powered belt system.

Tensions T_1 and T_2

The total tangential force F_U at the pulley circumference corresponds to the differences between tensionsT_1 (tight side) and T_2 (output side). From these is derived the necessary torque to begin to move the belt and transmit power.

Fig.**21**



$$F_U = T_1 - T_2$$

Moving from point A to point B Fig. **21** the belt tension changes exponentially from value T_1 to value T_2.

The relationship between T_1 and T_2 may be expressed:

$$\frac{T_1}{T_2} \leq e^{fa}$$

where:

f_a = coefficient of friction between belt and drum, given by the angle of wrap

e = natural logarithmic base 2.718

The sign (=) defines the limiting condition of belt adherence. If the ratio $T_1/T_2 > e^{fa}$ the belt will slide on the drive pulley and the movement cannot be transmitted.

From the above formula we may obtain:

$$T_1 = F_U + T_2$$

$$T_2 = F_U \frac{1}{e^{fa} - 1} = F_U \times Cw$$

The value Cw, which defines the wrap factor, is a function of the angle of wrap of the belt on the drive pulley (may 420° when there are double pulleys) and the value of the coefficient of friction f_a between the belt and pulley.

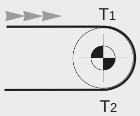
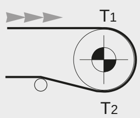
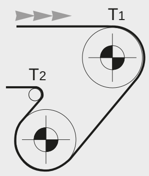
Thus the calculation of the minimum belt tension values is able to be made to the limit of adherence of the belt on the pulley so that the position of a tensioner may be positioned downstream of the drive pulley.

A belt tensioning device may be used as necessary to increase the adherence of the belt to the drive pulley. This will be used to maintain an adequate tension in all working conditions.

On the following pages various types of belt tensioning devices commonly used are described.

Tab. **12** gives the value of the wrap factor Cw in relation to the angle of wrap, the system of tensioning and the use of the pulley in a lagged or unlagged condition.

Tab. **12** - **Wrap factor Cw**

Drive arrangement	Angle of wrap	tension unit or counterweight pulley		screw tension unit pulley	
		unlagged	lagged	unlagged	lagged
	180°	0.84	0.50	1.20	0.80
	200°	0.72	0.42	1.00	0.75
	210°	0.66	0.38	0.95	0.70
	220°	0.62	0.35	0.90	0.65
	240°	0.54	0.30	0.80	0.60
	380°	0.23	0.11	-	-
	420°	0.18	0.08	-	-

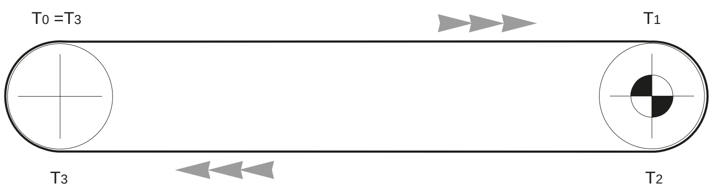


Fig. **22**

Given the values T_1 and T_2, we may analyse the belt tensions in other areas that are critical to the conveyor. These are:

- Tension T_3 relative to the slack section of the return pulley;

- Tension T_0 minimum at tail end, in the material loading area;

- Tension Tg of the belt at the point of connection to the tension unit device;

- Tension T_{max} maximum belt tension.

Tension T_3

As already defined,

$$T_1 = Fu + T_2 \qquad \text{and} \qquad T_2 = F_U \times Cw$$

The tension T_3 that is generated at the belt slackside of the tail pulley (Fig.**22**) is given from the algebraic sum of the tensions T_2 and the tangential forces Fr relative to a single return section of the belt.

Therefore the tension T_3 is given by:

$$T_3 = T_2 + (Fr_1 + Fr_2 + Fr_3 \ldots) \text{ [daN]}$$

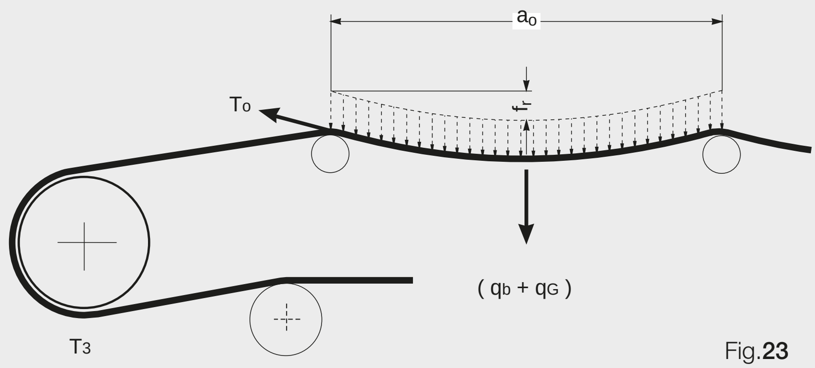


Fig.23

Tension T_0

The minimum necessary tension T_3 at the slack side of the return pulley, besides guaranteeing the belt adhesion to the driving pulley so as to trasmit the movement must also guarantee a deflection not superseding 2% of the length of pitch between consecutive trounghing sets.

Furthermore the tensions must avoid material spillage from the belt and excessive passive resistance caused by the dynamics of material as the belt travels over the troughing sets Fig. 23.

The minimum tension T_0 necessary to maintain a deflection of 2% is given by the following formula:

$$T_0 = 6.25 (q_b + q_G) \times a_0 \times 0{,}981 \text{ [daN]}$$

where:

q_b = total belt weight per linear metre

q_G = weight of conveyed material per linear metre

a_0 = pitch of troughing sets on upper strand in m.

The formula derives from the application and essential simplification of theory, when considering “catenaries”.

To alter as desired the deflection to a value less than 2%, the figures 6.25 may be substituted by:
- for 1.5% deflection = 8,4
- for 1.0% deflection = 12,5

In order to have a tension able to guarantee the desired deflection, it will be necessary to apply a tensioning device, also effecting the tensions T_1 and T_2 to leave unchanged the circumferential force $F_U = T_1 - T_2$.

Tension Tg and tensioning devices

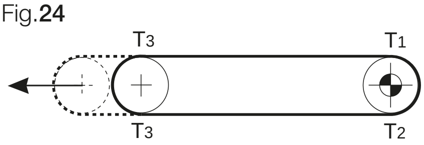
Tension devices used generally on belt conveyors are screw type or counterweight. The screw type tension unit is positioned at the tail end and is normally applied to conveyors where the centres are not more than 30/40 m.
Where conveyors are of larger centres the counterweight tension unit is used or winch style unit where space is at a premium.

The tension unit minimum movement required is determined as a function of the type of belt installed, that is:

- the stretch of a belt with textile core needs a minimum 2% of the conveyor centres;
- the stretch of a belt with metal or steel core needs a minimum of 0.3 + 0.5% of the conveyor centres.

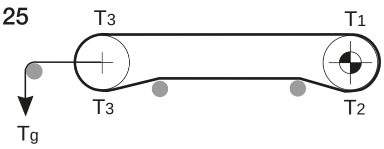
Typical tension device

Fig.**24**



In this arrangement the tension is regulated normally with the occasional periodic check of the tensioning screw.

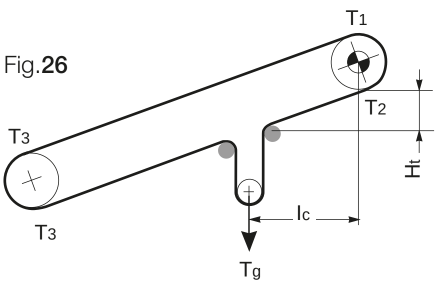
Fig.**25**



In this arrangement the conveyor is tensioned using a counterweight.

$$Tg = 2\,(T_3) \quad [daN]$$

Fig.**26**



Also in this arrangement the conveyor is tensioned using a counterweight.

$$Tg = 2T_2 + 2\,[(\,I_C \times Cq \times Ct \times f\,)\,(\,q_b + q_{RU}\,) \pm (\,Ht \times q_b\,)]\ 0{,}981 \quad [daN]$$

In which:

I_C = distance from centre of drive pulley to the counterweight attachment point

Ht = belt height change from the point where the counterweight applies itself to the point where the belt exits from the slack side of the pulley, measured in metres.

Correct dimensioning verification

The belt will be adequately dimensioned when the essential tension T_0 (for the correct deflection of the belt) is less than the calculated tension T_3 the tension T_2 has always to be $T_2 \geq Fu \times Cw$ and is calculated as $T_2 = T_3 \pm Fr$ (where $T_3 \geq T_0$).

Maximum tension (T_{max})

This is the belt tension at the point where the conveyor is under the greatest stress.

Normally it is coincidental in value with tension T_1. Along the length of a conveyor with variable height change and in particular where conditions are variable and extreme, T_{max} may be found in different sections of the belt.

Working load and belt breaking strain

T_{max} is used to calculate the unitary maximum tension of the belt Tu_{max} given that:

$$Tu_{max} = \frac{T_{max} \times 10}{N} \quad [N/mm]$$

where:

N = belt width in mm;

T_{max} = tension at the highest stress point of the belt in daN.

As a security factor one may consider the maximum working load of the belt with textile core to correspond to 1/10 of the breaking load of the belt (1/8 for a belt with steel core).

1.5.6 - Belt conveyor drives and pulley dimensions

Type of drives

Conveyors requiring power up to 250 kW are traditionally driven at the head pulley with electric motor, gearbox, pulley, guards, transmission accessories etc., or, alternatively by motorised pulley. Fig.**27**.

In the drawings Fig.**28** a comparison is made between the space needed for two drive systems.

Belt conveyors that need power over 250 kW utilise the conventional drive pulley arrangement but also with two or more motor gearboxes.

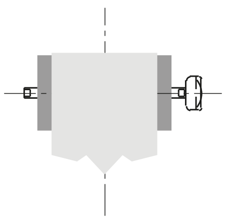


Fig.**27**

The motorised pulley is used today more and more as the drive for belt conveyors thanks to its characteristics and compactness. It occupies a minimal space, is easy to install, its motor is protected to IP67, all working parts are inside the pulley and therefore it needs very limited and occasional maintenance (oil change every 10.000 or 50.000 working hours with synthetic oil).

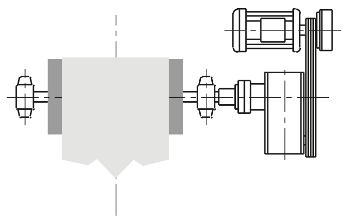
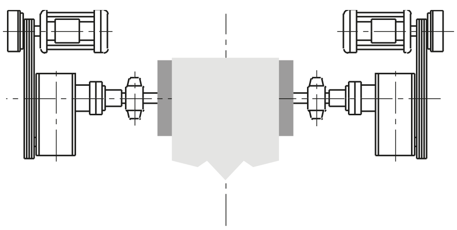


Fig.**28**



Pulley diameters

The dimensioning of the diameter of a head pulley is in strict relationship to the characteristics of the type of belt used.

In *Tab.* **13** the minimum diameters recommended in relation to the type of belt used are indicated, avoiding damaging de-layering of the belt layers or laceration of the reinforcing fabric.

Tab. **13 - Minimum pulley diameters recommended**

Belt breaking load	Belt with textile core EP DIN 22102			Belt with steel core ST DIN 22131		
	Ø motorised pulley	return pulley	direction change	Ø motorised pulley	return pulley	direction change
N/mm	mm		drum	mm		pulley
200	200	160	125	-	-	-
250	250	200	160	-	-	-
315	315	250	200	-	-	-
400	400	315	250	-	-	-
500	500	400	315	-	-	-
630	630	500	400	-	-	-
800	800	630	500	630	500	315
1000	1000	800	630	630	500	315
1250	1250	1000	800	800	630	400
1600	1400	1250	1000	1000	800	500
2000	-	-	-	1000	800	500
2500	-	-	-	1250	1000	630
3150	-	-	-	1250	1000	630

Minimum diameters recommended for pulleys in mm up to 100% of the maximum working load as recommended RMBT ISO bis/3654.

This table must not be applied to belt conveyors that convey material with a temperature over +110°C or for conveyors installed where the ambient temperature is less than -40°C.

Sizing of the drive pulley

The shaft of the drive pulley is subject to alternating flexing and torsion, causing fatigue failure.

To calculate correct shaft diameter it is necessary to determine the bending moment Mf and the torsion moment Mt.

The bending moment of the shaft is generated as a result of the sum of the vector of tensions T_1 and T_2 and the weight of the pulley itself q_T Fig.**29**.

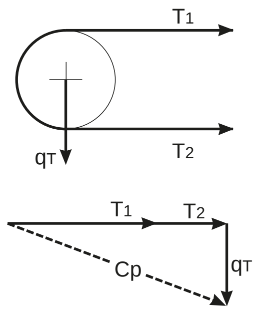


Fig. **29**

The dimensioning of the shaft diameter requires the determination of various values.

These are: the resultant of tensions Cp, the bending moment Mf, torsional moment Mt, the ideal bending moment Mif and the module of resistance W.

Proceeding in order we have:

$$Cp = \sqrt{(T_1 + T_2)^2 + q_t^2} \quad [daN]$$

$$Mf = \frac{Cp}{2} \times a_g \quad [daNm]$$

$$Mt = \frac{P}{n} \times 954{,}9 \quad [daNm]$$

where:

P = absorbed power in kW
n = r.p.m. of the drive pulley

$$Mif = \sqrt{Mf^2 + 0{,}75 \times Mt^2} \quad [daNm]$$

$$W = \frac{Mif \times 1000}{\sigma_{amm}} \quad [mm^3]$$

$$W = \frac{\pi}{32} \times d^3 \quad [mm^3]$$

from the combination of simultaneous equations we may discover the diameter of the shaft as follows:

$$\mathbf{d} = \sqrt[3]{\frac{W \times 32}{\pi}} \quad [mm]$$

Tab. **14 - Suggested value of σ**

Steel type	daN/mm^2
38 NCD	12,2
C 40 Tempered	7,82
C 40 Normalised	5,8
Fe 37 Normalised	4,4

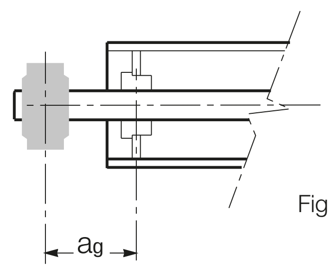


Fig.**30**

Fig.**31 - Tail or return pulley**

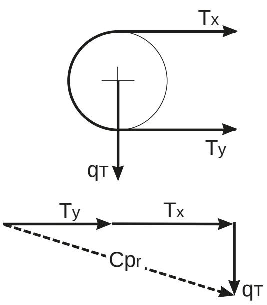
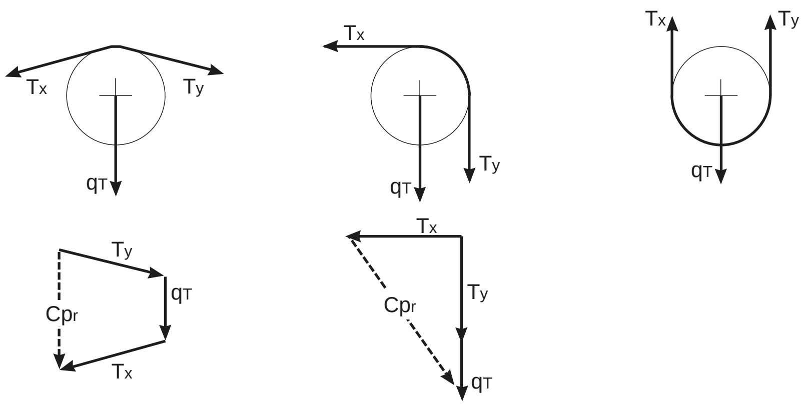


Fig.**32 - Change direction pulley**



Sizing of the tail or return pulley shaft and change direction pulley

In this case only shaft flexure must be considered, torsional loads are not a factor in fatigue failure.

The bending moment **Mf** must be determined as generated by the resultant of the sum of the vectors of belt tensions where the belt is before or after the pulley and the weight of the pulley itself.
In this case, treating the pulley as an idler one may consider Tx=Ty.
In Fig.**31** and **32** various arrangements for an idler return pulley are indicated.

The bending moment is given by:

$$Mf = \frac{Cp_r}{2} \times a_g \quad [daNm]$$

the module of resistance is found from:

$$W = \frac{Mf \times 1000}{\sigma_{amm}} \quad [mm^3]$$

given the module of resistance:

$$W = \frac{\pi}{32} \times d^3 \quad [mm^3]$$

the diameter of the shaft is given by:

$$\mathbf{d} = \sqrt[3]{\frac{W \times 32}{\pi}} \quad [mm]$$

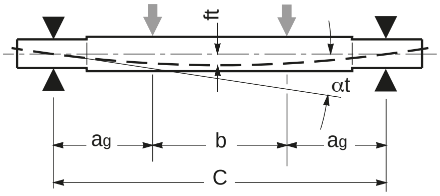
Limits of deflection and angle for drive and idler pulleys

After having sized the shafts of different pulleys, one is required to verify that the deflection and angle of the shaft does not exceed certain values.

In particular the deflection **ft** and the angle **αt** must respect the relationship:

$$ft_{max} \leq \frac{C}{2000} \qquad \alpha t \leq \frac{1}{500}$$

Fig.**33**



$$ft = \frac{(Cp_r/2)a_g}{24xExJ}\left[3(b+2a_g)^2 - 4a_g^2\right] \leq \frac{C}{2000}$$

$$\alpha t = \frac{(Cp_r/2)}{2xExJ}\, a_g\,(C - a_g) \leq \frac{1}{500}$$

where:
- a_g = expressed in mm
- E = module of elasticity of steel (20600 [daN/mm²])
- J = sectional moment of inertia of the shaft (0,0491 D^4 [mm^4])
- Cp_r = load on shaft [daN]
- ft = shaft deflection [mm]
- αt = shaft angle at the pillow blocks [rad]

1.6 - Rollers, function and design criteria

In a conveyor, the elastomer belt represents the most perishable and costly item. The rollers that support the belt along its length are no less important, and therefore they should be designed, chosen and manufactured to optimise their working life and that of the belt itself.

The resistance to start up and rotation of rollers has a great influence on the belt and in consequence to the necessary power to move the belt and keep it moving.

The body of the roller and that of its end caps, the bearing position and its accompanying system of protection, are the principal elements which impact the life and torque characteristics of the roller.

Refer to chapter 2 where the construction criteria of rollers for belt conveyors are presented along with the factors which must be taken into account for a correct project design.

In the following sections we should examine other factors such as the:

- balance and start up resistance;
- tolerances;
- type of roller shell; characteristics of the tube and thickness
  - the fitting of the end caps;
- frictional resistance and impact resistance;

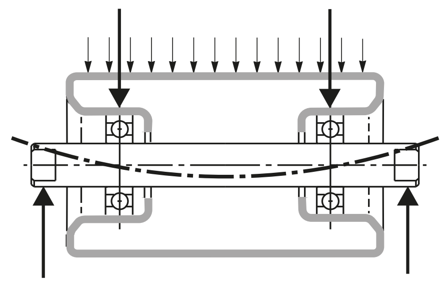


Fig.**34**

- type of bearing
  - protection system;
  - fit to the spindle and end caps;
  - lubrication;
  - alignment;

- spindle: characteristics and manufacturing tolerances.

1.6.1 - Choice of roller diameter in relation to speed

We have already stated that one of the important factors in the design of a conveyor is the speed of the belt movement in relation to the load conditions required.

From the belt speed and roller diameter we are able to determine the revolutions per minute of the roller using the formula:

$$n = \frac{v \times 1000 \times 60}{D \times \pi} \text{ [r.p.m.]}$$

where:

D = roller diameter [mm]
v = belt speed [m/s]

Tab. **15** gives the existing relationship between maximum belt speed, roller diameter and the relative r.p.m.

In choosing the roller it is interesting to note that even if a roller of larger diameter exhibits a higher inertia on start up, it actually yields, other conditions being equal, many advantages such as: less revolutions per minute, less wear of bearings and housing, less rolling friction and reduced wear between the roller and the belt.

Tab. **15 - Maximum speed and numbers of roller revolutions**

Roller diameter mm	Belt speed m/s	r.p.m. n
50	1.5	573
63	2.0	606
76	2.5	628
89	3.0	644
102	3.5	655
108	4.0	707
133	5.0	718
159	6.0	720
194	7.0	689

The correct choice of diameter must take into consideration the belt width.
Tab. **16** shows the diameter of rollers in relation to belt width.

*Tab.***16 - Roller diameter advised**

Belt width mm	For speed ≤ 2 m/s Ø roller mm			2 ÷ 4 m/s Ø roller mm			≥ 4 m/s Ø roller mm		
500	89			89					
650	89			89	108				
800	89	108		89	108	133	133		
1000	108	133		108	133		133	159	
1200	108	133		108	133	159	133	159	
1400	133	159		133	159		133	159	
1600	133	159		133	159	194	133	159	194
1800	159	159	194	159	194				
2000	159	194		159	194		159	194	
2200 and more	194			194			194		

One may have indicated more diameters where the choice will be made in relation to the material lump size and the severity of working conditions.

1.6.2 - Choice in relation to load

The type and dimensions of rollers used in belt conveyors depends mainly on the width of the belt itself, the pitch of the troughing sets, and above all, the maximum load on the rollers most under pressure, not withstanding other correction factors.

The calculation of load forces is normally made by the project designer of the plant. Nevertheless, as a check or in the case of simple conveyors, we present the following concepts for determining the facts.

The first value to define is the load on the troughing sets. Following this, depending on the type of troughing set (carrying, return or impact), the number of rollers in a transom or frame, the angles of the side roller, the material lump size and other relevant factors as listed below. One is able to calculate the roller load with the maximum force for each type of troughing set.

Furthermore there are some correction factors keeping count of the plant working hours per day (service factor), of the environmental conditions and of the speed for the different diameters of the rollers.

The load value obtained in this way may be compared with the load capacity of the rollers indicated in this catalogue valid for a project life of 30,000 hours. For a theoretically different life, the load capacity may be multiplied by a coefficient reported on *Tab.***22** corresponding to life required.

Principal relevant factors:

Iv	=	belt load	t/h
v	=	belt speed	m/s
a_o	=	pitch of the troughing sets upper strand	m
a_u	=	pitch of the return roller set	m
q_b	=	weight of belt per linear metre	Kg/m
Fp	=	participation factor of roller under greatest stress see *Tab.***17** (depends on the angle of the roller in the transom)	
Fd	=	impact factor see *Tab.***20** (depends on the material lump size)	
Fs	=	service factor see *Tab.***18**	
Fm	=	environment factor see *Tab.***19**	
Fv	=	speed factor see *Tab.* **21**	

Tab. **17 - Participation factor Fp - loaded rate on the most loaded roller**

0°	20°	20°	30°	35°	45°	30°-45°	60°
1.00	0.50	0.60	0.65	0.67	0.72	~ 0.52 - 0.60	0.47
						Shorter central roller	5 rollers garland

Tab. **18 - Service factor**

Life	Fs
Less than 6 hours per day	0.8
From 6 to 9 hours per day	1.0
From 10 to 16 hours per day	1.1
Over 16 hours per day	1.2

Tab. **19 - Environment factor**

Conditions	Fm
Clean and regular maintenance	0.9
Abrasive or corrosive material present	1.0
Very abrasive or corrosive material present	1.1

Tab. **20 - Impact factor Fd**

Material lump size	Belt speed m/s						
	2	2.5	3	3.5	4	5	6
0 ÷ 100 mm	1	1	1	1	1	1	1
100 ÷ 150 mm	1.02	1.03	1.05	1.07	1.09	1.13	1.18
150 ÷ 300 mm in layers of fine material	1.04	1.06	1.09	1.12	1.16	1.24	1.33
150 ÷ 300 mm without layers of fine material	1.06	1.09	1.12	1.16	1.21	1.35	1.50
300 ÷ 450 mm	1.20	1.32	1.50	1.70	1.90	2.30	2.80

Tab. **21 - Speed factor Fv**

Belt speed	Roller diameter mm						
m/s	60	76	89-90	102	108-110	133-140	159
0.5	0.81	0.80	0.80	0.80	0.80	0.80	0.80
1.0	0.92	0.87	0.85	0.83	0.82	0.80	0.80
1.5	0.99	0.99	0.92	0.89	0.88	0.85	0.82
2.0	1.05	1.00	0.96	0.95	0.94	0.90	0.86
2.5			1.01	0.98	0.97	0.93	0.91
3.0			1.05	1.03	1.01	0.96	0.92
3.5					1.04	1.00	0.96
4.0					1.07	1.03	0.99
4.5					1.14	1.05	1.02
5.0					1.17	1.08	1.00

Tab. **22 - Coefficient of theoretical life of bearing**

Theoretic project life of bearing	10'000	20'000	30'000	40'000	50'000	100'000
Coefficient with base 30'000 hours	1.440	1.145	1.000	0.909	0.843	0.670
Coefficient with base 10'000 hours	1	0.79	0.69	0.63	---	---

Load calculation

Having defined the roller diameter in relation to the speed and the number of revolutions one may then proceed to calculate the static load on the carrying troughing set using the following formula:

$$Ca = a_o \times \left(q_b + \frac{I_v}{3.6 \times v} \right) 0{,}981 \quad [daN]$$

Multiplying then by a working factor we have the dynamic load on the transom:

$$Ca_1 = Ca \times Fd \times Fs \times Fm \quad [daN]$$

Multiplying then by the participation factor one may obtain the load on the roller carrying the most force (central roller in the case of a troughing set transom where all the rollers are of equal length):

$$ca = Ca_1 \times Fp \quad [daN]$$

The static load on the return roller set, not having any material load present, is given by the following formula:

$$Cr = a_u \times q_b \times 0{,}981 \quad [daN]$$

The dynamic load on the return roller set will be:

$$Cr_1 = Cr \times Fs \times Fm \times Fv \quad [daN]$$

And the load on the rollers of the return roller set, single or double, will be:

$$cr = Cr_1 \times Fp \quad [daN]$$

Given the values of “ca” and “cr” one may look in the catalogue for rollers (first by diameter) that have a sufficient load capacity.

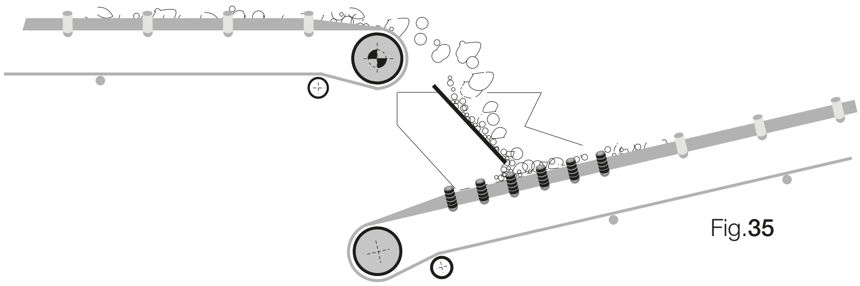


Fig.**35**

1.7 - Loading of belt and impact rollers

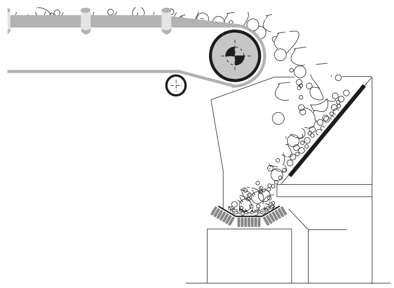


Fig.**36**

The feed system of material falling or dropping onto a belt conveyor must be constructed to minimise or eliminate impact damage to the belt material and surface. This is of particular importance when the material falls from a considerable height and consists of large lumps with sharp edges.
The rollers supporting or carrying the belt in the loading zone are normally installed as impact design (with rubber rings), mounted onto troughing set frames set close to each other. In this way the belt is supported in a flexible manner.

It is a widely held view that the use of suspension sets of the "garland" design Fig.**37-38**, thanks to their intrinsic flexible characteristics absorb with great efficiency the impact of materials falling onto the belt and, what is more, the "garland" is able to adapt to conform to the shape of the charge (or load).

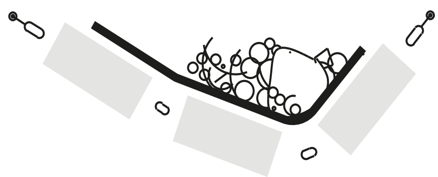


Fig.**37**

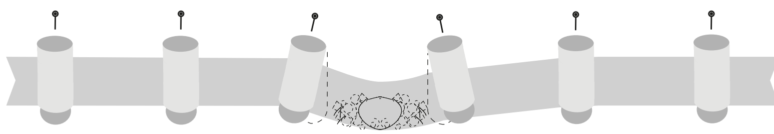


Fig.**38**

1 Technical Information

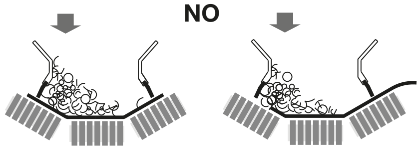
project and design criteria for belt conveyors

Particular attention must be paid at the project stage to the feed system and to the design of impact troughing sets.

The project designer of the conveyor system must take into account that:

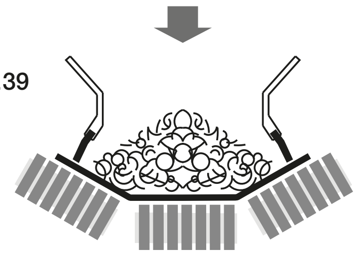
- the impact of material onto the belt must take place in the conveyor direction and at a speed that approximates to the speed of the belt;

NO



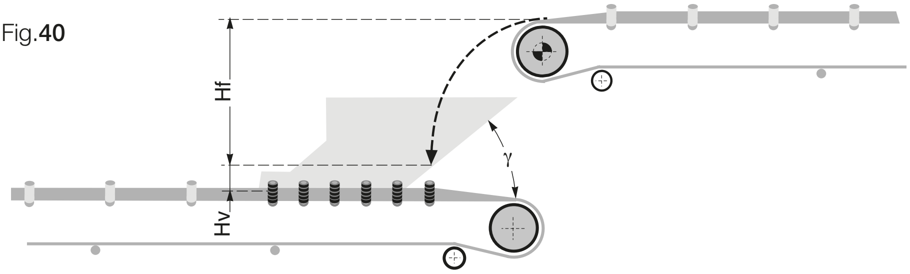
- the loading hopper is positioned so that material falling from it is deposited as near as possible to the centre of the belt;

Fig.**39**



- the height that the material falls must be reduced to the minimum possible, compatible with the requirements of the plant design.

Fig.**40**



Please refer to chapter 3 of this catalogue for greater detail regarding the programme of the design of impact rollers with rubber rings of high shock absorbing qualities and for the programme of suspension sets as "garland" design.

1.7.1 - Calculation of associated forces on impact rollers

The definition of the correct load fall height Hc may be given by the folowing formula:

$$Hc = Hf + Hv \times sen^2 \gamma$$

where:

Hf = fall height from the upper face of the loading belt to the contact point of material contained in the hopper;

Hv = height from the contact point of material contained in the hopper to the belt face of the lower belt;

γ = hopper inclination angle.

In the choice of impact rollers we propose to follow two significant design aspects:

- constant loading with uniform fine material;

- loading with material consisting of large lumps.

Constant loading with uniform fine material

Impact rollers must be designed not only to carry the load of material arriving on the belt (as in a normal carrying troughing set) but also the impact load from falling material.

For loose, homogenous fine material the impact force pi, given the corrected fall height, is calculated according to the following formula:

$$pi \cong Iv \times \frac{\sqrt{Hc}}{8} \quad [Kg]$$

where:

Iv = flow of material in t/hr (the belt load capacity)

The force acting on the central roller pic, clearly the roller with the most stress, is obtained on consideration of the previously mentioned participation factor Fp. Various factors depend principally on the angle λ wich is the side roller angle:

$$pic \cong Fp \times pi = Fp \times Iv \times \frac{\sqrt{Hc}}{8} \quad [Kg]$$

One assumes as a rule:

Fp = 0.65 per $\lambda = 30°$
Fp = 0.67 per $\lambda = 35°$
Fp = 0.72 per $\lambda = 45°$

Example:

Let us calculate the central roller load in a transom, given that the loading of the material onto the belt is:

Iv = 1800 t/h, Hc = 1.5m and $\lambda = 30°$:

$$pi = 1800 \times \frac{\sqrt{1.5}}{8} = 275 \text{ Kg}$$

On the central roller we have:

pic = Fp x pi = 0.65 x 275 = 179 Kg

Adding to this load value as considered on a horizontal belt we may obtain the total load on the troughing set central roller.

Refer to the paragraph "roller choice" for design characteristics of the most suitable roller.

Loading with material consisting of large lumps

The force of dynamic load pd on the central roller may be calculated using Gm which is the weight of large blocks of single lumps of material and takes into account the elasticity Cf of the transom and rollers.

$$pd \cong Gm + \sqrt{(2 \times Gm \times Hc \times Cf)} \quad [Kg]$$

where:

Gm = weight of large lumps of material [Kg]
Hc = corrected fall height [m]
Cf = elasticity constant of the transom/ impact rollers.

The impact force is considered as distributed over the 2 bearings of the central load carrying roller.

The approximate weight of the lump may be extracted from the graph in Fig.**41**: one may note that as well as taking the length into account the weight depends on the form of the lump itself.

The graph of Fig.**42** records the constant of elasticity for the most commonly used systems of support and shock absorbing (fixed troughing sets with steel rollers, fixed troughing sets with rollers with rubber rings, troughing sets with "garland" suspension design) and the impact forces resultant on the roller for varying drop energies of the falling load Gm x Hc.

The graph shows above all the static load on the roller bearings derived from Gm x Hc but with a safety factor 2 and 1.5.

The coefficient of elasticity depends on various factors such as the type of rubber used in the rings, length and weight of the rolers, number and articulation of the suspension set as a "garland", and type and elasticity of the flexible parts used by the stock absorbing supports.

The calculation of the dynamic load force pd must fore cast an accurate valuation of these factors.

Example:

A load of 100 Kg falls from a height Hc of 0.8 m onto a suspension "garland" style set, with rollers made from normal steel (coeff, Cf hypothetically 20,000 Kg/m = 200 Kg / cm).

Calculation of the drop energy:

Gm x Hc = 100 x 0.8 = 80 Kgm

Calculating from the table the dynamic force of fall:

pd = 1800 Kg.

Assuming a safety factor of 2 we must have bearings that may withstand a static load of 1800 Kg (2 bearings) that is rollers from series PSV/7-FHD (bearings 6308; Co = 2400 Kg).

Fig.**41** - **Weight of lump of material**

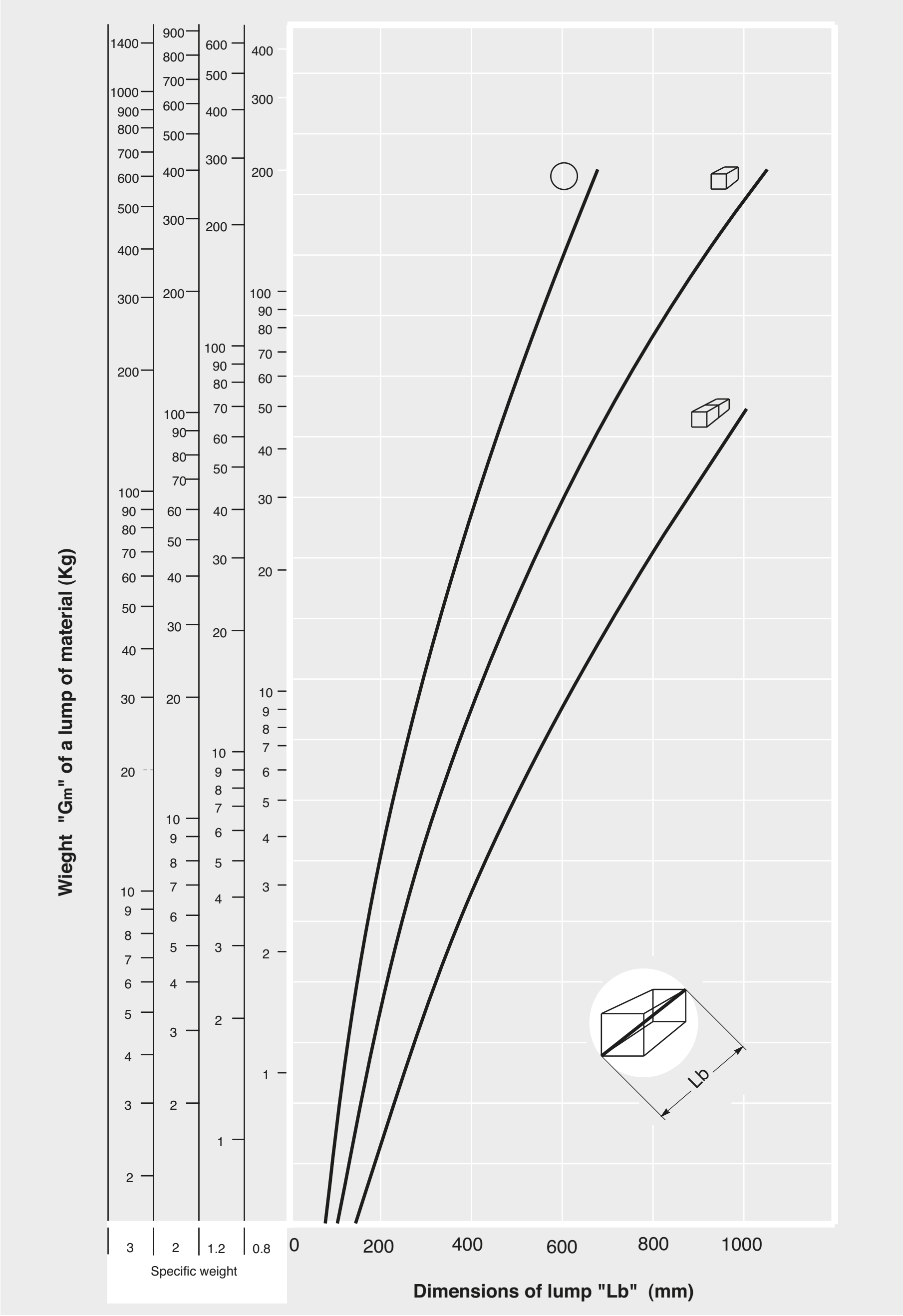
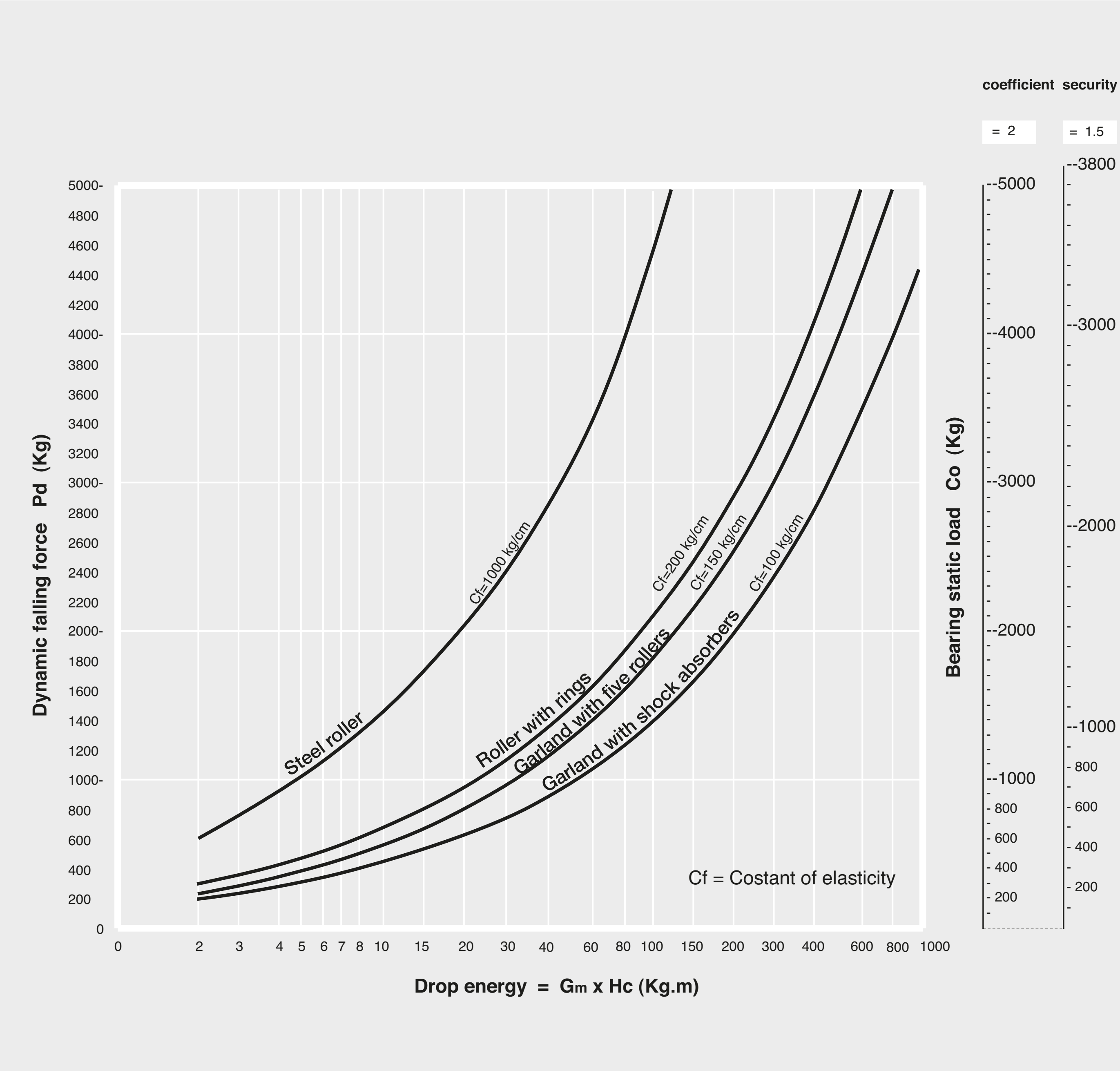


Fig.**42 - Constant of elasticity Cf**

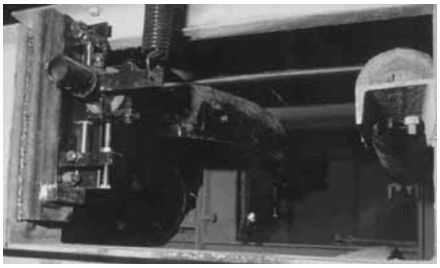


1.8 - Other accessories

Amongst all of other conveyor components, the belt cleaning system and covers are regarded in certain situations of fundamental importance and must be considered at an early stage in the project design of the conveyor itself.

1.8.1 - Belt cleaners

Savings in utilising efficient systems of belt cleaning may be amply demonstrated, in particular resulting from a reduction in belt maintenance time and increased production, proportional to the quantity of material recovered in the process and a large increase in the life of moving parts.



There are a variety of devices used for belt cleaning. The majority of these may be divided into two groups: static and dynamic.

Fig.**44**

The static systems that are utilised the most are the most diverse as they may be applied along all positions on the dirty side of the belt. They are acting directly on the belt using a segmented blade. Fig. **44**

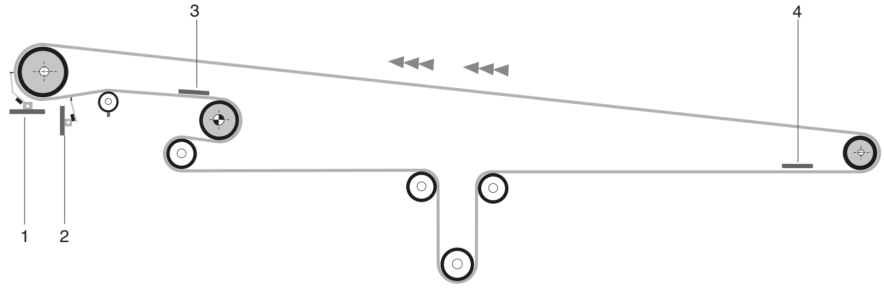


Fig.**43** - Ideal positions for the installation of cleaning devices

1 on drive pulley

2 at about 200mm after the tangential point where belt leaves pulley

3 on internal side of belt on the return section and before the snub pulleys or directional change pulley

4 on internal side of belt before the return pulley

The dynamic systems where motors are used are of less variety and more costly in terms of capital cost, installation and commissioning.

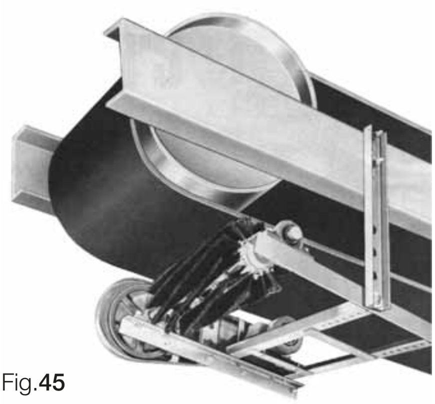


Fig.**45**

They consist of pulleys or motorised pulleys on which are assembled or fixed special brushes, that are then in direct contact with the belt. Fig.**45**

Other cleaners are those of plough or deviator design that are applied to the inside strand of the belt return section.

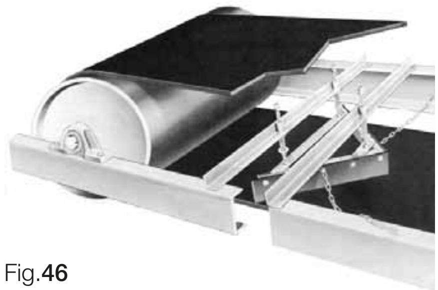


Fig.**46**

They are used to remove material deposited before the drive and return pulleys or certain other points where the material may become trapped between the pulley and belt, affecting the orderly tracking of the belt. Fig.**46**.

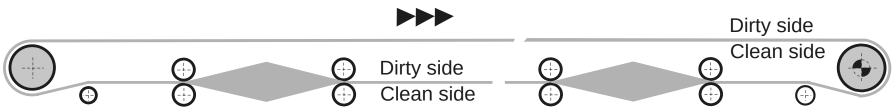
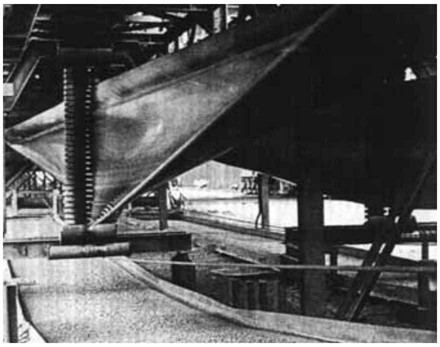


Fig.47

1.8.2 - Belt inversion

On return sections of the belt on very long conveyors, the belt is turned over 180° to reduce the phenomena of adhesion of material residue on the rollers and on the cross member of the troughing sets. The return strand of the belt may be turned over 180° after the drive drum and subsequently turned to its original position before the return drum.



Turning the belt over is generally effected by means of a series of rollers orientated as required. The minimum length to turn over a belt is generally about 14/22 times its width.

The rollers on the return set, thanks to this device, are no longer in contact with the carrying upper strand of the belt which is encrusted with material residue.

1.8.3 - Belt conveyor covers

After having defined the components of primary importance the project designer considers secondary accessories, such as covers.

The necessity to protect the belt conveyor is dictated by the climate, the characteristics of the conveyed material (dry, light, "volatile") and the type of plant.

1.9 - Project examples of a belt conveyor

To clarify our presentation of critical tensions in various sections of the belt conveyor here is a project example.
The relative data concerning the conveyed material and its physical/chemical characteristics are as follows:

Material:
- clinker of cement (*Tab.* 2 pag.20)
- specific weight: 1.2 t/m^3
- lump size 80 to 150 mm
- abrasiveness: very abrasive
- angle of friction natural or at rest: ~ 30°

Required load:
I_V = 1000 t/h corresponding to the volumetric load
I_M = 833 m^3/h

Plant characteristics:
- centres 150 m
- change of height H = + 15 m (rising)
- inclination = 6°~
- working conditions: standard
- utilisation: 12 hours per day

From the data supplied we are able to calculate:
speed, belt width, design and type of conveyor troughing sets.

Furthermore we may define: the belt tensions in various critical areas and from these the absorbed power and the belt type.

Speed and belt width
From ***Tab.* 3** (pag.23) we are able to define that the said material may be grouped into B and given that the lump size is 80/150 mm the maximum advised speed results as 2,3 m/s.

From ***Tab.* 5** (pag.26-30) we may evaluate which type and design of carrying troughing sets are needed, given the speed just found, that satisfies the volumetric load I_M required as 833 m^3/h.

To obtain the result one must calculate the volumetric load I_{VT} (for the speed v = 1m/s) given the inclination of the conveyor δ = 6°.

$$I_{VT} = \frac{I_M}{v \times K \times K_1} \quad [m^3/h]$$

in which:
I_M = volumetric load

v = belt speed

K = crrection coefficient to suit the inclination 6°: 0,98 (diagram Fig 8 pag.31).

K_1 = correction coefficient to suit the feed irregularity: 0,90 (pag.31)

Substituting we have:

$$I_{VT} = \frac{833}{2,3 \times 0,98 \times 0,90} = 410 \ m^3/h$$

Given the angle of repose of the material in question is about 30° from *Tab.* **1** pag.19 we may deduce that the angle of surcharge would be established in the order of 20°.

Having chosen a carrying troughing set with a transom side roller angle of $\lambda = 30°$, the belt width that meets the load requirement I_{VT} of 410 m^3/h at 1 m/s is 1000 mm.

Troughing set pitch

The pitch may be chosen as a function of the deflection of the belt between two consecutive troughing sets.

Tab. **6** pag.34 shows how to determine the maximum pitch of troughing sets, as a function of the belt width and the specific weight of the conveyed material.

We need to verify that the deflection does not supersede 2% of the pitch.

A greater deflection may give rise to material mass deformation during the belt movement, and consequently elevated friction.

Then we would be able to determine a major factor: that is major power absorption, giving rise to unusual stresses whether on the rollers or in the belt over and above the premature wear in the cover of the belt.

In our example, given that the belt width is 1000 mm with specific weight of material of 1.2 t/m^3 the tables indicate that:

- for the carrying troughing sets the advised pitch is that of 1.2 m;

- for the return sets the advised pitch is that of 3.0 m.

Roller choice

In *Tab.* **16** pag.49 with a belt of 1000 mm and a speed of 2.3 m/s we may choose rollers with diameter 108 mm.

We may now proceed to determine the load falling on the roller in the carrying strand and those of the return strand.

Assuming we may use a belt with a resistance class equal to 315 N/mm, with cover thickness 4+2, and with a value q_b of 9,9 kg/m, we have:

- for carrying rollers the static load will be:

$$Ca = a_o \times \left(q_b + \frac{I_V}{3,6 \times v} \right) \times 0,981 \quad [daN]$$

$$Ca = 1,2 \left(9,9 + \frac{1000}{3,6 \times 2,3} \right) 0,981 = 153,8$$

the dynamic load will be:

$$Ca^1 = Ca \times Fd \times Fs \times Fm \quad [daN]$$

$$Ca^1 = 153,8 \times 1,03 \times 1,1 \times 1 = 174,2$$

where:

$Fd = 1,03$ from table 20 pag.51
$Fs = 1,1$ from table 18 pag.51
$Fm = 1$ from table 19 pag.51

the load on the central roller of a carrying troughing set is given by:

$$ca = Ca^1 \times Fp \quad [daN]$$

$$ca = 174,2 \times 0,65 = 113,2$$

where from *Tab.* **17** pag.50 the participation factor of a troughing set 30° $Fp = 0,65$

- for the return rollers the static load will be:

$$Cr = a_u \times q_b \times 0,981 \quad [daN]$$

$$Cr = 3 \times 9,9 \times 0,981 = 29,2$$

the dynamic load will be:

$$Cr_1 = Cr \times Fs \times Fm \times Fv \quad [daN]$$

$$Cr_1 = 29,2 \times 1,1 \times 1 \times 0,97 = 31,2$$

where:

$Fv = 0,97$ speed factor (it has been considered that relative to 2,5 m/s see Tab. 21, pag.51)

choosing the return troughing set with plain roller the load on the return roller will be:

$$cr = Cr_1 \times Fp \quad [daN]$$

$$cr = 31,2 \times 1 = 31,2$$

where from *Tab.* **17** the participation factor with return plain roller set $Fp = 1$

We are able therefore to choose a belt 1000 mm, the rollers for carring and return idlers both of loaded and return belt (see Chapter 2):

- rollers for carrying idlers type PSV/1-FHD, ø 108 mm, with bearings 6204 of length C = 388 mm with load capacity 148 Kg that satisfies the required loading of 113,2 Kg;

- return roller type PSV/1-FHD, ø 108 mm, with bearings 6204, length C = 1158 mm with load capacity 101 Kg that satisfies the required loading of 31,2 Kg.

Tangential force and absorbed power

We may now determine the total tangential force Fu at the drum periphery extracting the values q_{RO}, q_{RU} and q_G.

given:

D = 108 roller diameter
f = 0,017 friction coefficient inside material and of the rotating parts (*Tab.* **9** pag.38)
Cq = 1,5 fixed coefficient of resistance (*Tab.* **7** pag.38)
q_b = 9,9 Kg/m (utilising a belt resistance class 315 N/mm with a cover thickness 4+2 *Tab.* **10** pag.39)

Ct = 1 coefficient of passive resistance given by the temperature
(for q_{RO} - q_{RU} see *Tab.***11** pag.39)

$$q_{RO} = \frac{\text{weight of rotating parts upper troughing set}}{\text{pitch of upper sets}} = \frac{17{,}8}{1{,}2} = 14{,}8 \text{ Kg/m}$$

$$q_{RU} = \frac{\text{weight of rotating parts lower troughing set}}{\text{pitch of upper sets}} = \frac{13{,}3}{3{,}0} = 4{,}4 \text{ Kg/m}$$

$$q_G = \frac{I_V}{3{,}6 \times v} = \frac{1000}{3{,}6 \times 2{,}3} = 120{,}8 \text{ Kg/m}$$

The total tangential force Fu is given by the algebraic sum of the tangential forces Fa and Fr relative to upper and lower sections of belt for which:

$$Fu = Fa + Fr \quad [daN]$$

$$Fa = [\, L \times Cq \times f \times Ct\, (q_b + q_G + q_{RO}) + H \times (q_G + q_b)\,] \times 0{,}981 \quad [daN]$$
$$Fa = [150 \times 1{,}5 \times 0{,}017 \times 1\,(9{,}9+120{,}8+14{,}8)+15 \times (120{,}8+9{,}9)] \times 0{,}981 = 2469$$

$$Fr = [\, L \times C_q \times f \times Ct\, (q_b + q_{RU}) - (H \times q_b)\,] \times 0{,}981 \quad [daN]$$
$$Fr = [150 \times 1{,}5 \times 0{,}025 \times 1\,(9{,}9 + 4{,}4) - (15 \times 9{,}9)] \times 0{,}981 = -92$$

$$Fu = Fa + Fr = 2469 + (-92) = 2377$$

We consider an efficiency of the reduction gear and of possible transmissions as η = 0,86 will be:

$$P = \frac{Fu \times v}{100 \times \eta} \; [kW] \qquad = \frac{2377 \times 2{,}3}{100 \times 0{,}86} \cong 64 \text{ kW}$$

Tensions T1 - T2 - T3 - T0 -Tg

Let us propose to design a conveyor driven by a single driving pulley, rubber covered and positioned at the head, given that the snub pulleys are positioned to give a wrap angle of 200°; a tension device with counterweight positioned at the tail.

From *Tab.* **12** pag. 41 one may determine the wrap factor Cw = 0,42.

The tension downstream from the drive pulley is given by:

$$T_2 = Fu \times Cw \quad [daN]$$

$$T_2 = 2377 \times 0{,}42 = 998$$

The maximum tension upstream of the drive pulley will be:

$$T_1 = Fu + T_2 \quad [daN]$$

$$T_1 = 2377 + 998 = 3375$$

While the tension downstream of the return pulley is:

$$T_3 = T_2 + Fr \quad [daN]$$

$$T_3 = 998 - 92 = 906$$

To derive the maximum deflection between two consecutive carrying troughing sets equal to 2% we must apply the following formula:

$$T_0 = 6{,}25\ (q_b + q_G) \times a_0 \times 0{,}981 \quad [daN]$$

$$T_0 = 6.25 \times (120{,}8 + 9{,}9) \times 1{,}2 \times 0{,}981 = 961$$

The tension T_3 is lower than the T_0 therefore we have to provide a counterweight dimensioned to obtain the tension T_0.
We have therefore to assume T_3=T_0 and we have to recalculate consequently the tensions T_2 and T_1 that result:
T_2 = 1053 [daN]
T_1 = 3430 [daN]

One may now determine the tension "Tg" in the belt at the tension unit connection point.
The plant project data has foreseen a counterweight tension unit positioned at the conveyor tail end.
The counterweight load Tg necessary to maintain the system in equilibrium is given by:

$$Tg = 2 \times T_3 \quad [daN]$$

$$Tg = 2 \times 961 = 1922$$

Belt choice

Given the maximum working tension of the conveyor: T_1 = 3375 daN.

The unitary working tension of the belt for mm of width is given by:

$$Tu_{max} = \frac{T_{max} \times 10}{N} \quad [N/mm]$$

$$Tu_{max} = \frac{3430 \times 10}{1000} = 34{,}3 \ N/mm$$

The breaking load of the belt will correspond with the working load multiplied by a security factor "8" for belts with steel inserts and "10" for belts with textile inserts.
In our case we may proceed to choose a belt with resistance equal to 400 N/mm.

Because this belt resistance is higher than the one selected in the starting data of this calculation (315 N/mm), the belt weight is higher and we have to recalculate the T_1 and T_2 accordingly.

The resulted tensions are anyway lower than T_1 and T_2 above, therefore the following calculations will be made using

T_2 = 1053 daN
T_1 = 3430 daN

Diameter of drive pulley shaft

Let us utilise a motor gearbox to drive the conveyor in question.
Drive pulley data:

D = 400 mm diameter (as *Tab.***13**)
q_T = 220 daN weight of pulley
n = 110 r.p.m.
a_g = 0,180 m distance between the supports and pulley flange

Let us determine the resultant Cp of the tensions and the pulley weight (for simplicity let us suppose T and q_T perpendicular between them).

$$Cp = \sqrt{(T_1 + T_2)^2 + q_T^{\,2}} \quad [daN] \qquad = \sqrt{(3430 + 1053)^2 + 220^2} = 4488 \text{ daN}$$

The bending moment will be:

$$Mf = \frac{Cp}{2} \times a_g \quad [daNm] \qquad = \frac{4488}{2} \times 0{,}180 = 404 \text{ daNm}$$

The torsional moment will be:

$$Mt = \frac{P}{n} \times 954{,}9 \quad [daNm] \qquad = \frac{64}{110} \times 954{,}9 = 555{,}6 \text{ daNm}$$

One may now determine the ideal bending moment:

$$Mif = \sqrt{Mf^{\,2} + 0{,}75 \times Mt^2} \quad [daNm] \qquad = \sqrt{404^{\,2} + 0{,}75 \times 555{,}6^{\,2}} = 629 \text{ daNm}$$

Consequently we derive the value of the module of resistance W given that σ_{amm} 7,82 daN/mm² for heat treated steel C40

$$W = \frac{Mif \times 1000}{\sigma_{amm}} \quad [mm^3] \qquad = \frac{629 \times 1000}{7{,}82} = 80435 \text{ mm}^3$$

from which we may find the diameter of the drive pulley motor shaft:

$$d = \sqrt[3]{\frac{W \times 32}{\pi}} \quad mm \qquad = \sqrt[3]{\frac{80435 \times 32}{3{,}14}} \cong 93 \text{ mm}$$

The drum shaft diameter on the bearing seats, will be made according the above formula, or the nearer larger diameter available on the bearing.
The shaft diameter inside the hub and/or inside the drum (normally the raw shaft diameter) is determined with the formulas described in the paragraph "Limits of deflection and angle for motor and idler pulleys" at pag.47 and in this case the raw shaft diameter results 110 mm.

Diameter of return pulley shaft

Non-drive pulley data:

D =	315 mm	diameter (as *Tab.* **13**)
qR =	170 daN	pulley weight
ag =	0,180 m	distance between the support and pulley flange

Let us determine the resultant Cpr of the tensions and the pulley weight (for simplicity let us suppose T_3 and q_T is perpendicular between them).

$$Cpr = \sqrt{(2T_3)^2 + q_T^{\,2}} \quad [daN] \qquad = \sqrt{(2 \times 961)^2 + 170^2} = 1930 \text{ daN}$$

The bending moment will be:

$$Mf = \frac{Cpr}{2} \times ag \quad [daNm] \qquad = \frac{1930}{2} \times 0{,}180 = 174 \text{ daNm}$$

Consequently we derive the value of the module of resistance W given that σ_{amm} 7,82 daN/mm^2 for heat treated steel C40

$$W = \frac{Mif \times 1000}{\sigma_{amm}} \quad [mm^3] \qquad = \frac{174 \times 1000}{7{,}82} = 22250 \text{ mm}^3$$

from which we may find the diameter of idler return pulley shaft:

$$d = \sqrt[3]{\frac{W \text{ X } 32}{\pi}} \quad mm \qquad = \sqrt[3]{\frac{22250 \text{ X } 32}{3{,}14}} \cong 61 \text{ mm}$$

The drum shaft diameter on the bearing seats will be made according the above formula or the nearer larger diameter available on the bearing.

The shaft diameter inside the hub and/or inside the drum (normally the raw shaft diameter) is determined with the formulas described in the paragraph "Limits of deflection and angle for motor and idler pulleys" at page 47 and in this case the raw shaft diameter results 90 mm.

1 Technical Information

project and design criteria for belt conveyors

Conclusions

Using successive steps we have obtained from the data of the relative characteristics of the belt conveyor components the following summary:

- the speed of the conveyed material is v = 2,3 m/s
- carrying troughing sets with side rollers at $\lambda = 30°$
- return sets with plain roller
- belt width 1000 mm with breaking load 400 N/mm
- carrying troughing set pitch 1,2 m
- lower return sets pitch 3 m
- load roller in carrying troughing set series PSV/1-FHD, Ø 108 mm, C = 388 mm
- return rollers series PSV/1-FHD, Ø 108 mm, C = 1158 mm
- power needed to move the belt conveyor 64 kW
- belt deflection between two adjacent troughing sets < 2%
- drive pulley D = 400 mm, Ø shaft 100 mm (at the bearing seats and Ø 110 of the raw shaft in the middle)
- return pulley D = 315 mm, Ø shaft 65 mm (at the bearing seats and Ø 90 of the raw shaft in the middle)

One may consider the use of a traditional drive arrangement (drive pulley + gearbox + transmission gearing) or a motorised pulley.

In the later case, a pulley motor may be chosen using the relevant catalogue. The type TM801 of 75 kW with a shaft of 120 mm diameter meets the specification.