CZECH TECHNICAL UNIVERSITY IN PRAGUE

FACULTY OF MECHANICAL ENGINEERING

Department of designing and machine components

BACHELOR THESIS

Design of a Drive for a Belt Conveyor System

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Abstract:

The Bachelor thesis presents the way to design an effective drive for a belt conveyor aimed to transfer gravel. First it is described major objectives of conveyor application and its types. Next is the belt conveyor calculation and last is the designed gearbox to drive the conveyor.

Disclaimer:

I declare that I have my bachelor thesis developed independently and I used only the documents listed in the reference list.

In Prague on ....

Signature

Acknowledgments

I take this opportunity to express my sincere gratitude to Prof. Vaclav Vacek for the support and valuable guidance. I would like to extremely thank my supervisor Ing. Martin Janda for his excellent knowledge and sharing it with me.

Thousand times indebted to my mum for being patient and helpful and my eternal gratitude goes to my father who provided me with absolutely everything.

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1 Nomenclature

T_s ....sag tension $[N]$

g ....acceleration due to gravity $[m/s^2]$

t_d ....return idler pitch $[m]$

t_h ....carrying idler pitch $[m]$

h ....belt sag $[m]$

q_B ....mass of belt $[kg/m]$

q_G ....mass of conveyed material $[kg/m]$

L ....length of conveyor $[m]$

H ....height of conveyor $[m]$

D_b ....drum diameter $[mm]$

Q.......................transport capacity $[ton/h]$

v ....conveyor belt speed $[m/s]$

ρ ....gravel density $[ton/m^3]$

ε ....inclination of conveyor [°]

Ψ ....angle of repose [°]

β.......................angle of trough [°]

F ....cross-sectional area of load on belt $[m^2]$

B.......................width of belt $[mm]$

G_p ....mass of cords $[kg/m^2]$

s........................rubber thickness $[mm]$

k_1 ....mass of rubber $[kg/m^2.mm]$

O_H ....main resistance $[kp]$

f ....global friction coefficient $[-]$

q_{rh} ....gravitational force of run idlers $[kp/m]$

q_{rd} ....gravitational force of return idlers $[kp/m]$

f_1 ....coefficient of friction at temperature $20°C$ $[-]$

k_2 ....temperature effect on friction $[-]$

G_{rh} ....gravitational force of run roller $[kp]$

G_{rd} ....gravitational force of return roller $[kp]$

n_h ....number of rollers per run idler $[-]$

n_d ....number of rollers per return idler $[-]$

d ....diameter of roller $[mm]$

O_V ....secondary resistance $[kp]$

S_N ....hopper resistance $[kp]$

S_{OC} ....wrapping resistance $[kp]$

S_{LC} ....friction of (non-driving) pulley bearings $[kp]$

v_0 ....velocity of the feed material in the direction of transport $[m/s]$

b_n ....width of the hopper $[mm]$

l_3 ....length of center roller $[mm]$

O_P ....additional resistance $[kp]$

S_Z ....slope resistance $[kp]$

S_{VV} ....resistance of deflected carrying rollers $[kp]$

S_C ....friction from belt cleaner $[kp]$

S_{SP} ....friction from belt scraper $[kp]$

S_S ....tripper car resistance $[kp]$

S_{BV} ....friction on the chute side walls $[kp]$

n_{VV} ....number of run idler sets $[-]$

Z_C ....number of belt cleaners $[-]$

μ_1 ....friction coefficient of chute side walls $[-]$

l_b ....length of hopper side walls $[mm]$

S ....overall resistance (effective tension) $[kp]$

P ....motor power $[kW]$

η ....total mechanical efficiency $[-]$

η $_{12}$ ....efficiency of first pair of mating gears $[-]$

η $_{34}$ ....efficiency of second pair of mating gears $[-]$

η $_{belt}$ ....efficiency of V-belt transmission $[-]$

T_1 ....carry side tension $[kp]$

T_2 ....slack side tension $[kp]$

e ....base of Naperian logarithms $[-]$

μ ....coefficient of friction between belt and pulley $[-]$

α ....angle of wrap $[radians]$

M_b ....drum torque $[Nm]$

Z ....tensile force $[kp]$

T ....allowable force $[kp]$

n_b ....drum revolutions $[min^{-1}]$

n_m ....motor revolutions $[min^{-1}]$

i_c ....total speed ratio $[-]$

i_p ....gearbox speed ratio $[-]$

i_{belt} ....speed ratio of belt transmission $[-]$

u_p ....transference number of gearbox $[-]$

u_{12}....................partial transference numbers of first pair of gears $[-]$

u_{34}....................partial transference numbers of second pair of gears $[-]$

z ....number of teeth of gears $[-]$

β_{12}....................helix angle of fist pair of gears $[\,°\,]$

β_{34}....................helix angle of second pair of gears $[\,°\,]$

m ....normal module of gear [mm]

d_1, d_2, d_3, d_4 ....pitch circle diameters [mm]

d_a ....addendum diameter [mm]

d_f ....dedendum diameter [mm]

d_w.....................rolling circle diameter [mm]

x ....basic rack tooth profile displacement [$-$]

τ........................shear stress [N/mm^2]

τ_D ....allowable shear stress [N/mm^2]

W_k ....modulus in torsion [mm^3]

d_H.....................shaft diameter [mm]

S_F ....safety factor against fatigue fracture into the tooth root [$-$]

S_H ....safety factor against fatigue failure of the tooth flank surfaces [$-$]

F_t ....tangential force of gearing [N]

F_a......................axial force of gearing [N]

F_r ....radial force of gearing [N]

α ....normal profile angle [$\circ$]

F_v ....force of belt drive [N]

F_R ....radial reaction force [N]

F_A......................axial reaction force [N]

2 Introduction

Today, industries try to minimize the manual handling of material. Material handling is all about moving objects from one position to another and this is done by people, people using lifting devices such as cranes or finally, using automated mechanical machines such as conveyors.

Conveyors are just one subset of larger group of material handling equipment. The major objectives of implementing conveyors is to reduce manual handling, ease the workload of operators, accelerate work flow between operations, increase throughput, etc. Conveyors are classified into several types according to application. Some types include belt conveyor, chain conveyor, bucket elevator and gravity conveyor. In this paper, we will pay our special attention to belt conveyors used in mining industry and the power to drive it.

The idea of using the conveyor belt is not new, indeed, the first belt conveyors were introduced at the end of the nineteenth century; the basic principles of operation have not changed. However, over the years the capacity rating of belt systems and the length over which material can be transported have increased very considerably, together with the power inputs, the size of components and the degree of sophistication.

The aim is to do this transport in the most efficient way, and for this purpose, a powerful drive is always required and may be considered as the main effective part of the whole mechanism. An electric motor, a gearbox and other transmission mechanisms are parts of this drive.

First of all, we will try to introduce the conveyor system itself and how it works. For a closer view, the constitutive parts of a conveyor will be demonstrated. Common types of conveyors together with their applications in industry will be described too. The conveyor designed, is a relatively small-sized belt conveyor used in mining industry to transport gravel.

It was mentioned that the basic principles of conveyor operation have not changed over the years. In fact, what makes a conveyor preferable is its power and capacity to transport load. We will present an optimum calculation of belt conveyor according to “ISO 5048” and the minimum power to drive it. Consequently, a drive, consisting of gearbox as the main part, that satisfies this minimum power will be designed.

3 Conveyor System

A wide variety of materials can be conveyed. Mechanical conveyors are the workhorses of most bulk solids handling plants. They are used not only for the transportation of bulk solids from one location in the plant to another but also for feeding, discharging, metering and proportioning such materials to and from bulk solids storage silos, and other solids handling and processing equipment [2].

Conveyors, as with all material handling equipment, do not add value to the parts, products, or pieces that are being moved. They do not shape, form, process, or change a product in any way. They allow quick and efficient transportation of materials which make them very popular in the material handling and packaging industries.

The following is a list of some of the major objectives of implementing conveyors:

- reduce actual manual handling to a minimum
- perform all handling operations at the lowest reasonable cost
- eliminate as many manual operations as possible
- ease the workload of all operators
- improve ergonomic considerations for each operator
- improve workflow between operations
- provide routing options for intelligent workflow
- increase throughput
- carry product where it would be unsafe to do so manually [3].

Conveyor systems are versatile and mobile. They are commonly used in many industries, including the automotive, agricultural, computer, electronic, food processing, aerospace, pharmaceutical, chemical, bottling and canning, print finishing and packaging. They are versatile solutions for both surface and underground applications; hence they are widely used in mining industries too.

There are many types of mechanical conveying and elevating equipment. A convenient classification of some of the more important conveyors is as follows:

- chain conveyors/elevators.
- bucket elevators
- screw conveyors/elevators
- belt conveyors

The choice of conveyor depends largely upon the specified capacity, conveying distance and configuration (whether horizontal, vertical or inclined), bulk solids and individual particle properties, particularly bulk density, maximum particle size, abrasiveness, flowability, toxicity, corrosiveness, explosibility and temperature [4].

For example, bulk solids with a temperature above about 120˚C can not normally be transported by belt conveyors, but provided the conveying distance is below about 100m and the bulk material is not friable, then screw conveyors may be used effectively. For bulk solids with temperatures below approximately 80˚C belt conveyors have been used successfully to cover horizontal distances of over 300m by a single belt. However, belt conveyors can not be used for vertical lifts; for such duties, bucket or screw elevators are often more appropriate [2].

3.1 Chain Conveyor

A chain is a reliable machine component, which transmits power by means of tensile forces, and is used primarily for power transmission and conveyance systems (see Figure 3.1). Because of its energy absorption properties, flexibility, and ability to follow contours, is a versatile medium for lifting, towing, pulling, and securing [17].The function and uses of chain are similar to a belt. The chain converts rotational power to pulling power, or pulling power to rotational power, by engaging with the sprocket (analogous to pulleys in belt conveyor).

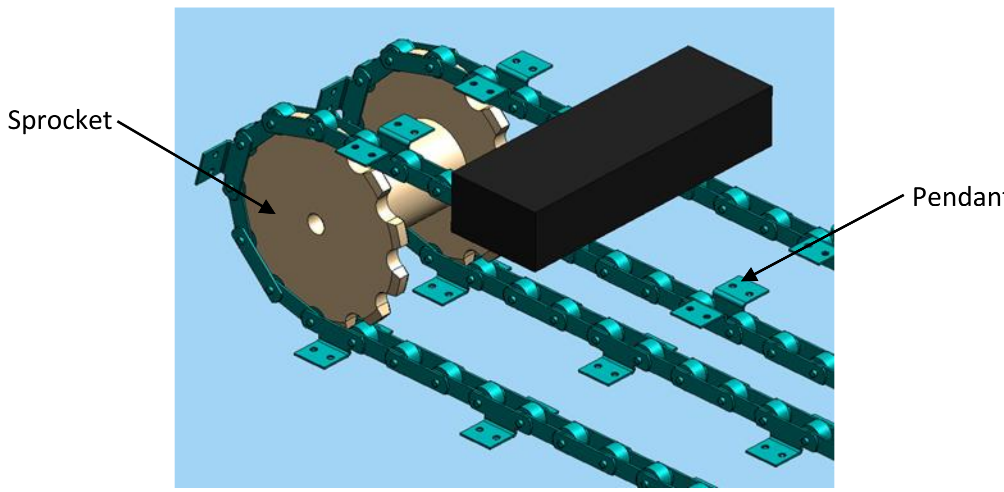


Figure 3.1 Chain engaged with sprocket [9]

Figure 3.2 shows a plain bearing which is commonly used in power transmission applications and simple chain conveyors. It consists of a series of journal bearings that are held together by constraining link plates. Each bearing consists of a pin and a bush on which the chain roller revolves. A conveyor chain is chain that has been designed specifically for chain conveyor systems; though the principle stays the same. Chain bearings in industry (e.g. automotive), where conveyors often undergo high load, are slightly different. For instance, overhead chain conveyors typically employ ball bearing rollers to bear in the conditions. An illustration can be seen in Figure 3.3.

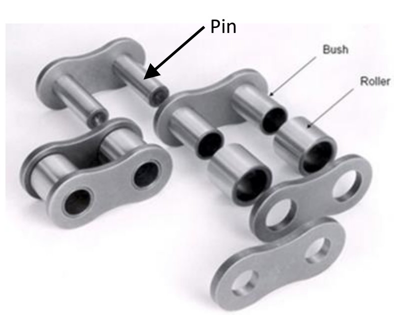


Figure 3.2 Chain bearing [8]

Chain conveyors use a continuous chain that runs from one sprocket to another at each end of a frame. Pendants or containers may be attached to the chain for product containment and transport (see Figure 3.1). One of the common type of chain conveyor is the tabletop chain conveyor as shown in Figure 3.4, which has flat plates connected to the chain.

Figure 3.3 Conveyor chain [20]

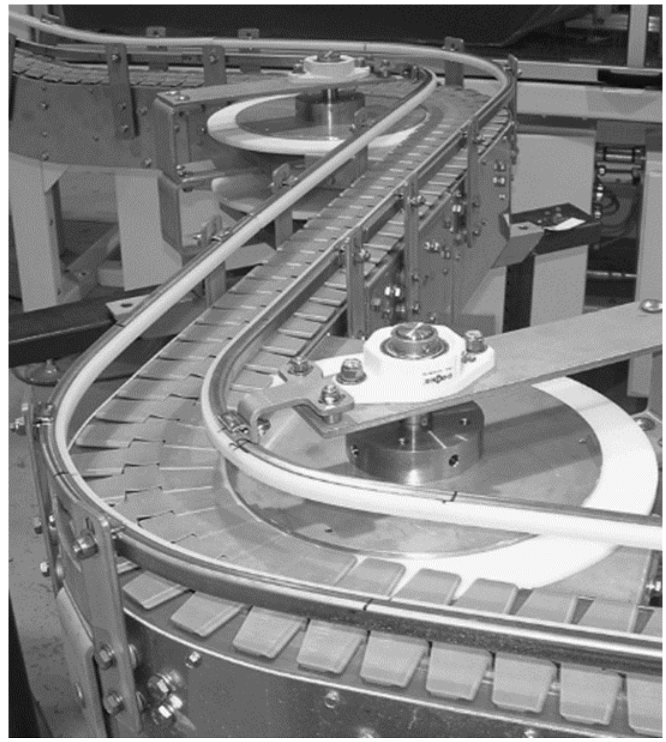


Figure 3.4 Tabletop chain conveyor [18]

Tabletop chain conveyors are placed under tension by a catenary. A catenary is typically used at the ends on the chain; this is a hanging loop that allows for an easy return of the chain on the underside of the conveyor frame.

Chain Conveyors are an essential part of many bulk handling systems, where they are used to convey bulk materials such as powders, grains, flakes and pallets. The automotive industry commonly use chain conveyor systems to convey car parts through paint plants. This type of conveyor is so called overhead chain conveyor which is shown in Figure 3.5. Many of the products that can be handled by an overhead chain system would be considered nonconveyable on a traditional conveyor. The primary design consideration in this type of conveyor is carrier design. The product stays on or in the carrier and the carrier follows the overhead track. There are two general types of tracks for overhead chain systems: open and enclosed.

The open track is typically a structural steel beam (for example, an I-beam) that uses a trolley to hang below the beam. Trolley is the component that rides in or on the track and supports the product. The enclosed track uses round or square tubing with a slot cut in the bottom to allow the carrier to ride inside (see Figure 3.6). In a power overhead chain system, the carriers are linked together by a chain. The chain is driven to make the carriers move. As shown in Figure 3.7, in an open track system, the chain is typically a heavy cast link chain that connects the trolleys. By contrast, in an enclosed track system the trolley becomes an integral part of the chain. An illustration of this can be seen in Figure 3.6.

As illustrated in Figure 3.6, unlike the heavy cast chain that connects the trolleys of an open track system, this chain has wheels mounted both vertically to support the load, and horizontally to minimize friction through the curves. The wheels can be steel or have a nylon face to minimize noise and track wear [3].

Figure 3.5 Overhead chain conveyor [22]

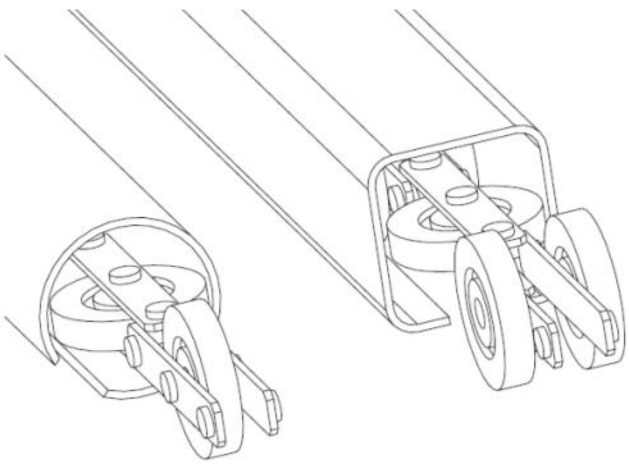


Figure 3.6 Enclosed track with trolley as an integral part of it [3]

Figure 3.7 Open track [21]

Tension in overhead chain conveyors is different than that in tabletop ones. A take-up is used to provide the tension in the system. This is because the chain stretches and the system must have means to take up the slack caused by stretching [3]. As it can be seen in Figure 3.8, take-up is a curve that is mounted in a rigid frame that allows it to move freely. It typically uses a spring or a counterweight to keep the chain tensioned properly.

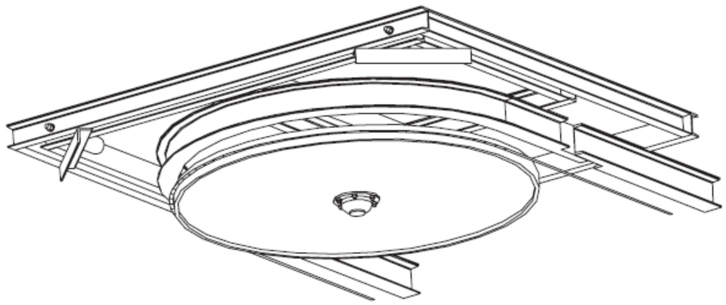


Figure 3.8 Take-up curve [3]

Chain conveyors also have widespread use in the white and brown goods, metal finishing and distribution industries. Chain conveyors are also used in the painting and coating industry, this allows for easier paint application. The products are attached to an above head chain conveyor, keeping products off of the floor allows for higher productivity levels [11].

Most chain is made of metal or engineered plastic, which may be impacted by the conditions under which the chain is operated. For example, the temperature or the amount of dust in the air can affect the chain. When heat-treated steel chains are run in temperatures higher than their tempering limits, the following problems may occur:

- Increased wear due to decreased hardness.
- Improper lubrication due to lubricant deterioration or carbonization.
- Stiff joints and increased wear due to oxide scale formation.
- Decrease in strength.

When chains are used in temperatures above 250°C, special attention should be paid to the composition and heat-treatment of the chain. The most popular type of chain for high temperature is SS specification, which is made of 304 stainless steel, and has a maximum working temperature of 650°C at low speeds. However, to maintain an adequate safety margin at a high temperature like this, use of NS-specification chain is suggested. NS chain is made of 316 stainless steel, which contains molybdenum and less carbon. NS specification has worked at low speed in environments up to 700°C [10].

When chains are used in low temperatures, the following problems may occur:

- Decrease in shock strength due to low-temperature brittleness.
- Lubricant solidification.
- Stiff joints caused by frost or ice.

Two types of chain are especially useful at lower temperatures. KT-specification chain is specially heat-treated to withstand very cold environments. SS-specification chain, which is made of 304 stainless steel, may also be used at low temperatures. Low-temperature brittleness does not occur in austenitic stainless steel. These chains cannot fix the problems of solidification of the lubricant or stiff joints caused by frost or ice. Cold-temperature oil or grease should be used and applied to the inner clearances and the outside of the chain [10].

Standard engineered plastic chain can be run at temperatures between −20°C and 80°C. At higher temperatures, it may become soft and not keep its shape; at lower temperatures it may become brittle [10].

Chain conveyors are also used for moving products down an assembly line and/or around a manufacturing or warehousing facility. These conveyors can be single or double chain strand in configuration. The load is positioned on the chains, the friction pulls the load forward [11].

3.2 Bucket Elevator

Most standard conveyors have limited capabilities for elevating bulk solids beyond a 20° incline [4] since as the inclination increases, the transfer capacity decreases. As plant layouts and other considerations often require materials to move at considerable elevations within relatively short horizontal distances, special type of equipment was developed for this purpose. This euipment is often referred to as „elevators“ or „conveyor/elevator“. Below is a general scheme of a sepical type of elevator called bucket elevator since it uses buckets to transport the load.

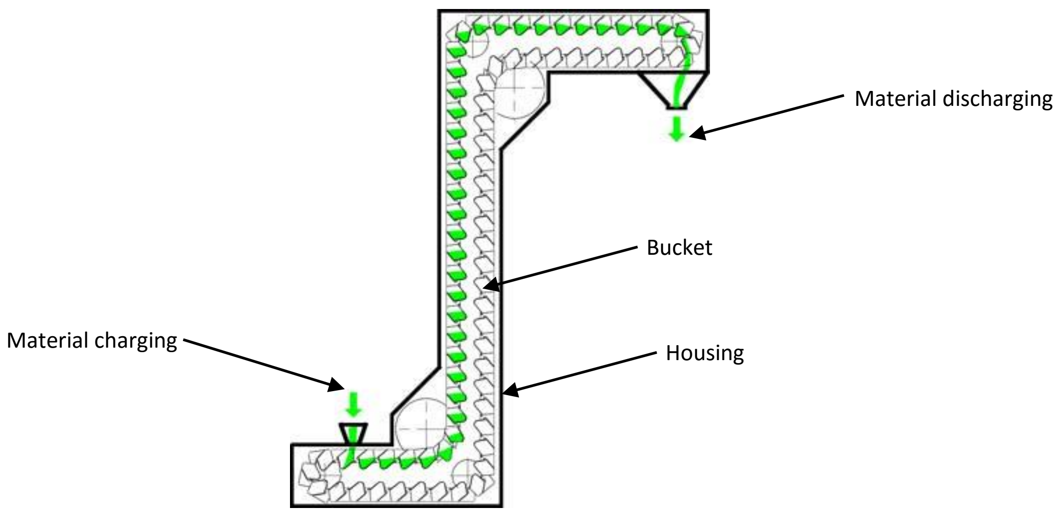


Figure 3.9 Bucket elevator [13]

Bucket elevators are designed to convey product in a primarily vertical direction. They come in a variety of sizes to handle products that range from dry, dusty powders to heavy iron ore or aggregate. They are available in a wide range of capacities (10 to 350 kg/s (4 to 140 ton/hr) [12]) and are totally enclosed to prevent dust and odors from escaping. Bucket elevators work best with dry materials, although some manufacturers have developed designs that will work acceptably with wet materials. The key to conveying wet materials is that they cannot be sticky or tend to build up or collect in the buckets [3].

Product is fed to the bucket elevator through a hopper. Buckets or cups dig into the product, convey it up, and dump it out at the discharge. Figure 3.9 shows a general scheme of a bucket elevator. The elevator bucket can be carried by chain or belt. Chain elevators typically are used for handling nonabrasive materials, whereas belt elevators will handle more abrasive material as long as there are no slivers or sharp objects that will damage the belt [3]. The main advantages of the belt-bucket elevator over the chain-bucket types are as follows:

- Higher speed and, consequently, higher capacities can be handled.
- A smoother and quieter operation is achieved.
- It has a high abrasion resistance to materials such as sand, coke breeze, glass and the like.
- It also has high resistance to many corrosive materials such as caustic soda and salt.

3.2.1 Types of Bucket Elevator

In general, bucket elevators are classified into three basic types: centrifugal discharge, continuous discharge, positive discharge. This classification is according to the way they convey and discharge material. Various types of bucket elevator are shown in Figure 3.10.

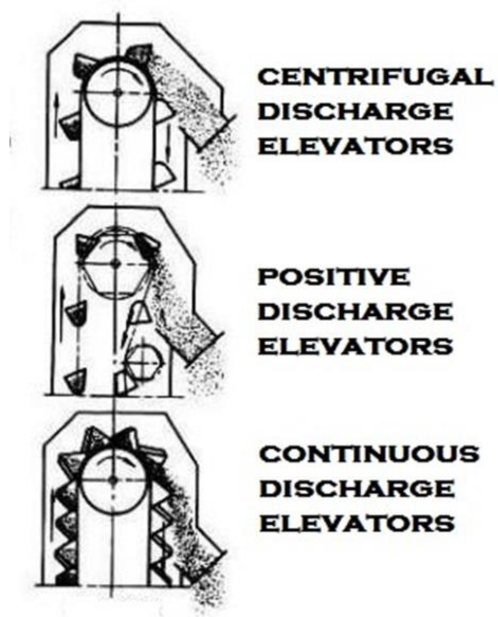


Figure 3.10 Types of bucket elevator [14]

Centrifugal Bucket Elevators: Centrifugal bucket elevators operate at higher speeds (up to 90 m/min (300 ft/min) [12]) to throw the product out of the buckets using centrifugal force as the buckets pass over the drive sprocket or pulley. To prevent interference between discharging product and adjacent buckets, the buckets are spaced a little further apart than in a continuous elevator. Therefore, they have the fewest buckets. Centrifugal bucket elevators are frequently used to move free-flowing products such as grain, sand, minerals, sugar, and aggregates.

Continuous Bucket Elevators: Continuous bucket elevators can handle the same products as a centrifugal bucket unit, but are designed to handle products that are easily damaged, sluggish, or abrasive. These elevators operate at lower speeds (38 m/min (125 ft/min) [12]), usually determined by the pitch of the chain and the wheel diameter[4], to minimize breakage of friable product. Materials that should not be aerated should be avoided [3]. The buckets are typically spaced closer together. As product is poured out, the product flows over the back of the previous bucket. Frequently the backs of the buckets have flanges on them to form a chute to guide the product to the discharge.

Positive Discharge Elevators: Positive-discharge elevators are designed for sticky materials that tend to pack. They are similar to centrifugal-discharge units, except that their buckets are mounted on two strands of chain [12], are large and closely spaced, and are snubbed back under the head sprocket to invert them for positive discharge. The buckets also move slower (35 m/min (120 ft/min) [12]) than those in centrifugal-discharge units.

3.2.2 Elevator Buckets

From an understanding of the material characteristics, or material flowability, the selection of elevator buckets can be made. In here, Buckets are classified generally into two styles: Rigid and Suspended.

Rigid Buckets: Rigid buckets are mounted to a belt or chain or can be an integral part of a plastic belt. The buckets can be constructed from steel, stainless steel, aluminum, or molded plastic, typically nylon [3]. The buckets are bolted through the flat belt using flathead screws and are evenly spaced over the length of the belt.

The buckets can be bolted to special attachment links on a roller chain. With chain-mounted buckets, it is critical that the chains be kept consistently in sync with each other. Otherwise the buckets can get twisted and this prevents them from navigating the end sprockets properly [3].

Suspended Buckets: Suspended buckets are typically mounted so that they hang freely between two chains. The suspended bucket will retain its contents until it is tipped over. This allows the elevator to have multiple discharge points [3]. This is accomplished using a mechanism that can be triggered to tip the buckets as they go by. This cannot be used as a sorter for multiple products, but can be used to balance the product load among multiple discharge points. The buckets are typically made from the same materials as the rigid buckets [3].

3.3 Screw Conveyor

A screw conveyor is a mechanism that uses a rotating screw blade called *flighting* to transport material. The screw is a helix usually wrapped around a central shaft or pipe. As the screw turns, the product is pushed along by the face of the helix. With each revolution of the screw, the product moves forward the distance of one pitch of the screw. Details of a simple screw are shown in Figure 3.11.

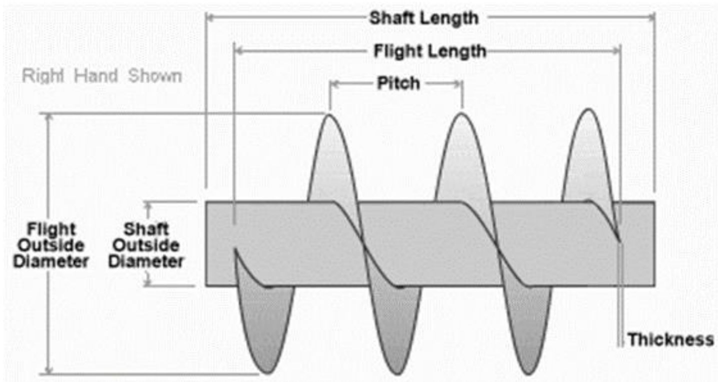


Figure 3.11 A right-hand screw

The standard pitch of the screw is equal to the flight outside diameter. There is a variety of others as well as shown below.

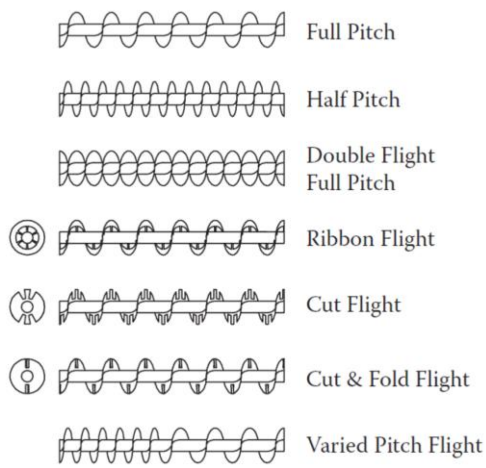


Figure 3.12 Various types of conveyor screws [3]

Standard pitch is used for conveying product horizontally or up slight inclines, *half pitch* is used in screw feeders (also for conveying material up an incline), *double flight full pitch* (two helixes on a common shaft) is used when an even discharge of product is required; typically used only as the last few pitched prior to the discharge, *ribbon flight* is used to convey sticky products or for mixing product, *cut flight* is used where mixing product is desired, *cut and fold flight* is used where high levels of mixing product and increased retention time are desired an *varied pitch flight* is used in screw feeders (short pitch below inlet, full pitch beyond to allow product loading to drop) [3].

Screw conveyors are one of the most economical options for moving dried biosolids because they can be sealed to completely contain odors and dust [12]. In industrial control applications, the device is often used as a variable rate feeder by varying the rotation rate of the shaft to deliver a measured rate or quantity of material into a process. In industry, they are often used horizontally or at a slight incline (less than 45 degrees [12]). As the angle of inclination increases, the capacity of a given unit rapidly decreases.

The screw operates either inside a pipe or a in a semicircular trough. The screw is suspended in the trough by bearings on each end and hanger bearings along the length of the conveyor. They usually are not normal ball bearings due to high probability of contamination. The bearings can be hard iron, UHMW (ultra high molecular weight polyethylene), Arguto (a wax- and oil-impregnated wood) or ceramic-coated phenolic resin [12]. For light applications such as small-diameter screws. The screw is supported by bearings on either end and then the screw rests against the side of the pipe.

This spiral blade can be either coiled around a shaft (simply called auger) or a shaftless spiral. *Shafted screw conveyors* typically are available in 3- to 4-m-long sections [12] that are bolted together. *Shaftless screw spirals* typically are furnished in one piece (either fabricated or welded) that can be up to 50 m long [12]. They rely on polyethylene liners or steel wear bars for intermediate support. Conveyor length is limited by torsional capacity of the central shaft and flighting; so, screw conveyors typically are no more than 14 m long. The torsional limits of conveyor shafts and flighting may require designers to divide long transport distances among two or more short conveyors.

Inlet and discharge openings may be located where needed. The drive for a screw conveyor is a simple arrangement that drives center shaft. The drive can be shaft-mounted or it can incorporate sprockets and chain. The drive is typically located at the discharge end of the screw conveyor. The drive can be mounted at the charge end if access for maintenance is easier at that location. Figure 3.13 shows a shafted screw conveyor.

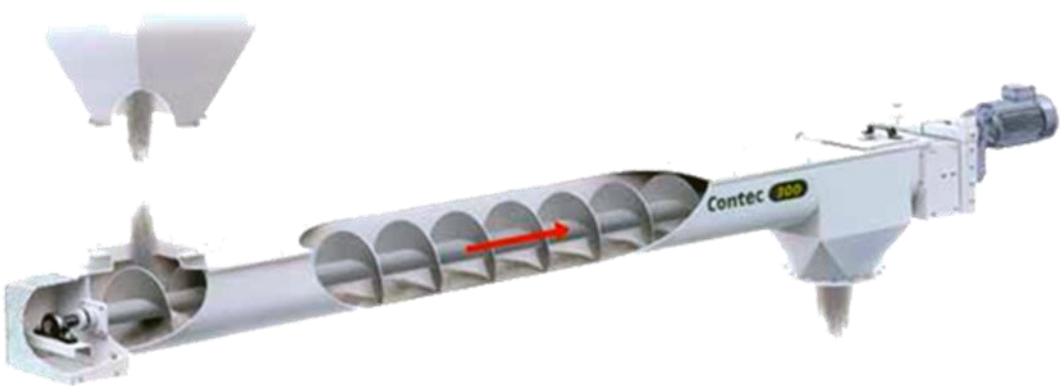


Figure 3.13 Typical shafted screw conveyor [16]

Screw conveyors require relatively little maintenance compared to other solid transport options. Routine maintenance typically consists of checking and adjusting the drive unit and greasing hanger bearings (shafted screw conveyors) or replacing polyethylene liners (shaftless ones).

3.3.1 Material of Screw

Carbon steel components are usually suitable, but galvanized steel also may be considered. Although, the dried solids eventually will wear away most of the galvanizing or paint in the interior, the areas not continuously in contact with the conveyed material should be protected against corrosion.

3.3.2 Capacity and Application

The rate of volume transfer is proportional to the rotation rate. Shafted screw conveyors transporting dried solids are typically lightly loaded (trough load less than 30%), while shaftless ones can handle loads up to 70% or 80% [12]. Also, the area of a shafted screw conveyor's trough is equal to the area of the screw spiral minus the shaft area. The area of a shaftless screw conveyor's trough is equal to that of the screw spiral. In other words, a shaftless screw conveyor has more dried solids capacity at slower rotational speeds than an identically sized one with a shaft.

If material degradation is a concern, shaftless screw conveyors are better than shafted screw conveyors because their slow turning spiral and continuous support along the trough generates less dust [12].

3.4 Belt Conveyor

The belt conveyor is a transporting machine made up of an endless wide belt which carries a load on its upper surface. It operates between a head pulley and a return or tail pulley and is supported by idlers, which in turn are supported by a frame or by steel cables.

Belt conveyors are one of the most widely used and efficient means of transporting bulk materials. They are usually the least expensive powered conveyor and are capable of handling a wide array of materials. As independent units, they are well suited for rapid transportation of loose material. They have less mobility and flexibility than trucks and scrapers, and are therefore used chiefly where large volumes of material are to be moved along one route. They are particularly applicable where the load must be lifted steeply, or carried across rough country where road construction would be difficult. They are desirable as feeders for processing plants because they provide an even and continuous flow [23].

Depending on the type chosen, belt conveyors can carry everything from loose gravel and coal to disposable cameras to flimsy bags to rigid cardboard boxes full of product. Typically, belt conveyors are used for carrying materials long distances with a single motor [3].

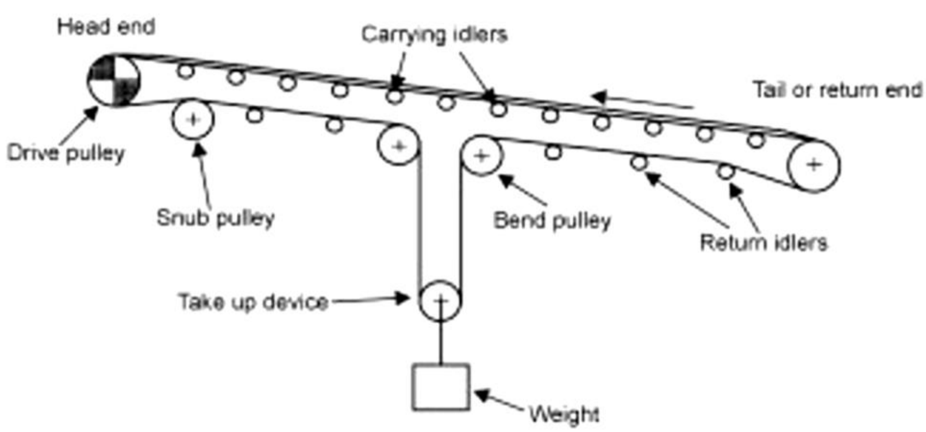


Figure 3.14 Basic belt conveyor [1]

They can vary in length from a few meters to several kilometers, but all have certain key elements in common. Figure 3.14 shows a typical basic conveyor. Essentially an endless belt moves between two points over a series of supporting rollers, propelled by a *driving pulley* or drum and returning via a second pulley.

The rollers that carry the belt are called *idlers*. On the upper, carrying strand of the conveyor, where the belt is travelling in the direction of the material being transported, there are *carrying (run) idlers* at each support point. Configuration and quantity of run rollers depends on the type of conveyed material. For instance, if the load is rigid, single roll idlers are used whereas in loose material transportation, three-roll ones are preferred. Three-roll idlers are commonly applied in mining industry to convey gravel, coal and materials alike. They are usually of the troughing type, in which a horizontal *center roll* supports the loaded part of the belt, and a pair of inclined *wing rolls* turn up the edges to create a trough cross section which keeps the load from spilling off the sides (Figure 3.20). The aim of this form of arrangement is to increase the carrying capacity of the belt [1].

On the empty lower or return strand, there is usually a single roll *return idler* at each support point. This is because there is no load transported on the lower strand. The end of the conveyor where the material is discharged is referred to as the *head end* and the other end, where the material comes onto the conveyor, the *tail* or *return end*.

Similar to chain systems, belt conveyors also need to be tensioned in order to remove slack and make the belt move. The tension is applied to the belt through a *take-up device*. The conveyor may include *snub pulleys* which is positioned right next to driving pulley so as to increase the amount by which the belt wraps around it. This increases belt contact angle which according to well-known Capstan equation (that will be explained in next chapter), leads to increase in tension forces. Conveyor may also include *bend pulleys* that alter the direction of travel of belt. There is usually minimum diameter for it; they should have larger diameters than idlers to be able to bend the belt.

Hoppers load the material onto the belt. Loading hoppers must have a restricted chute capacity, or a gate or feeding device to regulate the rate of delivery to the belt. Figure 3.15 shows a slightly elevated belt conveyor with the hopper. In some installations, increased flow is obtained by merely raising the chute, so that more material can pass under its forward edge. Various feeding devices are available to ensure even flow from the hopper to the belt. Coarse, lumpy materials, particularly when they include sharp edges, require greater precautions at the loading point than fine or soft ones do. Impact damage to the belt can be reduced by using thicker rubber on both the top and bottom surfaces, and by using closely spaced rubber-cushioned idlers at the loading point.

The installation may also include a *tripper*, which is a device mounted in either a fixed position or on a travelling carriage for discharge of material from the conveyor at a fixed point or at various points along the conveyor's length. The tripper has a discharge pulley, which is mounted above and in front of a second pulley, the purpose of which is to return the belt to its original course. For illustration, a simplified tripper car is shown in Figure 3.16.

Figure 3.15 Elevated belt conveyor [24]

Figure 3.16 Illustration of tripper car

Wet dirt and many other materials stick to belts and will build up to considerable amounts if permitted to do so. There are many designs of belt cleaner and suppliers or manufacturers offering different approaches to the removal of mineral. Device design generally falls into one of the following headings:

- scraping with rigid blades
- scraping with spring mounted or movable blades
- brushing with rotating brushes
- high pressure water or air sprays
- belt washing bath
- vibrating 'belt beating' devices.

Of these, the first three are the most commonly found [1]. The most commonly employed designs are *scraper* blades with cleaning tips made from rubber, plastic, ceramics, steel or tungsten carbide [1]. Figure 3.17 shows a scraper placed just after the drum to drop the scrapings. Rotating *cleaners* may be a serrated rubber roll or bristle brush which revolves oppositely to the belt. See Figure 3.18. Proper pressure against the belt and adjustment for wear are provided by counterweight or springs.

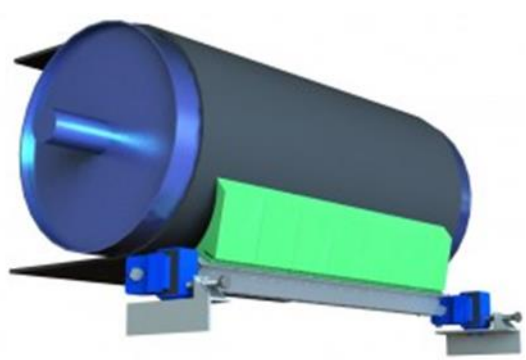


Figure 3.17 Belt scraper

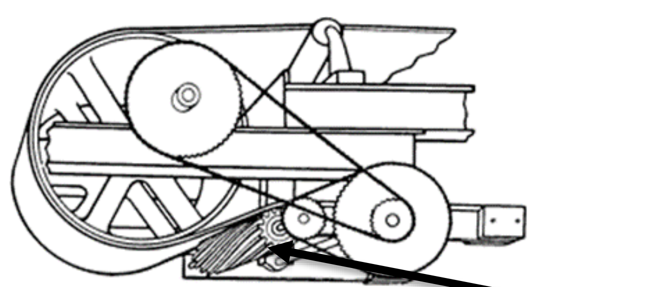


Figure 3.18 Belt cleaner [23]

3.4.1 Troughed Belt Conveyor

The main conveyor that will be discussed in this paper is the troughed belt conveyor. Their wide application in mining makes it special for our purpose. They are used almost exclusively for bulk material handling [3]. Troughed conveyors can range from a few meters long up to thousands of meters long [3]. These longer conveyors follow the terrain over which they are constructed as shown in Figure 3.19. It is not uncommon for these conveyors to turn corners and climb hills [3].

Figure 3.19 Troughed belt conveyor [3]

The idlers are mounted on a structural steel frame [3] and as stated before, are a series of three rollers. The *angle of trough* (incline of roller), which is denoted as β in Figure 3.20, is defined as the angle between the axis of the wing roller and the horizontal. The material carried is said to have an *angle of repose* (denoted as ψ in this paper) which is the angle between the tangent to the outside edge of the load where it contacts the belt and the horizontal, assuming that in cross section the load forms an arc of a circle on the belt.

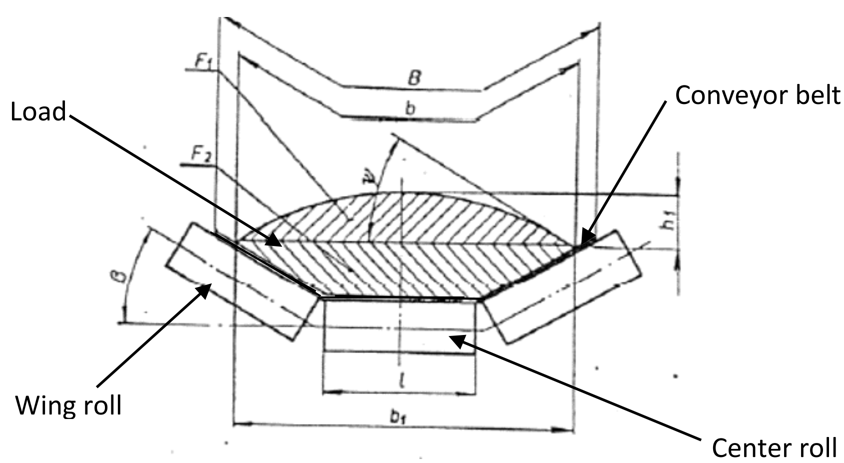


Figure 3.20 Troughed idler arrangement [5]

For a better view, cross-section of troughed belt conveyor is shown below.

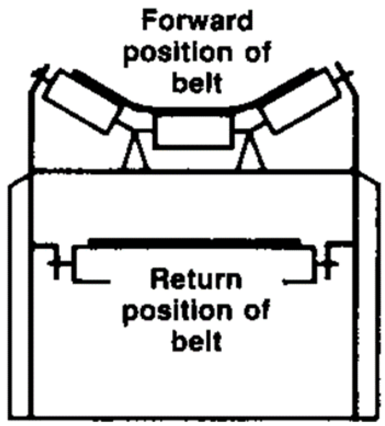


Figure 3.21 Cross-section of a troughed belt conveyor [25]

3.4.1.1 Idlers (Sets and Design)

Idlers form the supports for the carrying and return stands of the belt. For carrying idlers it was said that there were three rollers to a set, one center and two wing rollers. Five-roller sets also exist, with a horizontal center roller and two wing rollers at either side, the outer being inclined at a steeper angle to the horizontal than the inner. The economics of five-roller sets seem questionable and they are not common.

With three-roller sets the horizontal axis of the center roller may be in line with the axes of the two wing rollers which is usually called in-line idler sets or it may be offset (also called staggered idler sets). Following figure illustrates these two sets. Top is the case with in-line rollers and the lower is staggered roller sets.

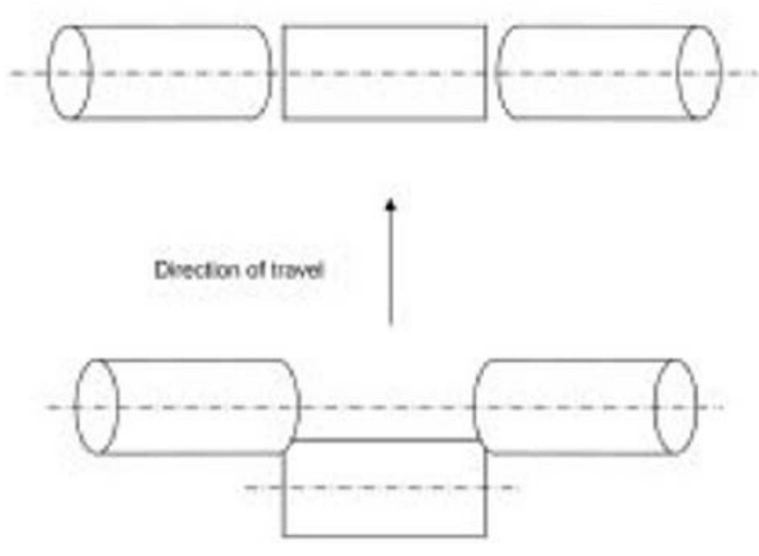


Figure 3.22 Representation of in-line (top) and staggered (bottom) idler sets [1]

Offset idler design allows the ends of the center roller to overlap those of the wing rollers and this removes the gap between the center and wing idlers that must exist if the axes of three idlers are all in line. This has the advantage of removing a potential pinch point that can nip the belt and cause damage to it [1].

These staggered sets are said to give better control of the belt by the center roller [26]. Figure 3.23 shows a real case of staggered idler set.

Figure 3.23 Staggered idler set [1]

We also defined return idlers, stating that there is usually a single flat return roller. There may be two rollers in a 'V' shape, each at an angle of up to 10° [1] to the horizontal. To aid belt tracking, the carrying wing idlers and the vee return idlers may be tilted forward by up to 3° for carrying idlers and up to 1° for vee return idlers [1]. However, every standard gives its details of geometrical arrangements of idlers and the clearances needed between rollers in a set and between idlers and supporting structure to minimize damage to the belt.

Another type of idler that is said to aid belt control and alignment is the garland or catenary idler, which may come in two, three or five roller sets. Garland idlers (Figure 3.24 and Figure 3.27) are suspended from the conveyor support framework. The advantage is that they have flexible connections between the individual rollers so that the shape of the trough formed can follow an off-center load caused by large lumps. The two-roller idlers are used on the return strand as the rigid idler sets.

Figure 3.24 Catenary idler [27]

There are various ways of supporting carrying and return idlers of all types. It is clear that the choice depends on the application but is often in the form of a steel framework. For example, the support may be suspended from the roof in coal mines.

Specialty idlers are made for certain areas of a conveyor. Some idlers are made with heavy rubber coatings [3] that are used where heavy product such as aggregate is being loaded onto the belt. They are also called impact idlers and are found under hoppers or chutes and used to absorb the shock of the falling product. They are generally designed to allow some resilience in the support of the belt to avoid belt damage. One design is shown below.

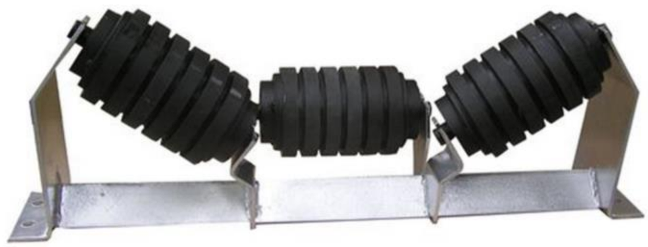


Figure 3.25 Impact idler [28]

A conveyor idler roller basically consists of a tube that turns on a shaft via rolling element bearings fitted into end plates attached to the tube. Seals are fitted to prevent the ingress of contaminants to the bearings. Figure 3.26 shows components of a typical roller.

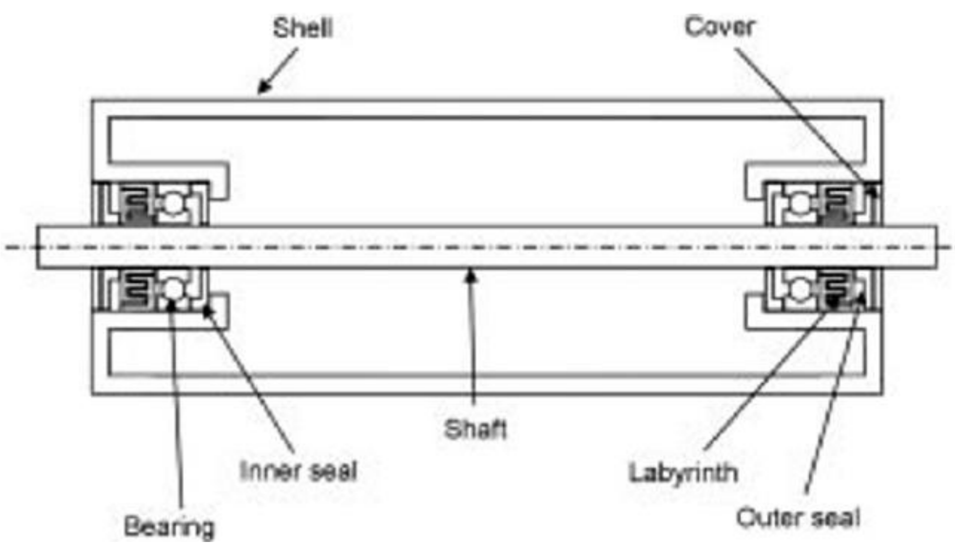


Figure 3.26 Section through a conveyor idler roller [1]

The detailed design and assembly of these components affects both the resistance of the idler to rotation (and hence the power requirement for the conveyor) and the life of the idler. Some factors that affect the resistance of the idler to rotation may be roller and bearing rating and size, type of sealing, type and quantity of lubricant, manufacturing accuracy and consistency, rotational speed and ambient temperature [29]. Resistance increases with rotational speed, and that the better the sealing arrangement, the higher the resistance to rotation, with labyrinth seals being the least resistant and face seals the most effective. British Coal in its Specification for heavy duty idlers stated that static torque should not exceed

1.2 Nm, the dynamic torque after one hour's running should not exceed $2.0 Nm$ at 680 rev/min and after one hour's standing, the dynamic torque on resumption of rotation should not exceed 3.0 Nm.

Conveyor idler bearings are subjected to forces through the belt tension, the belt weight, the weight of the material loaded onto the belt and the weight of the rotating parts of the idler. The bearing life is defined as L_{10}; this is the life at which ten percent of the bearings in that application can be expected to have failed due to classical fatigue failure. This can be calculated from a knowledge of the bearing's rated capacity (given in manufacturers' data) and the load on the bearing. For the bearing life calculations it is generally assumed that two-thirds [1] of the load on the carrying idlers is borne by the center roller.

Different standards and organizations require different minimum life. British coal, for example, states that the minimum bearing life shall be 50 000 hours. Obviously, the required life will affect both the size and type of bearing and idler selected. Plain bearings have much higher internal friction than ball bearings with consequent increases in the power required to drive the conveyor. For a given shaft or outside diameter, taper roller bearings have a higher load capacity than ball bearings.

It was said that the bearing life is defined as $L_{10;}$ this is only representing fatigue life but bearings can also fail by wear and seizure rather than fatigue because the seal has failed and dirt and moisture have penetrated into the bearing. To help cope with this, the deep groove ball bearings used in some conveyor idlers have larger balls than normal bearings of the same outside diameter, polyamide cages and internal clearance higher than normal. Considering the environment in which conveyors work, it is clear that the design and performance of idler bearing seals is a key factor in the life of conveyor idlers [29]. They are frequently exposed to abrasive minerals, sometimes to rain or pressure washing, possibly to extremes of temperature that may cause the seals to damage or dirt. Collapsed idler bearings are a significant cause of fires in underground coal mines. Some industries put much effort to make as many as possible of the components used in idlers fire resistant, including the lubricating grease used in the bearings in an attempt to limit the fire hazard on conveyors. The phosphate ester-based grease is used for this purpose but appears to be not lubricating as well as mineral oil-based one.

Belt sag is defined as the vertical displacement between idler sets. The minimum tension in the system must not only prevent belt slip, but also has to be such as to limit to an acceptable level the belt sag. The sag tension is calculated according to following formulae

$$T_s = \frac{t_d \cdot g \cdot q_B}{8(h/t_d)} \text{ [30][31]} \qquad \text{3-1}$$

for the return side and

$$T_s = \frac{t_h \cdot g \cdot (q_B + q_G)}{8(h/t_h)} \text{ [30][31]} \qquad \text{3-2}$$

for the carrying side, where t_d is the return idler pitch (m), t_h is the carrying idler pitch (m), q_B is the mass of the belt (kg/m), q_G is the mass of the material conveyed (kg/m), h is the allowable sag (m) and g is the acceleration due to gravity (m/s^2). A value of 2% of the idler pitch (spacing) is usually taken as appropriate for the belt sag. Assuming this, the carrying idlers spacing is calculated by rearranging equation 3-2:

$$t_h = \frac{0.16 \cdot T}{g \cdot (q_B + q_G)} \qquad 3\text{-}3$$

Idler pitches are chosen according to density of the material conveyed and the belt width. The recommended values again may slightly vary in different standards. Carrying idler pitches vary from $1600mm$ for densities of 400 to $1200\,kg/m^3$ with belts below $600mm$ wide, to $800mm$ for densities from 2000 to $3800\,kg/m^3$ with the largest belts (1400 to $2000mm$). Return idler pitches of around 3 meters are usual [30].

Idlers are produced in various standard diameters. Lighter duty applications typically use lighter weight rollers as small as $48.26mm$ [3] in diameter. Conversely, heavy-duty applications use larger rollers, typically up to $177.8mm$ [3] in diameter. Concentricity of the idler barrels is important, as deviations in this factor will cause vertical displacement of the belt and lead to increased noise and vibration. The length of rollers has also been standardized and is selected to be in accordance with the width of the belt used.

Steel has been the material of choice for idler barrels because of its durability and availability [1]. However, steel corrodes and in some environments this can be a life-limiting factor. If the weight of the rotating parts of an idler could be reduced then the power requirements of the conveyor would also be reduced. Polymeric materials, which resist corrosion and are of low density, have been used for idler barrels. The cost of the raw material is important, as is the cost of processing and the ability to process it into the appropriate shape. The material needs to have the right stiffness to make a barrel. There may be other disadvantages too; for example, plastics such as nylon, absorb large amounts of water and swell significantly, so are not suitable from the point of view of dimensional stability. Therefore, it may be said that none of the common polymeric materials can match steel in the absence of corrosion and when all of these design factors are taken into account the choice is limited.

3.4.1.2 Frame

The essential feature of conveyor structure is that it should provide a firm and stable basis for the mounting of pulleys and idlers, whilst allowing these items to be adjusted to allow the conveyor belt to run to its intended course without fouling either the frame itself or other objects.

Structure comprises stools and stringers. Stools are constructed from two vertical stanchions connected by a single horizontal cross tie. A plain steel square foot is attached to each stanchion leg. Brackets to mount the return idler spindles are mounted on each stanchion leg. The stringer is a tubular steel member that locates between mountings fitted to the top of each adjacent stanchion. Structure can either be placed on the ground, or in the case of tunnels or mine roadways, be suspended from the roof or wall. Top idlers are either mounted on fixed brackets at appropriate intervals on top of each stringer or as flexing units strung by end fixings between stringers (Figure 3.27). Whilst the modern breed of long distance, high power, high speed conveyors might use much stronger and heavier structures, the basic design principles of the carrying structure remain largely unchanged.

Figure 3.27 Top idlers as flexing units strung by end fixings (Garland idler set) [1]

3.4.1.3 Pulleys

A belt conveyor system consists of two or more pulleys (also referred to as drums) about which the belt rotates. Pulley design is fully covered in standards and manuals. For instance, ISO 1536 and ISO 1816 present complete guide to belt pulleys and their characteristics for belt conveyors (troughed and non-troughed). What matters in conveyor design in particular is that the pulleys should have the correct diameters and the correct coefficient of friction for the transmission of power to the belt. The size of pulley is, of course, important in terms of the belt speed since the speed increases with increase of diameter at constant revolution, but the stresses induced in the belt by its passing round the pulley also need to be considered. Figure below shows a general illustration of a drum with the necessary dimensions to identify it.

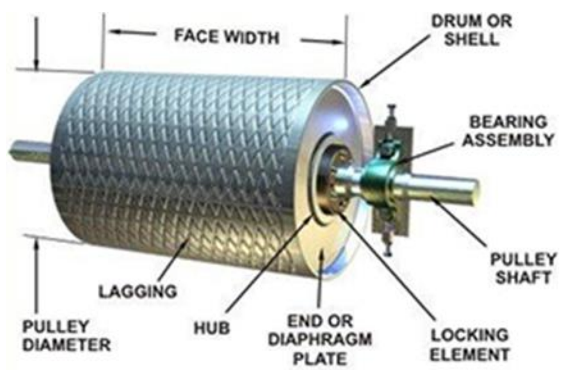


Figure 3.28 A conveyor drum [32]

Standardized ranges of pulley sizes have been introduced. ISO 3684 provides guidance on the choice of minimum pulley diameter. Recommended pulley minimum diameters vary with the anticipated maximum belt tension, the thickness of the belt, the materials from which the belt carcass is made and the percentage of the maximum allowable belt stress that is applied. The diameters are largest for those applications such as drive pulleys where the tension is highest and smallest for bend pulleys where the change of direction is less than 90°. Recommended minimum pulley face widths for standard belt widths are also provided in the standards.

One of the common type of pulleys used in modern industries, is motorized pulley. This pulley embed each part of a required drive, including the motor and speed reducer, inside. A Motorized Pulley is a compact, highly efficient conveyor drive unit that would be unaffected by dust, water, oil, grease or other harmful substances. Drum motor is space saving, quiet, efficient and reliable with virtually no maintenance. It offers a versatile, less complex and more efficient way to power belt conveyor. ISO 1816 provides basic characteristics of motorized pulleys. Some other features and advantages of drum motor include increasing operator safety, enhance space utilization, reducing noise levels, lower energy and operation costs. An example can be seen in Figure 3.29.

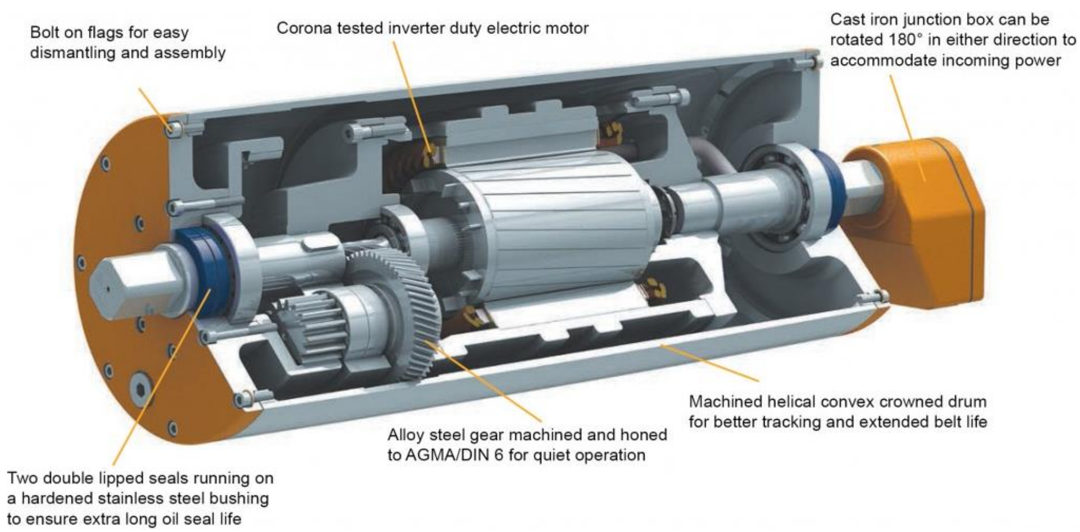


Figure 3.29 Motorized pulley [35]

In some instances so-called 'crowned' pulleys are used. The diameter at the center of the face of a crowned pulley is larger than that at the outside edges (Figure 3.30). The extent of crowning is small; it means the diameter in the center of the pulley shall not be more than 1% greater than the pulley edge diameter [30]. There are various methods of crowning; it may be in the form of the arc of a circle or of straight-sided tapers that may or may not meet in the center of the pulley. The reason of crowning is to aid the tracking of a belt that may assist in getting it to run to its intended path but it does introduce additional stresses into the belt and is therefore not usually favored.

Figure 3.30 Crowned pulley [32]

There is another type of pulley which is employed as a reliable solution for the continuous extraction of ferrous metals from a product stream, so-called pulley magnet. Magnetic pulley separators are installed as a replacement head pulley at the discharge end of a conveyor. A non-magnetic adjustable diverter plate is usually installed beneath the pulley to split the ferrous material away from the non-ferrous materials in the product stream.

The coefficient of friction between belt and pulley is important for the transmission of power, the magnitude of the belt tension and the relationship between the carry side tension and the slack side tension (T_1, T_2 see chapter 4.4). This friction coefficient can vary widely depending on the nature of the pulley surface, the type of belt and the conditions under which the conveyor is operating. Pulleys are generally made from steel but the surface of the pulley may be covered, or lagged, with rubber, polyurethane or ceramic to increase the coefficient of friction under certain conditions. The lagging may be grooved with a herringbone pattern (see Figure 3.28).

The drive pulley is typically rubber lagged to provide the traction needed to pull the belt. Tail or idler pulleys are sometimes winged or self-cleaning pulleys. Unlike the cylindrical shape that is usually thought of when talking about pulleys, the winged pulley looks like a ten- or twelve-pointed star with the points blunted. It is shown in Figure 3.31. This allows anything stuck to the bottom of the belt to navigate the pulley without damaging the belt. Frequently as the belt flexes around the pulley the debris will come loose and fall off. The center hub of the winged pulley is tapered so that any debris will fall out the ends of the pulley, thus the name self-cleaning.

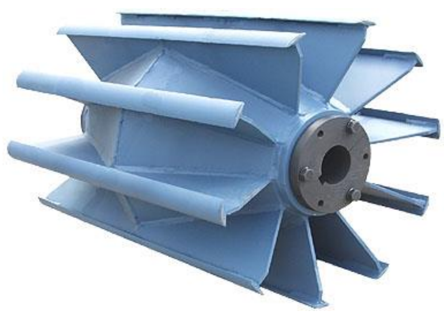


Figure 3.31 Winged pulley [33]

It is recommended to avoid using lagging underground because of the danger of generating fires through friction of the belt against the lagging in the event of belt slip or stalling. Frictional heat can be developed quickly and large volumes of acrid smoke and toxic gases given off from the heated belt. Drive drums lagged with fire resistant polymeric material give better adhesion than steel drums, but when slip occurs, smoke emission is made worse. With adequate design, installation and maintenance, the need for lagged drums is unnecessary and they should not be fitted [34].

Values of friction coefficient vary widely, from 0.05 for a bare steel pulley in wet, dirty conditions to 0.35 for a ceramic-lagged pulley in the same conditions, and to 0.45 for ceramic or rubber lagging in dry conditions [1].

3.4.1.4 Belt Construction

There are basically two elements to the conveyor belt itself; the part that is in contact with, and carries, the bulk material, and the part that transmits and withstands the tensions imposed on it. Belts are constructed so that the tension-transmitting element forms the central part, termed the carcass, of the belt, which is overlaid on both sides by covers that are made of materials with appropriate friction and wear properties. Belt construction can be seen below.

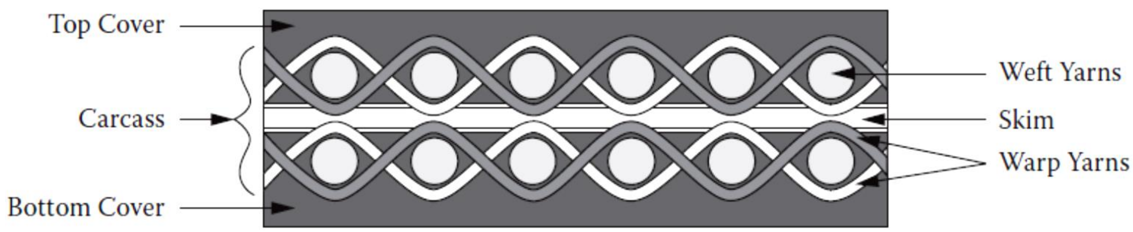


Figure 3.32 Belt construction [3]

The top cover can be made of urethane, PVC (Polyvinyl chloride), Teflon, or synthetic rubber and can come in a variety of finishes and textures. PVC is the least expensive of the three materials so it is the most popular for horizontal package conveyors [3]. The drawback is that PVC will not work for many incline conveyors because it is too hard to grip the product being conveyed. PVC is unsuitable for bulk applications where the high impact from loading requires the rubber.

The carcass is the part of the belt that actually does the bulk of the work. The carcass has four primary purposes: provide the strength required to move the loaded belt, absorb impact from material being loaded, provide load support, and provide fastener (that may be used to join lengths of belt together) holding. For those types of belt that have carcasses, the carcasses may be of various textiles or consist of steel cords.

Textile carcasses are woven from a variety of materials and are of two basic constructions, plied and solid woven. Plied belts are by far the most common type of belt used in bulk materials handling [1]. The carcass is made up of multiple layers of fabric; these layers are called plies. A layer of bonding material between the plies is called the skim. The material used for the skim is typically the same as the top or bottom cover. The skim layer is an important aspect of the belt’s strength, impact resistance, and flexibility. In special circumstances, such as when the belt needs to be fire resistant, the plies are impregnated with PVC rather than having rubber interlayers since PVC is naturally fire resistant. General plied belt construction is shown in following figure.

Figure 3.33 Plied belt construction [1]

The solid woven carcass can be made up of fabric, typically cotton or polyester nylon. The belt needs to have appropriate properties in both the longitudinal, or warp, direction and the transverse, or weft, direction. Some of these properties are achieved through the choice of materials from which the carcass is made and some through the way in which the materials of the carcass are put together. Yet others arise from the choice of cover compounds. Considering firstly the carcass, in the warp direction, the main property required is obviously strength, which arises from the number of warp yarns and for plied belts, the number of plies used, but flexibility and stretch are also important. In the weft direction, troughability, i.e., the ability to conform properly to the shape of the idler sets so that the belt contacts the idlers, is important, as is the ability to retain the mechanical fasteners that may be used. Resistance to impact damage and tearing are features that can be built into the belt carcass, features that can be of great importance operationally.

Cotton, polyamide, cotton/polyamide, cotton/polyester, polyester and Rayon are possible materials for the load carrying warp threads of textile carcass conveyor belts [30]. In plied belts, polyester (Terylene) is the most common warp thread with polyamide (Nylon) being used in the weft. In solid woven belts the warp threads are usually either polyester or polyamide, as are the weft threads that provide the transverse strength needed in the belt. With some designs of solid woven belts, additional cotton warp yarns are included on the top and bottom of the carcass to provide a 'pile' that protects the load-bearing yarns from impact and assists in cover adhesion and aids fire resistance. A three-dimensional view of solid woven belt construction is shown below.

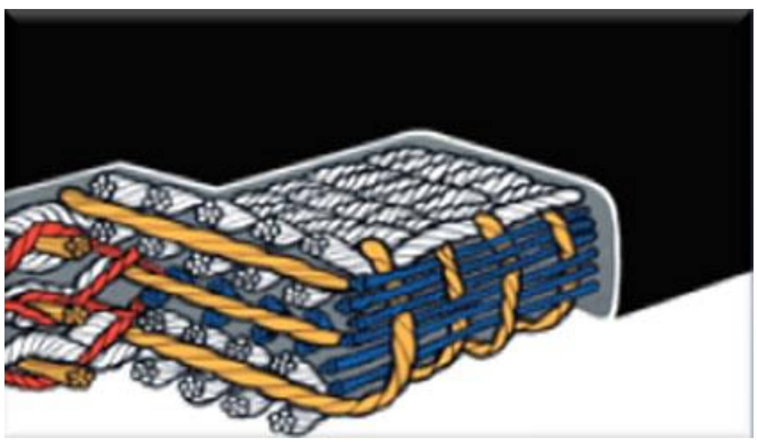


Figure 3.34 Solid woven belt construction [36]

When the belt gets longer, as with troughed belt conveyors, the fabric of the carcass can be reinforced with steel cords or wire mesh. Because of their construction, steel cord belts have high strength ratings and can be made in tensile strengths that cannot be achieved with textile belts. This can enable extremely long conveyor runs to be installed. The strength elements of steel cord belts are stranded wire ropes which are laid longitudinally as a single layer and embedded in rubber to form the core of the belt, over which the covers are applied. One area of weakness of this type of belt can be its susceptibility to damage. If a foreign object penetrates the belt cover, water can gain access to the steel cords and corrosion of the cords can seriously weaken the belt. Belts are therefore often made with the incorporation of a transverse reinforcement between the cords and the covers. This reinforcement may be a 'breaker', usually a woven fabric, or a 'weft', which is made of steel wires. The breaker or the weft may be incorporated both above and below the steel cords or either above or below them (see Figure 4.1). For steel cord belts a breaker is defined as a transverse layer lying at least one millimeter[1] and up to three millimeters [1] from the cords of the belt core; it is considered to be part of the cover of the belt. A weft is defined as lying not more than one millimeter [1] from the cords and is considered to be part of the core.

In steel cord construction, the strength of the belt is defined by the properties of the steel cords, their diameter and their pitch (the distance between adjacent cords). Clearly vast numbers of combinations of cord diameter and pitch are possible to give a very large range of possible belt strengths and properties.

The bottom cover is the third part of a belt. On the vast majority of flat conveyors, the bottom has what is called a brushed surface [3].

Depending on the application, the belt can be made with the sides sealed so that no moisture can be absorbed by an exposed edge of the carcass. This is true for all food applications, as well as most troughed belts, because of their frequent exposure to moisture.

When the belt gets longer, there may be the necessity to join lengths of belt together. The joining of textile and steel cord carcass conveyor belts is a key area of conveyor technology. No matter what properties are engineered into the belt, the joints can present a serious weakness to the performance of the system [1]. Like the belt, they have to withstand the tensions imposed during the passage round the conveyor and the differential tensions at transitions or vertical or horizontal curves and must possess the flexibility to pass around all of the system. In addition they must not damage or be damaged by belt cleaning systems, idlers, pulleys or any other part of the conveyor system. Two systems of belt joining have been developed. These have been termed mechanical and chemical systems. Mechanical fasteners join the belt ends together by some form of hardware, whereas in chemical systems the belt ends are in essence bonded together. These ‘chemical’ joints are referred to generally as ‘spliced joints’ and the terms ‘mechanical joints’ are used for those made with hardware.

4 Calculation of Belt Conveyor [3]

One of the most important features of conveyor design is the control and management of the stresses in the belt itself. Since the technology of conveyors is mature and conveyors have been used for many years, standard methods for their design exist. These allow the user to determine all of the parameters needed for the design, including calculating belt carrying capacity, belt width and speed, belt tensions and power requirements, and specifying structures, idlers and drive configurations.

The design of a conveyor is usually an iterative process, with initial estimates of belt widths, speeds and so on being revised by subsequent calculations. Several organizations including Organization for Standardization (ISO), Deutsches Institut für Normung (DIN) and the Conveyor Equipment Manufacturers' Association (CEMA) of the USA provide detailed design methods, specifications and analyses of belt conveyor. The calculations done in this paper are according to Czech standard ČSN 26 3102 which is based on ISO 5048.

The starting point for any conveyor design has to be a knowledge of what is to be moved, how far it is to be moved, the gradient over which it is moved and what constraints there might be to the design of the conveyor. As formerly stated, the variety of materials moved by conveyor is vast and therefore an extensive range of properties, some of which can affect the design of the conveyor. The particular properties that need to be considered are those that might affect the way in which the material feeds onto and off the conveyor, sits on the conveyor and travels on the conveyor. These include size and size range, angularity, abrasiveness and tendency to degrade or to produce dust and whether the material is wet or sticky [1].

The choice of belt width and speed will be influenced by the nature of the material to be conveyed, available space and the overall economics of the system. The choice of belt width may be affected by the size of the lumps of material being conveyed, since the belt needs to contain the material adequately and to avoid material being too close to the belt edge. It is also determined by other physical characteristics of the product such as angle of repose.

Some other factors that need to be considered in the design include the ability of the belt to conform properly to the trough formed by the idlers and the effect on the belt of forming the trough. For example, the thicker and stronger textile carcass belts will trough and track to a given course properly, only at low troughing angles. It is also warned that there is a possibility that troughing at 45° [1] may induce excessive stresses in the belt at the transition between the wing and center rollers and cause premature failure of the belt.

4.1 Belt Capacity

The belt capacity is derived by simple geometry from a diagram such as Figure 3.20, in which there are three rollers of equal length. All calculation methods assume that the belt is filled uniformly along its length and in accordance with ISO, it is also assumed that the material adopts a semi-circular profile (However, DIN 22101 differs from this in assuming a triangular profile and hence a rather larger capacity for a given angle of repose.) as shown in Figure 3.20.

Given the cross-sectional area (m^2), the load in tons per meter run of belt or belt capacity is then given by

$$cross - sectional\ area \text{ x } density\ ,$$

when density is measured in ton/m^3.

Tons per hour or so called transport capacity is given by:

$$tons\ per\ meter\ run \text{ x } belt\ speed \text{ x } 3600\ ,$$

when belt speed is measured in m/s.

Our aim is to design a conveyor belt transporting gravel with the conveyor parameters as follows:

- Length of conveyor $\boldsymbol{L}$: $15m$
- Height of conveyor $\boldsymbol{H}$: $3m$
- Drum diameter $\boldsymbol{D_b}$: $400mm$
- Transport capacity $\boldsymbol{Q}$: $450\frac{ton}{h}$
- Conveyor belt speed $\boldsymbol{v}$: $1.25\frac{m}{s}$
- Gravel density $\boldsymbol{\rho}$: $1.8\frac{ton}{m^3}$

The cross section of material on belt is then calculated by known transport capacity and speed:

$$F' = \frac{Q}{3600 \cdot \rho \cdot v} = \frac{450}{3600 \cdot 1.8 \cdot 1.25} = 0.056m^2 \qquad \text{4-1}$$

Angle of incline of conveyor is defined as the inverse of the ratio of its height over corresponding length:

$$\tan\varepsilon = \frac{Height}{Length} = \frac{3}{15} = 0.2 \qquad \text{4-2}$$

$$\varepsilon = 11.31°.$$

Angle of repose of dry gravel is $\psi = 14°$. For an effective and possible transport of material on the conveyor, inclination of it has to be less than the angle of repose. It means:

$$\varepsilon < \psi$$

And $11.31° < 14°$; therefore it is possible to convey gravel over the given height.

Knowing the angle of repose and theoretical area, the closest values to them on are chosen and thus, the exact area of material, troughing angle (inclination of roller) and width of the belt will be determined:

$$\beta = 30°$$

$$F = 0.059m^2$$

$$B = 800mm$$

Where F is the cross-sectional area of material on belt and B is width of the belt. Area of material found in Table 4-1 is bigger than the theoretical one; it means that transport capacity of the conveyor can become slightly bigger.

Table 4-1 Various belt widths corresponding to area of material and defined angles [5]

Teoretický průřez náplně na pásu F (m²)											
B (mm)		400*)	500*)	650*)	800	1000	1200	1400	1600	1800	2000
β (°)		20°			30°				35°		
Sypný úhel ψ (°)	15	0,012	0,019	0,034	0,059	0,096	0,141	0,196	0,279	0,356	0,442
	20	0,013	0,021	0,038	0,065	0,106	0,156	0,216	0,304	0,388	0,482
	25	0,014	0,024	0,043	0,072	0,117	0,171	0,238	0,330	0,422	0,524
	30	0,016	0,026	0,047	0,079	0,128	0,188	0,261	0,359	0,459	0,570
	35	0,018	0,029	0,052	0,087	0,141	0,206	0,286	0,391	0,499	0,620
	40	0,020	0,033	0,058	0,096	0,155	0,228	0,315	0,427	0,545	0,677

4.2 Belt Characteristics

Knowing transport capacity and the belt speed, mass of gravel per unit length is simply calculated by following formula:

$$q_G = \frac{Q}{3.6 \cdot v} = \frac{450}{3.6 \cdot 1.25} = 100\frac{kg}{m} \qquad 4\text{-}3$$

We select a proper belt from Table 4-2 and continue calculations using its specific properties. As design of conveyor is an iterative process with an initial estimate, we cannot expect initially a perfect result at the end. If the selected belt type does not fulfill the required criteria, it then will be revised and followed by new calculations. Belt type chosen from following table is **"PA 160"**.

Table 4-2 Various belt types [5]

Druh pásu	Počet nosných vložek	Hmotnost nosné kostry bez pryžových krycích vrstev G_p kg/m²	Dovolená namáhání kp/cm
PA 80	2		
	3		20
PA 100	2	3,3	16
	3	5,0	25
	4	6,6	32
PA 160	2	4,1	20
	3	6,2	32
	4	8,2	40
PA 300	2	4,8	50
	2 (+1)*)	6,1	50
	3	7,2	63
	4	9,6	80
PAK 160	4 (+3)*)	10,3	50
	5 (+3)*)	11,8	63
PAK 300	4 (+4)*)	12,0	100
	5 (+4)*)	13,4	125
PAK 700	3 (+4)*)	15,2	160
	4 (+4)*)	18,5	200
	4 (+5)*)	19,8	200

*) Vložky s vysokou příčnou pevností, označení PAN 25.

Poznámka: Uvedené hodnoty dovoleného namáhání jsou maximální a jsou závislé na průměru poháněcího bubnu. Přesnější hodnoty viz ČSN 26 0381, tab. 8.

V tabulce nejsou již uvedeny hodnoty pásů typu B 60 a B 80. Jsou to výběhové typy, které nahrazuje pás PA 80. U tohoto typu pásu budou údaje doplněny dodatečně.

Tabulka slouží jen pro informaci. Rozhodující jsou údaje uvedené ČSN, popř. závazné údaje výrobce.

Parameters of the belt from Table 4-2:

- Number of cord rows inside the belt: 2
- Mass of cords per unit area of the belt $\boldsymbol{G_p}$: $4.1\,\frac{kg}{m^2}$

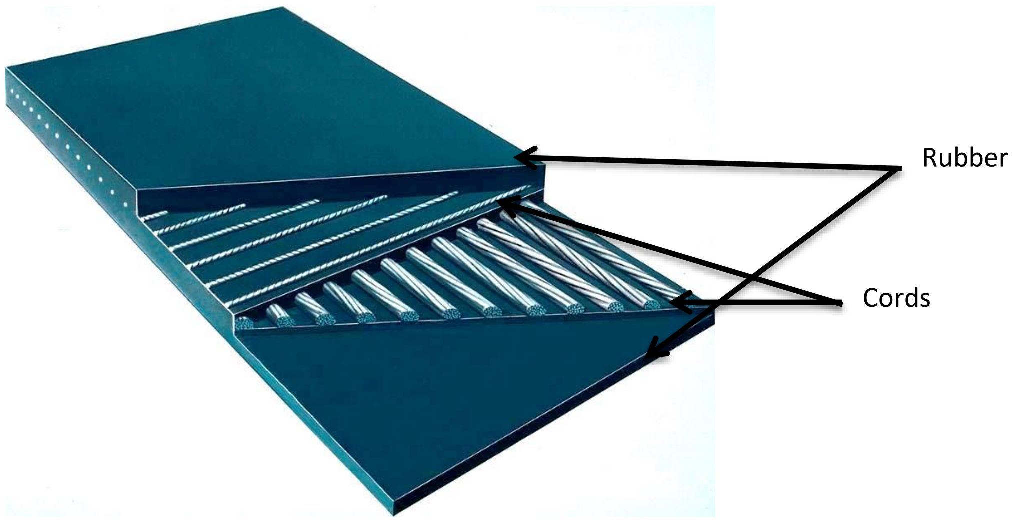


Figure 4.1 A type of conveyor belt

Below (as well as Figure 4.1) is an illustration of the conveyor belt selected for our purpose; it is composed of two rows of steel cords surrounded by rubber layers whose total thickness is s. This thickness is chosen to be $4mm$ and is distributed equally on both layers.

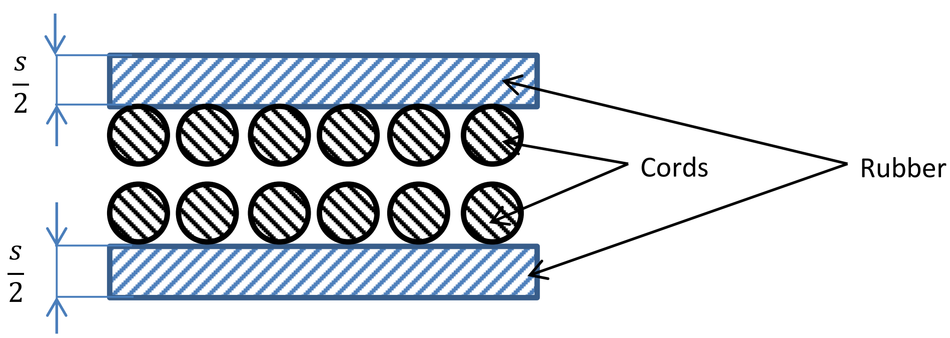


Figure 4.2 Illustration of belt layers

Hence, mass of belt having unit area, will consist of mass of cords inside and mass of $4mm$ rubber:

$$m_2{}' = (k_1 \cdot s) + G_p = (1.3 \cdot 4) + 4.1 = 9.3\,\frac{kg}{m^2} \qquad 4\text{-}4$$

Where k_1 is the mass of rubber having unit area and thickness; the standard value is $1.3\,\frac{kg}{m^2 \cdot mm}$.

Mass of belt per unit length will be:

$$q_B = \frac{B \cdot m_2'}{1000} = \frac{800 \cdot 9.3}{1000} = 7.44 \frac{kg}{m} \quad \text{4-5}$$

4.3 Power Requirements

The basic elements of a conveyor were shown in Figure 3.14. However, conveyor design can be much more complicated and include loading at various points, changes in slope, downhill sections and multiple drives. All of these and many other factors, such as the design of idlers and structure, belt characteristics and environment can affect the power requirements and belt tensions. The calculation of power and tensions can thus be a very complex process.

ISO 5048 states that it is applicable only to simple conveyors and does not cover complex conveyors that have multiple drives or undulating profiles. The conveyor designed here, is a simple relatively short belt conveyor with a single drive at its charge point. Figure 4.3 well illustrates our purpose. Consequently, ČSN 26 3102 (ISO 5048) will be a proper base for our calculations.

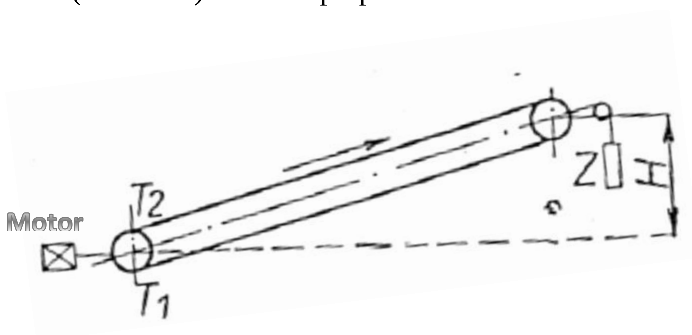


Figure 4.3 A single-drive elevated conveyor

In order to calculate required power to run the drum, it is necessary to compute the overall resistance to motion of the conveyor. This is made up of various resistances that can be divided into groups. Typically these may be: main resistance, secondary resistances and additional resistances.

4.3.1 Main Resistance

As the name implies, main resistance is the most significant resistance in a belt conveyor in terms of power consumption. The main resistance is defined as the resistance to belt travel due to the "motion alone" on idler (accounting for misalignment in installing the idlers but not accounting for forward/backward tilt of idlers). The carrying run belt travels along with material whereas the return run belt travels without material. It includes friction in bearings and seals in idlers and is due to the movement of the belt from indentation of the belt by idlers, flexing of the belt and movement of the material [1]. The following equations holds for the main resistance:

$$O_H = f \cdot L \cdot [(q_G + 2 \cdot q_B) \cdot \cos \varepsilon + q_{rh} + q_{rd}] \quad \text{4-6}$$

where f is global (also called artificial) friction coefficient, q_{rh} is gravitational force of work run rollers per unit length and q_{rd} is gravitational force of return rollers per unit length.

*** (The **kilogram-force** (**kgf** or **kg$_F$**),or **kilopond** (**kp)**, is a gravitational metric unit of force. It is equal to the magnitude of the force exerted by one kilogram of mass in a 9.80665 m/s^2 gravitational field. Therefore one kilogram-force is by definition equal to 9.80665N) [7].

The global friction coefficient is defined as:

$$f = f_1 \cdot k_2 \tag{4-7}$$

Where f_1 is friction coefficient at temperature $20°C$ and k_2 is the temperature effect on friction. f_1 is assumed 0.02 for ordinary conveyors under good operating conditions (such as dust-free working environment and good maintenance). The temperature of the environment in which the conveyor is going to operate, is assumed to be $-5°C$. The temperature effect factor is determined from Figure 4.4 and equals 1.1 .

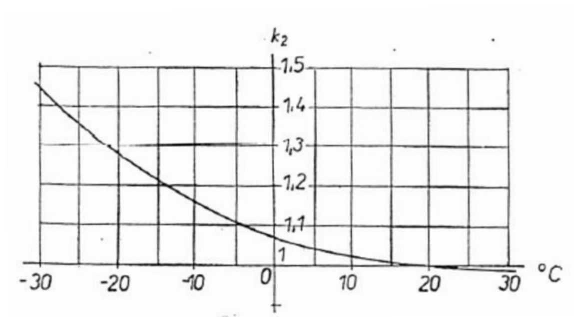


Figure 4.4 Temperature effect factor [5]

Then, global friction coefficient is calculated:

$$f = 0.02 \cdot 1.1 = 0.022\ .$$

For calculation of idlers, the following equations are given:

$$q_{rh} = \frac{G_{rh} \cdot n_h}{t_h} \tag{4-8}$$

for run idlers and

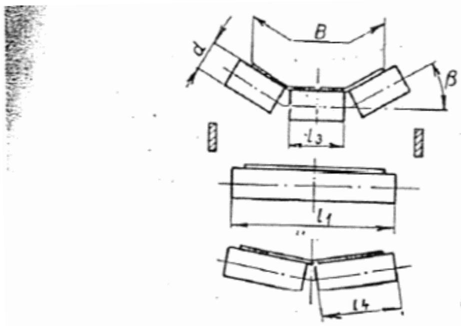
$$q_{rd} = \frac{G_{rd} \cdot n_d}{t_d} \tag{4-9}$$

for return idlers; where G_{rh} is gravitational force of run roller, n_h is the number of rollers per run idler, G_{rd} is gravitational force of return roller and n_d is the number of rollers per return idler.

As the conveyor is of troughed type, there will be three-roll run idler and single-roll return idler sets. With the previously explained criteria in optimum selection of idler pitch, the run idlers' spacing is chosen to be $800mm$ (same as belt width). Being no material transported by return idlers, implies greater interstitial distance and thus is assumed $2300mm$. With the known belt width, following parameters may be determined from Table 4-3:

- $G_{rh} = 3.4kp$
- $G_{rd} = 7.5kp$
- Diameter of rollers **d**: $108mm$.

Table 4-3 Characteristics of conveyor rollers of various belt widths [5]



Obr. 6

Tab. 10

B	d	l_1	G_r	l_2	G_r	l_3	G_r	l_4	G_r	β°
400	63	500	2,6	250	1,7	—	—	—	—	20°
	89		3,5		2,1					
500	63	600	3,0	315	1,9					
	89		4,0		2,5					
650	63	750	3,5	380	2,2					
	89		4,8		2,8					
	108				3,8					
800	80	950	5,9	—	—	315		465		30°
	108		7,5				3,4		4,4	
	133						4,6		6,1	
	133*)		13,3				6,2		7,9	
1000	108	1150	8.8	—	—	380	3,8	600	5,2	
	133		12,6				5,3		7,4	
	133*)		15,5				6,9		9,3	
1200	108	1400	10,5	—	—	465	4,4	670	5,7	
	133		15,0				6,1		8,0	
	133*)		18,3				7,9		10,2	
(1400)	108	1600		—	—	530		750	6,2	
	133		16,0				6,7		8,8	
	133*)		20,5				8,6		11,0	
	159		33,1							
1600	133	1800	—	—	—	600	7,4	900		35°
	133*)						9,3			
	159*)						15,5		20,6	
(1800)	159*)	—	—	—	—	670	16,6	1000	22,3	
2000	194*)					750	22,5	1150	33,9	

*) Hodnoty G_r u d *) platí pro válečky transportétů pro dálkovou dopravu.
Poznámka: Hodnoty G_r uvedené v tabulce platí pro válečky vyrobené o. p. Transporta, závod Břeclav; pro jiné výrobky platí údaje výrobce. Hodnoty G_r pro válečky dopadové a dolní diskové nejsou známy; je možné použít hodnot podle tabulky.

Therefore:

$$q_{rh} = \frac{3.4 \cdot 3}{0.8} = 12.75 \frac{kp}{m}$$

for carrying idlers and

$$q_{rd} = \frac{7.5 \cdot 1}{2.3} = 3.26 \frac{kp}{m}$$

for return idlers.

Eventually, main resistance to the conveyor motion is determined by equation 4-6 :

$$O_H = 0.022 \cdot 15 \cdot [(100 + 2 \cdot 7.44) \cdot \cos 11.31 + 12.75 + 3.26] = 42.46kp = 416.51N \, .$$

If angle of conveyor is less than 18° [5], the factor '$\cos \varepsilon$' is usually neglected due to its small effect.

4.3.2 Secondary resistances

These resistances are less prior than main resistance and are due to necessary actions and accessories required for a conveyor. Secondary resistances include inertial and frictional resistances due to the acceleration of the load in the loading area which generally is called hopper resistance here, friction of pulley bearings (not driving pulleys) and resistance due to wrapping of the belt round pulleys. Therefore, the total secondary resistance is sum of the mentioned ones:

$$O_V = S_N + S_{OC} + S_{LC} \qquad \text{4-10}$$

where S_N is hopper resistance, S_{OC} is the resistance due to wrapping of the belt round the pulleys and S_{LC} is friction of non-driving pulley bearings.

Hopper resistance is defined as:

$$S_N = \frac{q_G \cdot v}{g} \cdot (v - v_0)\left[1 + \frac{1000 \cdot q_G}{\rho \cdot {b_n}^2} \cdot \left(1 + \frac{v_0}{v}\right)\right] \qquad \text{4-11}$$

where g is the acceleration due to gravity, v_0 is velocity of the feed material in the direction of transport and b_n is width of hopper.

v_0 is assumed to be $0\frac{m}{s}$. Length of the center roller, l_3, equals $315mm$ from Table 4-3. Width of the hopper is typically larger than this length, thus it is chosen:

$$b_n = 480mm\,.$$

Hopper resistance is determined from equation 4-11:

$$S_N = \frac{100 \cdot 1.25}{9.81} \cdot 1.25\left[1 + \frac{1000 \cdot 100}{1.8 \cdot 480^2}\right] = 19.77kp = 193.93N\,.$$

Wrapping resistance for rubber and PVC belt is $15kp$ per drum. Having a conveyor with two drums (one driving and the other non-driving), total wrapping resistance will be:

$$S_{OC} = 2 \cdot 15 = 30kp = 294.3N\,.$$

Friction of bearings is $5kp$ per non-driving drum. The total resistance of bearings is:

$$S_{LC} = 1 \cdot 5 = 5kp = 49.05N\,.$$

Back to equation 4-10, the resulting secondary resistance will be:

$$O_V = 19.77 + 30 + 5 = 54.77kp = 537.29N\,.$$

4.3.3 Additional Resistances

These resistances cover, for example, additional accessories of the conveyor. Special secondary resistances are slope resistance, resistance of misalignment of carrying steel rollers, resistance due to trippers, resistance due to discharge ploughs, friction on the side walls of chutes in the loading area and friction from belt and pulley cleaners. Therefore, the total additional resistance:

$$O_P = S_Z + S_{VV} + S_C + S_{SP} + S_S + S_{BV} \qquad \text{4-12}$$

where S_Z is resistance due to slope, S_{VV} is resistance of deviated carrying steel rollers, S_C is friction from belt cleaner, S_{SP} is friction from belt scraper, S_S is tripper car resistance and S_{BV} is friction on hopper side walls.

The slope resistance is due to lifting or lowering the load on inclined conveyors. It is generally defined as:

$$S_Z = \pm q_G \cdot H \tag{4-13}$$

Where the positive sign presents an upward transport and negative sign presents a declined slope. As the conveyor designed is an elevated one (see Figure 4.3), positive sign will be taken:

$$S_Z = +q_G \cdot H = 100 \cdot 3 = 300kp = 2943N\ .$$

The resistance of deflection is calculated for carrying rollers, since there is no material on return strand:

$$S_{VV} = 0.004 \cdot n_{VV} \cdot (q_G + q_B) \cdot \cos\varepsilon \tag{4-14}$$

where n_{VV} is the total number of run idler sets and logically may be calculated as length of conveyor divided by the idler pitch:

$$n_{VV} = \frac{L}{t_h} = \frac{15}{0.8} = 18.75 \tag{4-15}$$

it is rounded to an integer and the resistance will be calculated applying equation 4-14:

$$S_{VV} = 0.004 \cdot 18 \cdot (100 + 7.44) \cdot \cos 11.31 = 7.58kp = 74.41N\ .$$

The conveyor is aimed for relatively short load transport and thus, a rotating cleaner will be employed with no need of a belt scraper. The below equation is given for calculation of cleaner friction:

$$S_C = z_c \cdot 0.03 \cdot B \tag{4-16}$$

z_c is the number of belt cleaners. Hence:

$$S_C = 0.03 \cdot 800 = 24kp = 235.44N\ .$$

The conveyor has only one discharge point and there will be no tripper applied:

$$S_S = 0\ .$$

As the load passes the chutes, there will be friction with the walls and this is calculated as following:

$$S_{BV} = \mu_1 \cdot \frac{q_G^2 \cdot l_b}{\rho \cdot b_n^2} \tag{4-17}$$

where μ_1 is friction coefficient of walls and is usually equal to 0.7 and l_b is the length of side walls.

The hopper employed for our application, will have walls with length of $500mm$ and therefore:

$$S_{BV} = 0.7 \cdot \frac{100^2 \cdot 500}{1.8 \cdot 480^2} = 8.44kp = 82.79N\ .$$

The additional resistance will be the sum of above calculated resistances:

$$O_P = 300 + 7.58 + 24 + 8.44 = 340.02kp = 3335.6N\ .$$

The overall resistance then becomes:

$$S = O_H + O_V + O_P \tag{4-18}$$

$$S = 416.51 + 537.29 + 3335.6 = 437.25kp = 4289.4N\ .$$

This resultant motion resistance is basically the circumferential force of the driving drum. The minimum motor power required to move the conveyor is then given by:

$$P = \frac{S \cdot v}{102 \cdot \eta} \tag{4-19}$$

where P is drum power in (kW), η is the efficiency of transmission including efficiency of speed reducer (gearbox), bearings and belt transmission:

$$\eta = \eta_{12} \cdot \eta_{34} \cdot \eta_{belt} \tag{4-20}$$

The reducer used, is a two-stage spur gearbox transmitting the power from motor by means of a V-belt transmission. η_{12} is the efficiency of first pair of mating gears, η_{34} is the efficiency of second pair and η_{belt} is the efficiency related to V-belt. Efficiency of the pairs of gears (including bearings) is assumed 98% and 92% for that of V-belt:

$$\eta = \eta_{12} \cdot \eta_{34} \cdot \eta_{belt} = 0.98 \cdot 0.98 \cdot 0.92 = 0.88\ .$$

The minimum power supplied to drum is calculated using equation 4-19:

$$P_b = \frac{437.25 \cdot 1.25}{102 \cdot 0.88} \cong 6kW\ .$$

4.4 Belt Tensions

Having determined the power required to move the conveyor and the belt speed, it is then possible to calculate the tensions in the belt. The magnitude of the tension is needed in order to make sure that a correct type of belt has been selected for the duty.

The overall resistance S is the effective tension at the pulley and by definition:

$$S = T_1 - T_2 \tag{4-21}$$

where T_1 is the carry side tension and T_2 is the slack side tension. Using the additional relationship used to model belt friction also called Capstan equation

$$\frac{T_1}{T_2} = e^{\mu\alpha} \tag{4-22}$$

where e is the base of Naperian logarithms, μ is the coefficient of friction between the belt and the pulley and α is the angle of wrap of the belt around the pulley in radians, values for T_1 and T_2 can be calculated. Relationship 4-22 represents the limiting tension ratio when the belt is about to slip. The coefficient of friction, the contact angle and the drive arrangement can all affect the belt tensions.

There are many possibilities of drive arrangement some of which are very complex. The arrangement used for our duty, is a plain drive which is shown in Figure 4.5.

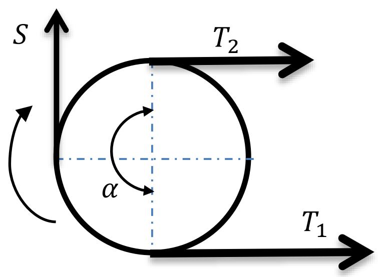


Figure 4.5 Plain drive

T_1 and T_2 can be calculated by simple substitution. Since

$$S = T_1 - T_2 = T_2 \cdot (e^{\mu\alpha} - 1)$$

then

$$T_1 = S \cdot \left[1 + \frac{1}{e^{\mu\alpha} - 1}\right] \qquad 4\text{-}23$$

and

$$T_2 = S \cdot \frac{1}{e^{\mu\alpha} - 1} \qquad 4\text{-}24$$

In some design standards and guides the quantity $1/e^{\mu\alpha} - 1$ is defined and referred to as the 'drive factor'. Since the power required is fixed, and this fixes the value of the effective tension S, if the speed is also fixed, decreasing the value of the drive factor decreases the value of T_1. This may allow a lighter, less expensive belt to be used and reduce the amount of wear in the system. The drive factor is dependent upon the coefficient of friction μ between the belt and the drive pulley, and on the angle of wrap α of the belt around the pulley. Increasing either or both of these quantities will result in a lower value of T_1.

Use of a snub pulley helps increase the angle of wrap and consequently decrease in T_1. Although, in practice there are many possibilities of complex drive configurations in order to decrease the drive factor based on the increment in contact angle. An example is Figure 4.6.

Friction factor from Table 4-4 is approximately 0.3 for a bare steel drum assuming clean and dry environment. Belt tensions are calculated

$$T_1 = 716.4kp = 7027.94N\,, T_2 = 279.15kp = 2738.51N\,.$$

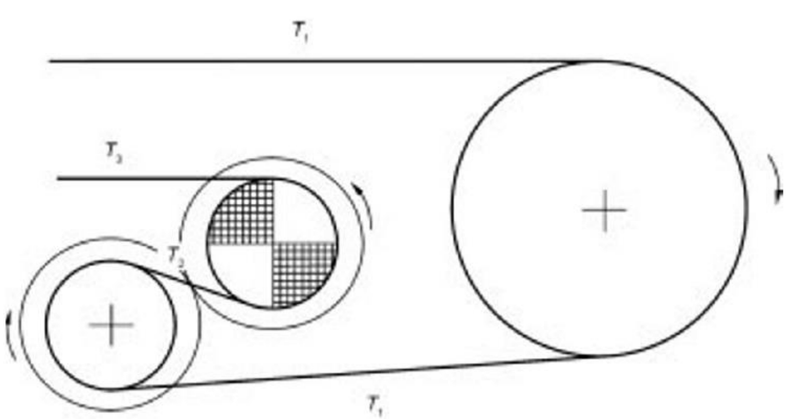


Figure 4.6 Dual-drum geared motor

Table 4-4 Coefficient of friction μ for rubber belts

Plášť bubnu	ocelový hladký					s obložením pryžovým hladkým					s obložením pryžovým drážkovaným				
Tlak na buben p (kp/cm²)	0,5	0,75	1,0	1,25	1,5	0,5	0,75	1,0	1,25	1,5	0,5	0,75	1,0	1,25	1,5
Povrch bubnu	Součinitel μ														
čistý a suchý	0,402	0,389	0,370	0,358	0,345	0,762	0,742	0,729	0,716	0,707	0,889	0,858	0,830	0,815	0,802
vlhký až mokrý	0,109—0,280	0,100—0,226	0,086—0,179	0,073—0,139	0,062—0,100	0,200—0,402	0,167—0,358	0,144—0,330	0,114—0,296	0,096—0,281	0,506—0,714	0,408—0,660	0,333—0,630	0,253—0,600	0,200—0,568
znečištěný jílem vlhkým až mokrým	0,027—0,093	0,024—0,077	0,022—0,064	0,021—0,056	0,020—0,051	0,080—0,137	0,062—0,100	0,043—0,090	0,036—0,074	0,025—0,062	0,272—0,420	0,214—0,333	0,167—0,259	0,124—0,203	0,086—0,154

Plášť bubnu	s brzdovým obložením					s keramickým obložením									
Povrch bubnu	Součinitel μ														
čistý a suchý	0,744	0,738	0,735	0,733	0,730	1,080	1,047	1,031	1,020	1,010					
vlhký až mokrý	0,426—0,563	0,388—0,538	0,372—0,512	0,355—0,462	0,338—0,363	0,698—0,784	0,680—0,766	0,660—0,747	0,654—0,735	0,642—0,722					
znečištěný jílem vlhkým až mokrým	0,144—0,163	0,142—0,162	0,140—0,160	0,138—0,158	0,138—0,156	0,300—0,400	0,284—0,370	0,265—0,352	0,259—0,340	0,247—0,328					

Torque of drum M_b is the effective tension multiplied by its radius

$$S \cdot \frac{D_b}{2} = 4289.4 \cdot 0.2 = 857.88Nm \quad 4\text{-}25$$

Following equations are for calculation of tensile force of single-drive (at charge end) elevated belt conveyor as Figure 4.3

$$T_1 = \frac{Z}{2} - q_B \cdot H \quad 4\text{-}26$$

$$T_2 = \frac{Z}{2} - S - q_B \cdot H \quad 4\text{-}27$$

where Z is tensile force (kp) and is calculated by simple substitution:

$$Z = 2 \cdot (T_1 + q_B \cdot H) = 1477.44kp = 14493.69N\ .$$

For safety, tensile force is usually increased by 10%:

$$Z = 1.1 \cdot 1477.44 = 1625.184kp = 15943.05N\ ,$$

therefore the corresponding tensions T_1 and T_2 will be revised

$$T_1 = 790.272kp = 7752.57N\ , T_2 = 353.022kp = 3463.15N\ .$$

4.4.1 Strength Check

The allowable stress $\left(\frac{kp}{cm}\right)$ for the selected belt is given in Table 4-2 and for a known belt width, the allowable force is calculated

$$T = 20 \cdot 80 = 1600kp = 15696N \tag{4-28}$$

It can be seen that the maximum tension in the system T_1 is less than the allowable force. Hence, the strength criterion is fulfilled and the selected belt meets all the requirements.

4.5 Transition Distance (Tension changes over the belt width)

The belt must pass from a troughed shape to a flat one and vice versa on its journey round the conveyor. This is important as they cause differential tensions to arise in the belt and must be carefully controlled. The change from troughed to flat and vice versa is referred to as the transition and the distance over which it takes place is the transition distance.

Consideration of the basic geometry of the transition under steady-state running reveals that there must be a difference in the distance travelled by the center and the edges of the belt. This causes a difference in the tensions in these two locations. The transition distance needs to be such that the edge tension does not rise so high that the elastic limit of the belt is exceeded and damage is caused to the belt edges, or fall below zero at the center and so cause buckling of the belt. For non-steady state conditions, such as starting and stopping, the edge stresses will be higher still [1].

Different standards such as ISO and DIN provide methods and formulae for the calculation of the belt edge tension and the minimum transition distance for a given trough cross-section and position of the terminal pulley relative to center roll (of idlers). They show the stress distribution across a belt at the transition and give methods of calculation for the maximum edge tensions and the tension at the center of the belt for the avoidance of buckling at the belt center.

ISO 5293 indicates that the maximum allowable edge stress should be agreed with the belt manufacturer, but gives guidance that for textile belts it may be 180% of the maximum running tension and 200% for steel cord belts. The transition distance may be reduced if the pulley is raised relative to the center roller because this reduces the edge stress, but should be lengthened if the pulley is lowered because this raises the stress in the belt edges.

5 Design of Drive

There is variety of options available for running conveying systems, including the hydraulic, mechanical and fully automated systems, which are equipped to fit individual needs. All of the powered conveyors discussed in this paper, use electric motors to drive them, to pull the belt, or turn the rollers. As briefly seen in previous chapter, the arrangement of pulleys in conveyor drives can vary enormously in complexity from the single pulley to systems with several driving pulleys. Despite this variability, all drives must have at least one prime mover, which is an electric motor and a speed reducer to convert the motor shaft speed to the appropriate pulley shaft speed. Other components may be present to give control of the starting, running and stopping characteristics of the conveyor.

Every type of conveyor and every manufacturer have specific drive configuration choices available depending on the application. Gear Reducers come in a wide variety of shapes and sizes, but they all have a few things in common. They all have an input and output shaft. Finally, they all have two or more gears inside the mesh together to reduce the speed from the input shaft to the output shaft.

The drive configuration used in here comprises a V-belt transmission in order to transfer the power from motor to the gearbox, a two-stage spur helical gearbox and a shaft coupling connecting the output shaft of the gearbox and the shaft of drum. This is simply seen in figure below.

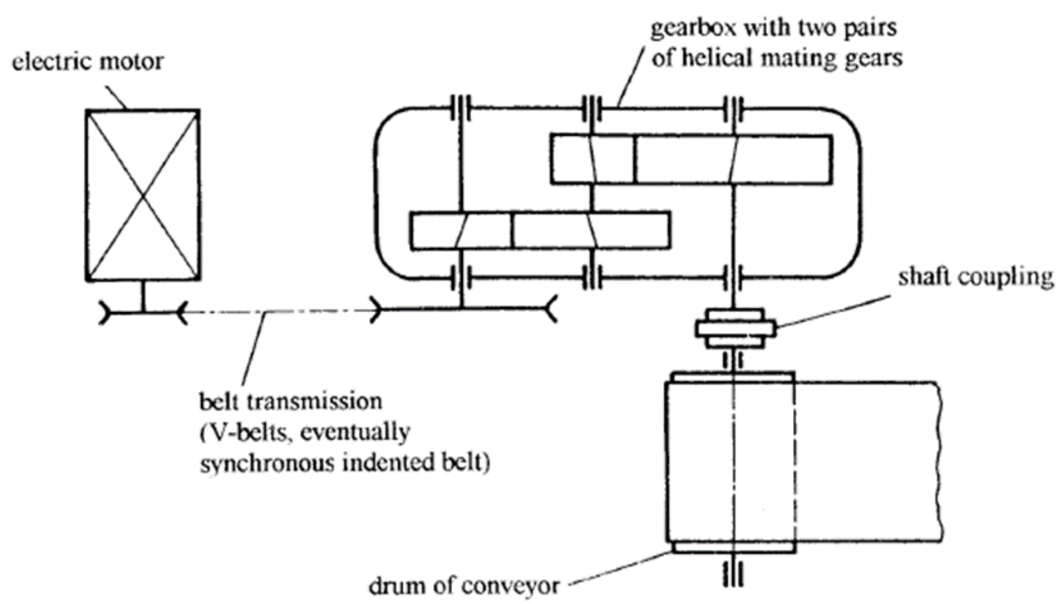


Figure 5.1 Conveyor drive configuration

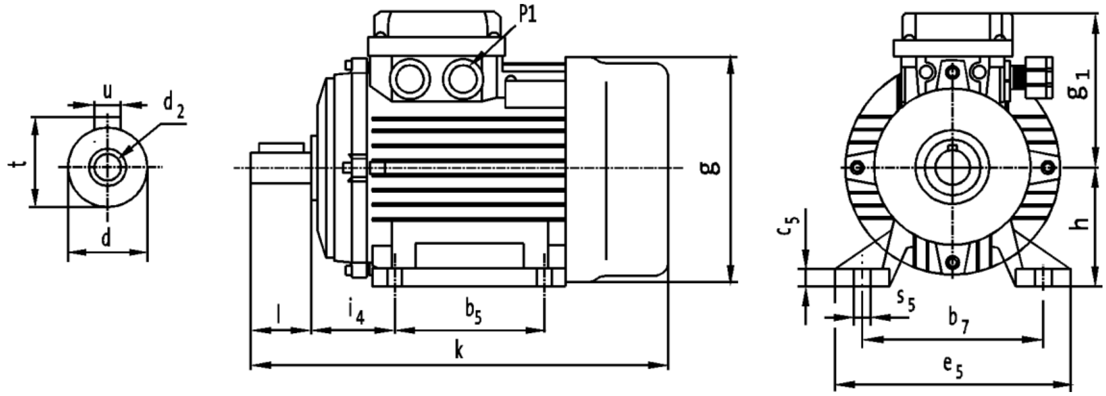
The motor must satisfy the minimum power required for the drum. Every part of the drive has its specific efficiency. Considering efficiency of V-belt and the gearbox, an appropriate motor can be chosen. Therefore for motor power we have

$$P = \frac{P_b}{\eta} = \frac{6}{0.88} \cong 6.82kW \qquad 5\text{-}1$$

Figure 5.2 shows the selected motor with its parameters.

	P_N	n_N	$U_{N,\Delta}$	$I_{N,\Delta}$	$U_{N,Y}$	$I_{N,Y}$	I_a/I_N
			± 10 %		± 10 %		
	[kW]	[r/min]	[V]	[A]	[V]	[A]	
MHERAXX132-22V1C	7.50	1460	400	14.7			8.20

Electric Motor



	d	d_2	l	t	u
	j6				
	[mm]	[mm]	[mm]	[mm]	[mm]
38	M12x28	80	41.0	10.0	

Figure 5.2 Selected motor [37]

Number of revolutions of driving drum is simply calculated knowing the belt speed and drum diameter

$$n_b = \frac{60 \cdot v \cdot 10^3}{\pi \cdot D_b} = \frac{60 \cdot 1.25 \cdot 10^3}{\pi \cdot 400} = 59.68\, min^{-1} \qquad 5\text{-}2$$

where n_b represents drum revolution. Total speed ratio of the drive is an important parameter for determination of gears and is calculated as a ratio of motor revolutions over drum revolutions

$$i_c = \frac{n_m}{n_b} \qquad 5\text{-}3$$

where i_c is the total speed ratio and n_m is revolutions of the selected motor. Thus, we will have

$$i_c = \frac{1460}{59.68} \cong 24.5\,.$$

Speed ratio of a gearbox, as a part of mechanism including V-belt transmission, must be calculated according to following relation

$$i_p = \frac{i_c}{i_{belt}} \qquad 5\text{-}4$$

where i_p is speed ratio of the gearbox and i_{belt} is speed ratio of belt transmission and typically equals 1.5. the speed ratio then is:

$$i_p = \frac{24.5}{1.5} = 16.33\ .$$

This ratio is equal to the transference number u_p of gearbox and is divided into the partial transference numbers u_{12} and u_{34} of the corresponding pairs of mating gears. The partial transference numbers of gears 1,2 and 3,4 will be chosen (partial transference number should **not** be any integer):

$$u_{12} = 4.4\ ;\ u_{34} = 4.1\ .$$

Having the general formula

$$M = \frac{P}{2 \cdot \pi \cdot n} \qquad 5\text{-}5$$

we will be able to calculate torque of electric motor, input shaft, countershaft and the output one. Where ***P*** is substituted by power (kW), $\boldsymbol{n}$ is the corresponding revolutions $(\,min^{-1})$ and $\boldsymbol{M}$ will be the calculated torque in Nm. Table below shows the calculated torques.

Table 5-1 Calculated torques

Torque	Electric motor	Input shaft	Countershaft	Output shaft
Nm	49.05	67.69	291.9	1172.86

5.1 Design of Gears

Selection of number of teeth is an important start for further calculation and is done according to surface type or heat treatment of pinions and the known transference numbers. Numbers of teeth of pinions are chosen to be:

$$z_1 = 18\ ;\ z_3 = 17,$$

and number of teeth of corresponding wheels will be determined by transference numbers of mating gears

$$z_2 = z_1 \cdot u_{12} \qquad 5\text{-}6$$

$$z_4 = z_3 \cdot u_{34} \qquad 5\text{-}7$$

therefore we will have:

$$z_2 = 79\ ;\ z_4 = 70\ .$$

The meshing gears are of helical type and their helix angle are chosen as following:

$$\beta_{12} = 12° \,;\ \beta_{34} = 8°\,.$$

The estimation of modules and strength check of gears are based on material of gears. They are tooth face hardened steel gears. Table 5-2 present the final calculated values of necessary dimensions.

Table 5-2 Necessary dimensions of designed gears

Variables	Pinion [1]	Gear [1]	Pinion [2]	Gear [2]
$\boldsymbol{z}$	$\boldsymbol{z_1}$	$\boldsymbol{z_2}$	$\boldsymbol{z_3}$	$\boldsymbol{z_4}$
	18	79	17	70
$\boldsymbol{\beta}$	12°	12°	8°	8°
$\boldsymbol{m}$	$\boldsymbol{m_1}$		$\boldsymbol{m_2}$	
	1.75		2.75	
$\boldsymbol{d}$ $[\boldsymbol{mm}]$	$\boldsymbol{d_1}$	$\boldsymbol{d_2}$	$\boldsymbol{d_3}$	$\boldsymbol{d_4}$
	32.2	141.34	47.21	194.39
$\boldsymbol{d_a}$ $[\boldsymbol{mm}]$	36.16	144.83	53.11	199.89
$\boldsymbol{d_f}$ $[\boldsymbol{mm}]$	28.29	136.96	40.73	187.52
$\boldsymbol{x}$	$\boldsymbol{x_1}$	$\boldsymbol{x_2}$	$\boldsymbol{x_3}$	$\boldsymbol{x_4}$
	0.132	0	0.073	0

Where m is the normal module, d is the pitch circle diameter, d_a is addendum diameter, d_f is dedendum diameter and x is the basic rack tooth profile displacement of the corrected gears.

5.2 Preliminary Design of Shaft Diameters

Shafts are loaded by bending, torsion and they can be in thrust too. These shafts are preliminarily designed according to strength condition at torsion

$$\tau = \frac{M_k}{W_k} = \frac{16 \cdot M_k}{\pi \cdot {d_H}^3} \leq \tau_D \qquad 5\text{-}8$$

where τ is shear stress $\left(\frac{N}{mm^2}\right)$, W_k is modulus in torsion (mm^3), M_k is the corresponding torque (Nmm), d_H is shaft diameter (mm) and τ_D is the allowable shear stress and is taken $25\,\frac{N}{mm^2}$ for all the shafts. Below is the summary of calculated shaft diameters.

Table 5-3 Shaft diameters

Diameter	Input shaft	Countershaft	Output shaft
mm	25	35	50

5.3 Check Calculation of Gearings

For the preliminary geometrical dimensions of mating gears and for their materials, the strength check of gearing is realized according to the Czech standard ČSN 01 4686. The calculation has been done by means of a prewritten program in Excel and the following figures present it for both pair of mating gears. It should be noted that S_F is the factor of safety against fatigue fracture into the tooth root and S_H is the safety factor against fatigue failure of the tooth flank surfaces. For orientation the safety factors may be considered: $s_F = 1.4 \div 1.7 \ and \ s_H = 1.1 \div 1.2$.

Gear [1,2]:

Gear check using ISO 6336				units: mm, Nmm, °, kW, MPa, m.s^{-1}					
			Roz.	pinion	wheel			pinion	wheel
z_1	18		d	32.20373	141.3386		σ_{Flim}	450	450
z_2	79		d_a	36.16141	144.8343		σ_{Hlim}	1140	1140
m_n	1.75		d_f	28.29073	136.9636		Y_{Fa}	2.616906	2.219489
x_1	0.132001		d_b	30.18195	132.4652		Ysa	1.610289	1.773605
x_2	0		d_W	32.28866	141.7113		Y_ε	0.706145	0.706145
α_n	20		h_a	1.978842	1.74784		Y_β	0.9	0.9
β	12		h_f	1.956498	2.1875		F_β	6	6
a_W	87		h	3.93534	3.93534		Y_{NX}	1	1
b_1	42		s_n	2.917049	2.748894		Z_H	2.42383	2.42383
b_2	40		s_t	2.982218	2.810305		Z_E	189.8	189.8
P	6.9		v_n	2.580738	2.748894		Z_ε	0.795192	0.795192
n_1	973		v_t	2.638393	2.810305		Z_β	0.989013	0.989013
M_{k1}	67718.55		α_W	20.81156			Z_B	1	1
v	1.640656						Z_{RT}	1	1
u	4.388889						K_A	1.1	1.1
							K_V	1.035617	1.035617
Checking dimensions							$K_{F\alpha}$	1	1
	1	2					$K_{F\beta}$	1.482435	1.479497
h_k	1.510081	1.306101					$K_{H\alpha}$	1	1
s_k	2.57582	2.57582		const. thickness			$K_{H\beta}$	1.543466	1.543466
z'	3	10							
M/z	13.54348	51.14147		over teeth					
d	2.583115	2.583115					S_F	2.389454	2.36296
M/d	35.09311	143.7756		over balls			S_H	1.186815	1.186815

Figure 5.3 Check calculations of fist pair of gears

Gear [3,4]:

Gear check using ISO 6336				units: mm, Nmm, °, kW, MPa, $m.s^{-1}$				
			Roz.	pinion	wheel		pinion	wheel
z_1	17		d	47.20944	194.3918	σ_{Flim}	450	450
z_2	70		d_a	53.09613	199.8895	σ_{Hlim}	1140	1140
m_n	2.75		d_f	40.72341	187.5168	Y_{Fa}	2.789946	2.247724
x_1	0.070722		d_b	44.3112	182.4579	Ysa	1.562731	1.75335
x_2	0		d_W	47.285	194.7029	Y_ε	0.706806	0.706806
α_n	20		h_a	2.943346	2.748861	Y_β	0.936422	0.936422
β	8		h_f	3.243015	3.4375	F_β	6	6
a_W	120.994		h	6.186361	6.186361	Y_{NX}	1	1
b_1	66		s_n	4.461264	4.31969	Z_H	2.458197	2.458197
b_2	63		s_t	4.505107	4.362142	Z_E	189.8	189.8
P	6.762		v_n	4.178116	4.31969	Z_ε	0.792348	0.792348
n_1	221.2121		v_t	4.219177	4.362142	Z_β	0.995122	0.995122
M_{k1}	291902.4		α_W	20.43608		Z_B	1	1
v	0.54681					Z_{RT}	1.1	1.1
u	4.117647					K_A	1.1	1.1
						K_v	1.00635	1.00635
Checking dimensions						$K_{F\alpha}$	1	1
	1	2				$K_{F\beta}$	1.554136	1.550849
h_k	2.226435	2.0547				$K_{H\alpha}$	1	1
s_k	3.939395	3.939395		const. thickness		$K_{H\beta}$	1.625999	1.625999
z'	3	8						
M/z	21.10224	63.66012		over teeth				
d	4.059181	4.059181				S_F	2.345501	2.319803
M/d	51.54959	198.3489		over balls		S_H	1.117676	1.117676

Figure 5.4 Check calculations of second pair of gears

SUMMARY OF RESULTS:

Table 5-4 Safety factors

Safety Factors	Pinion [1]	Gear [1]	Pinion [2]	Gear [2]
S_F	2.39	2.36	2.34	2.32
S_H	1.19	1.19	1.12	1.12

As seen in the summary, the values of safety factor in the tooth root (s_F) are higher than recommended ones. Therefore, the calculation seems slightly uneconomic and on the other side correction requires modification of a lot of variables that eventually might not lead to a perfect result.

5.4 Force Analysis

The force analysis on the gear-box shafts is mostly solved into rectangular coordinate x – y – z by means of two mutually perpendicular planes of the forces acting into the gear mesh of mating gears and of the forces acting into the additional drives. At all the calculations of the mutual acting forces arising into the meshes of spur, helical and bevel mating gears, the energetic loses are usually neglected. Figure 5.5

shows three types of force present in gear meshing called tangential force F_t, axial force F_a and the radial force F_r.

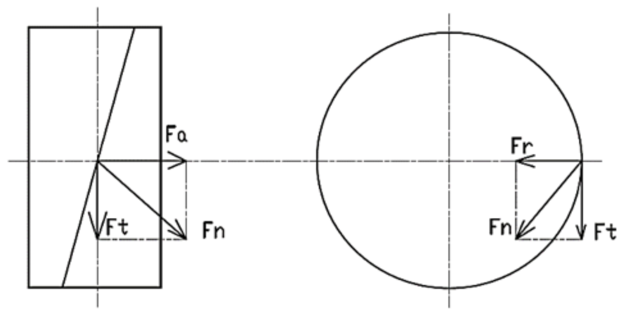


Figure 5.5 Forces present in gearing

Tangential forces are calculated as

$$F_t = \frac{2 \cdot M_k}{d_w} \tag{5-9}$$

axial forces as

$$F_a = F_t \cdot \tan \beta \tag{5-10}$$

and radial forces as

$$F_r = F_t \cdot \frac{\tan \alpha}{\cos \beta} \tag{5-11}$$

where d_w is the rolling circle diameter (corrected gear) and α is normal profile angle which is $20°$. Following table is the summary of the calculated forces for all gears.

Table 5-5 Forces into the gear mesh

	Tangential Force $[N]$	Axial Force $[N]$	Radial Force $[N]$
PINION [1,2]	$\mathbf{4193.13}$	$\mathbf{891.28}$	$\mathbf{1560.27}$
WHEEL [1,2]	$\mathbf{4119.68}$	$\mathbf{875.66}$	$\mathbf{1532.94}$
PINION [3,4]	$\mathbf{12346.51}$	$\mathbf{1735.19}$	$\mathbf{4537.92}$
WHEEL [3,4]	$\mathbf{12047.72}$	$\mathbf{1693.2}$	$\mathbf{4428.11}$

The belt drive in this design is applied as an additional partial transmission mechanism between the electric motor and the gearbox. The driving (little) pulley of this drive is fastened on the shaft of electric motor and the driven (big) pulley of this drive is fastened on the input shaft of gearbox. The rules for V-belt drive design are given by the Czech Standards: ČSN 02 3111 – Classical Driving V-belts and ČSN 02 3114 – Narrow V-belts for Industrial Purposes. Table 5-6 is the calculated forces for belt drive in horizontal and vertical directions.

Table 5-6 Belt drive forces

Force	F_{vx}	F_{vy}
$[N]$	426.7	63.77

F_{vx} is the horizontal component of force and F_{vy} is the vertical one.

Calculation of reaction forces on shaft supports (bearings) plays an important role in bearing selection and its durability. Table 5-7 is the summary of calculated reaction forces of six bearings assuming preliminary design with preliminary interstitial distances.

Table 5-7 Reaction forces

Bearing	A	B	C	D	E	F
FR $[N]$	3749.85	1014.69	7302.18	9441.82	4087.15	8954.11
FA $[N]$	891.28	0	859.52	0	1693.2	0

Where F_R is the radial force on bearings and F_A is the axial one.

5.5 Calculation of Shaft Bearings

Knowing the reaction forces and revolutions of each shaft, it will be possible to select suitable bearings. Using a computer based program, types of bearings and their durability were determined. A pair of tapered roller bearings are applied for the input and the counter shafts and a pair of angular contact ball bearings for the output shaft. Following figures present each of the selected bearings together with their necessary design calculations.

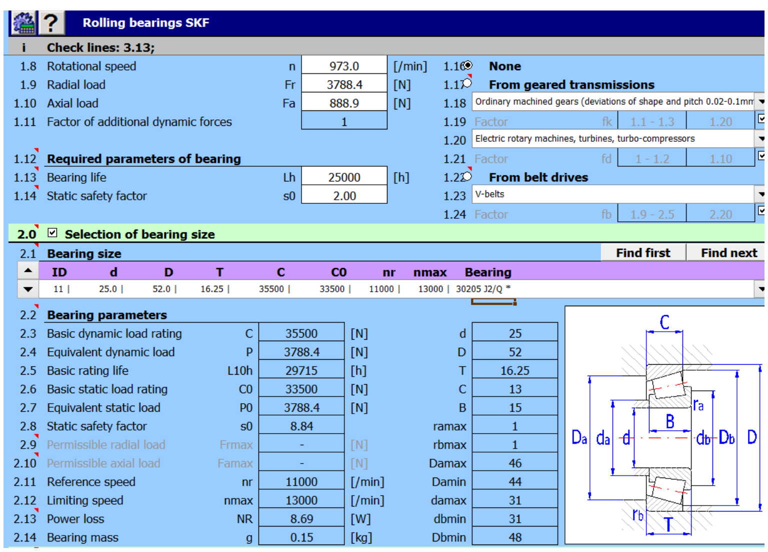


Figure 5.6 Bearing A (on the input shaft)

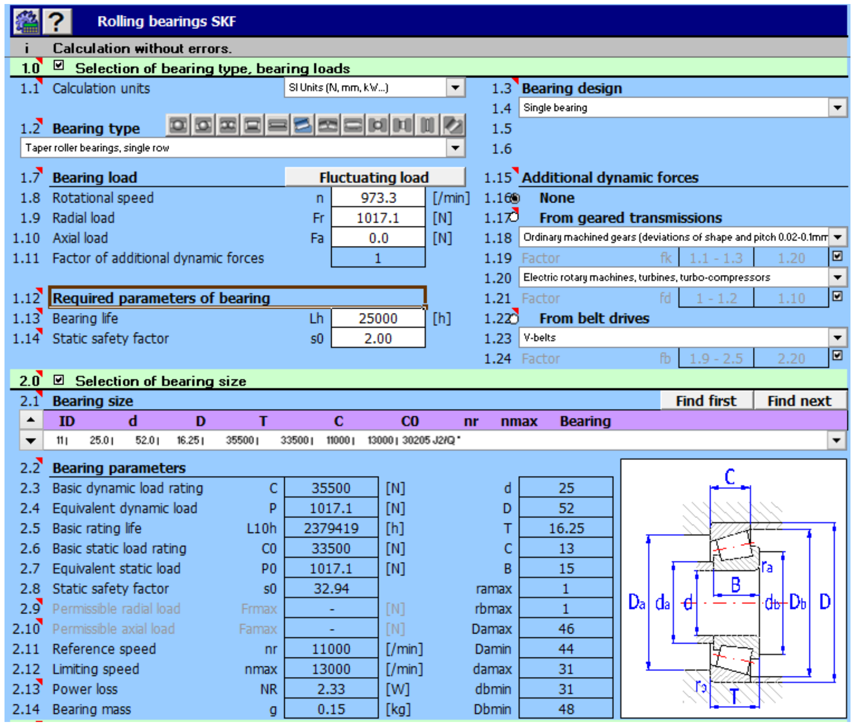


Figure 5.7 Bearing B (on the input shaft)

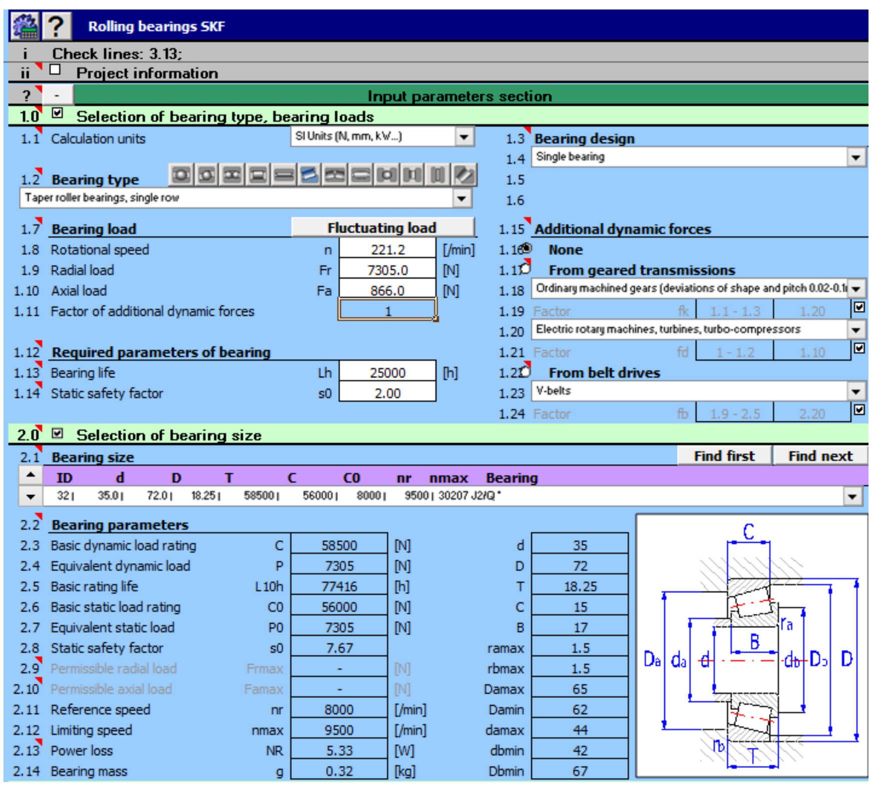


Figure 5.8 Bearing C (on the counter shaft)

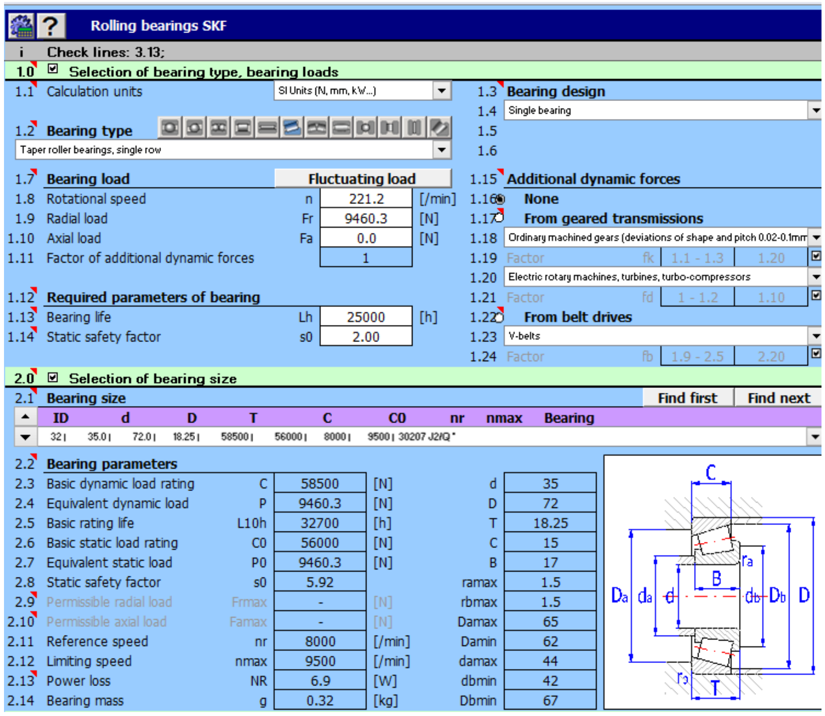


Figure 5.9 Bearing D (on the counter shaft)

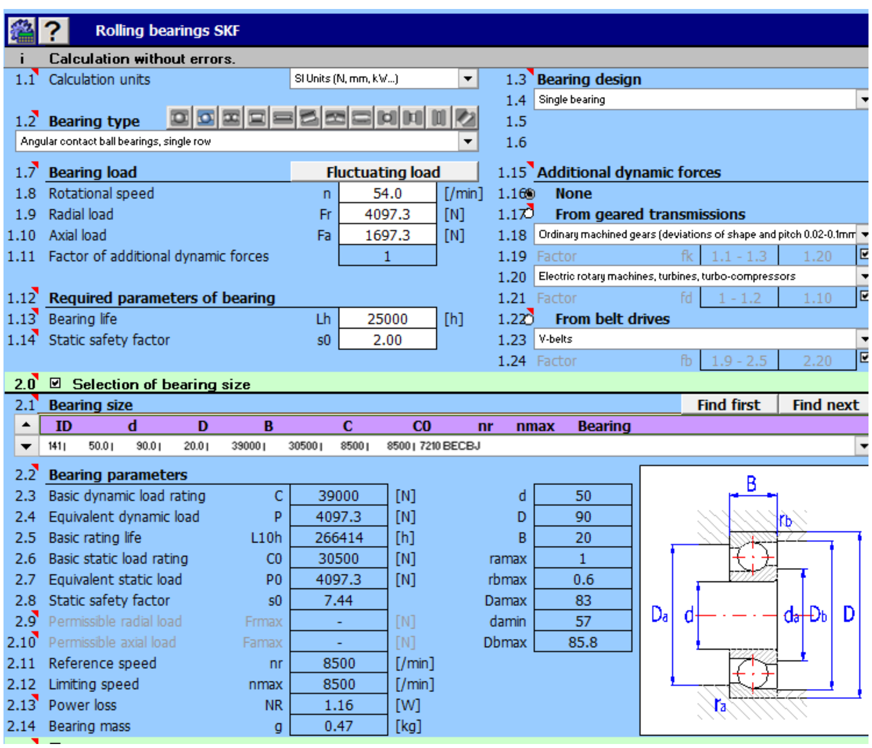


Figure 5.10 Bearing E (on the output shaft)

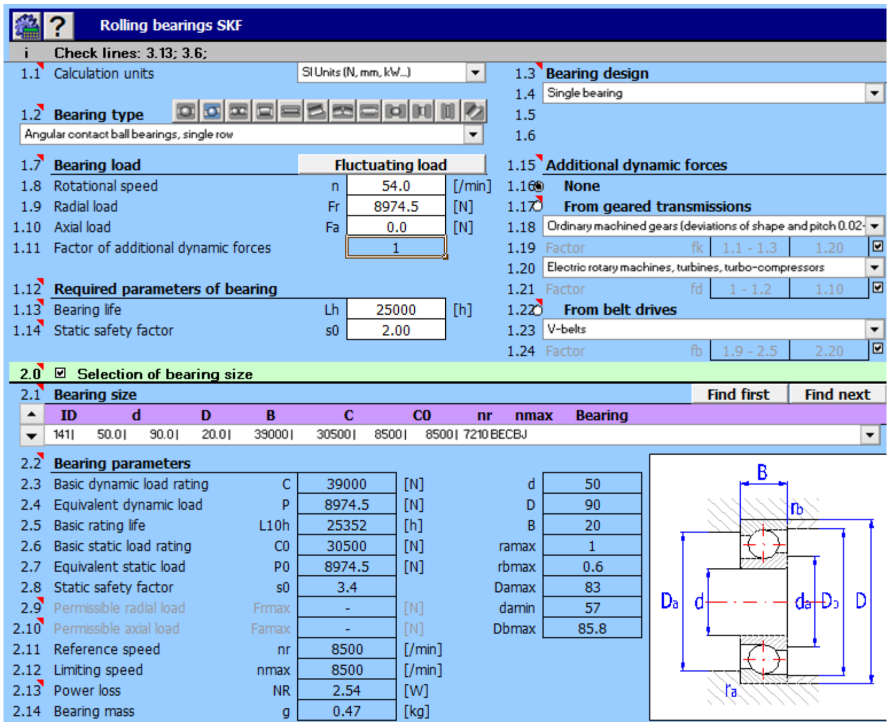


Figure 5.11 Bearing F (on the output shaft)

**The full analysis and calculation of the gearbox are done in subject Design.

Figure 5.12 shows final 3D model of drive including drum, shaft coupling, gearbox, V-belt and the electric motor mounted on a welded frame. (3D model and drawings are attached.)

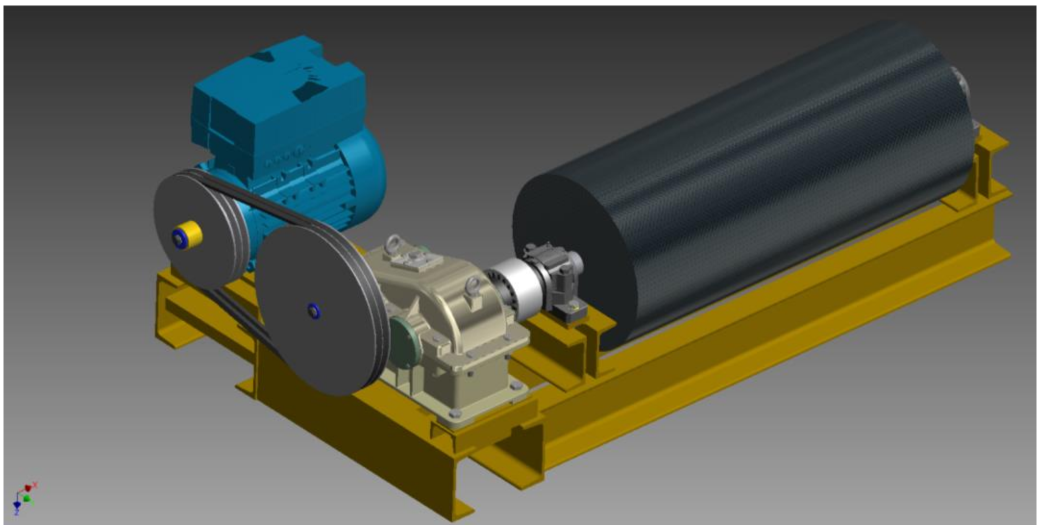


Figure 5.12 3D model of the designed drive

6 Conclusion

We encountered the problem of material handling and the ways to do it more efficiently. There exists many types of conveyor each of which claiming for an easier material transport, that it is still hard to select among them. The two main factors are often application and condition that determine the type of conveyor to be employed. Despite the vast capabilities of belt conveyors, there were some limitations such as certain elevation on which the load can be conveyed, which makes us think about other possible options too.

An appropriate drive always plays an important role in belt conveyor system. We were able to increase belt tension easily by changing pulley configuration and/or improving the efficiency by increasing that of every single part of the drive. However, it is harder in practice to do so than it seems in the theory and we should keep in mind that it requires hard work together with many experiments to enhance the transport capacity of a belt conveyor as well as the length over which the material is transported.

There were many possibilities to induce a higher tension into the belt, including various drive pulley arrangements. Other aspects to be considered, are the belt construction to bear this high tension and the proper pulley lagging to control the friction. When all of these factors are taken into account, the choice is limited.

As every industry improves over time, belt conveyor industries try to make bulk material handling easier and do the transport with higher capacity in longer distances still with less power.

7 References

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