To ensure correct selection and calculation of conveyor chains, the following factors should be considered:

- conveyor type
- total transported weight
- chainvelocity
- chainpitch
- chain interconnection
- operating environment
- lubrication
- tensile strength, breaking strength of chain

1) CONVEYOR TYPE

Conveyors fall into following categories:

a) horizontal
b) inclined
c) vertical
d) combined

2) TOTAL TRANSPORTED WEIGHT

Total transported weight is defined as total weight of transported materials that exerts force onto the conveyor chain and its possible supports (slats - light carrying elements - crossbar supports - joints etc.).

The load distribution must also be considered. The process of calculation for cases where the load is distributed equally over the conveyor differs significantly from that one where the load pushes on smaller supporting areas.

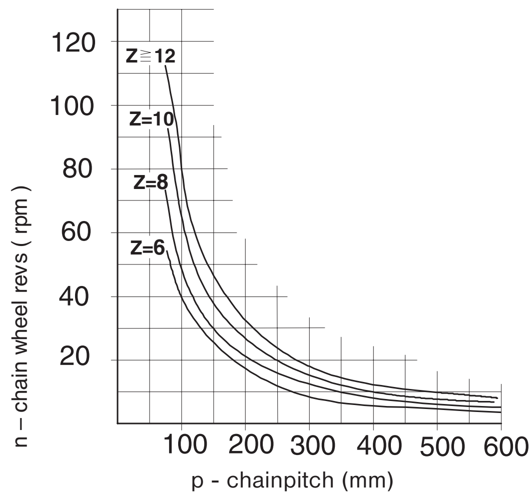
3) CHAIN VELOCITY

The chainvelocity represents the chain trajectory per unit of time. This velocity influences the conveyor performance. The conveyor performance depends on chainpitch and ratio of diameters of driving vs. driven sprockets.

The following figure shows these dependencies.
Fig. 1.

$$v = \frac{p.Z.n}{1000} \quad (m/min.)$$

v = chainvelocity
Z = number of teeth
p = chainpitch (mm)
n = sprocket revs (n/min.)



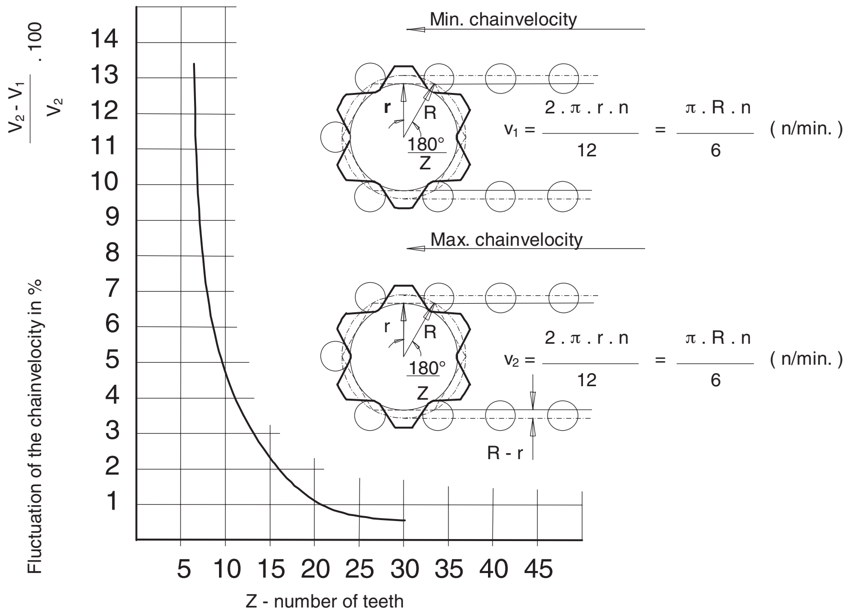
The chainvelocity for conveyors is limited to 60 m/s. The optimum is 0 to 30 m/min. Special care must be paid to conveyors where chainvelocity falls below 1 m/min.

If such low chainvelocity and conveyor length over 80 to 100 meters happen in coincidence, along with small driving sprockets (Z = 18 - 20) and high chainpitch, then the conveyor can "pendulate", e.g. the conveyor starts and stops repeatedly.

Such effect results from loss of synchronization of driving and driven sprocket; the polygonal effect occurs as the chain is taken by the sprockets.

The following figure shows the range of velocity fluctuations in %.

Fig. 2



Z = number of teeth
R = sprocket radius (m)
n = sprocket revs (rpm)

$r = R . \cos \frac{180°}{Z}$ (m)

r - R = range of velocity fluctuations

The chain does not "pendulate" if the above defined range of velocity fluctuations, expressed in %, falls below **1**%.

4) CHAINPITCH

The chainpitch is defined by the distance of subsequent chain pins, expressed in mm or inches.
The chainpitch is determined based on the following conveyor characteristics:

a) chainvelocity
b) diameter of the driving and driven sprocket
c) load distribution on the conveyor
d) dimensions of possible supporting elements (slats - light carrying elements - crossbar supports - joints etc.).

5) CHAIN INTERCONNECTION

Normally, two conveyor chains are interconnected with angles. The angle is welded onto chain plates. Chains can be interconnected with other elements. The chain interconnection is defined by chain dimensions, shape and number of supporting elements per meter of chain.

6) OPERATING ENVIRONMENT

The operating environment is defined by the area and conditions where the chain runs.
The following factors should be considered:

- cleanness, cleaning
- temperature
- presence of abrasives (e.g. dust, sand, gravel, flour)
- humidity, atmospheric agents
- etc.

These factors influence chain dimensioning, material quality, allowances, production tolerances, galvanizing and safety coefficients.

Table I contains the temperature coefficients.

Temperature	*Proper work load*	*Correction coefficient*
- 40°C ~ - 20°C	(maximum permissible work load)	x 0,25
- 20°C ~ - 10°C	(maximum permissible work load)	x 0,30
- 10°C ~ 160°C	(maximum permissible work load)	x 1,00
160°C ~ 200°C	(maximum permissible work load)	x 0,75
200°C ~ 300°C	(maximum permissible work load)	x 0,50

7) LUBRICATION

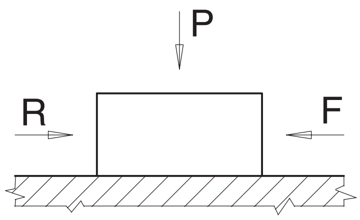
The chain operating life is very much influenced by its lubrication. Proper lubrication increases wear resistance of the chain, its corrosion resistance and oxidization resistance of all chain elements.
Chain lubrication must fulfil these main criteria:

a) to lower the friction coefficients
b) to lower chain wear and bring energy savings
c) to protect the chain from corrosion
d) to ensure proper chain operation

a) to lower the friction coefficients

Every motion must overcome some resistance, called friction.
Suppose that we have a solid body having mass "P" laying on level surface.

Fig. 3



F - direction of motion
The friction " R " affecting a solid body moving over another, is counter to the direction of motion.

The friction depends on two factors :

- mass of the solid body "P"
- static "μ" or dynamic "τ" friction coefficient

The formula for R follows:

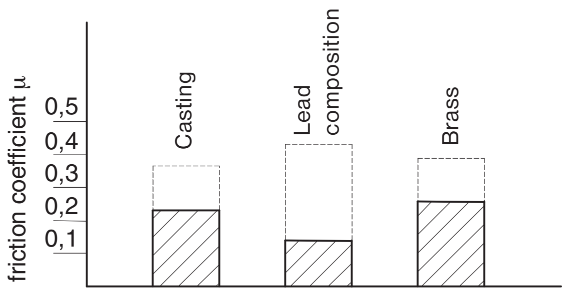
$R = \mu . P$ (kg)
$R = \tau . P$ (kg)

The friction coefficient "τ" is always smaller than "μ". Thus the force needed to start the motion is greater (1.5 to 3 times) than the force needed to keep the solid body in motion.

The values of "τ" and "μ" depend on the quality of surfaces in contact, on the contact itself (slide, slippage, roll) and the presence of lubrication.

Fig. 4 shows how the presence of mineral oil on a contact of two metals lowers the coefficient "μ".

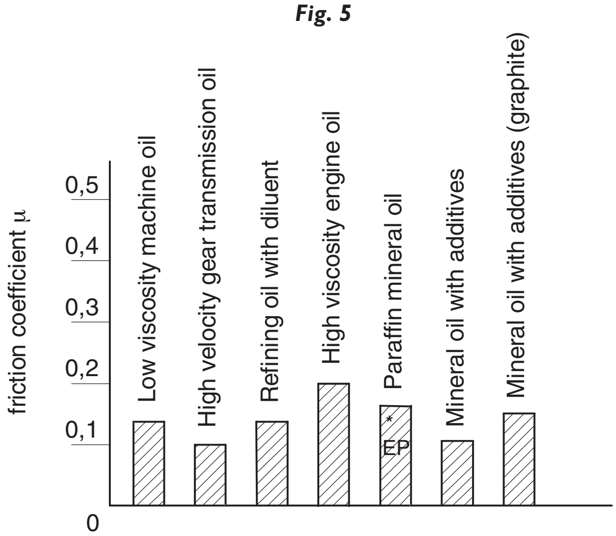
Fig. 4



The values of static friction coefficient "μ" for various materials on steel, lubricated with mineral oil (solid line) and without lubrication (dashed line).

Fig. 5 shows the influence of lube type on the value of static friction coefficient "μ".

Fig. 5



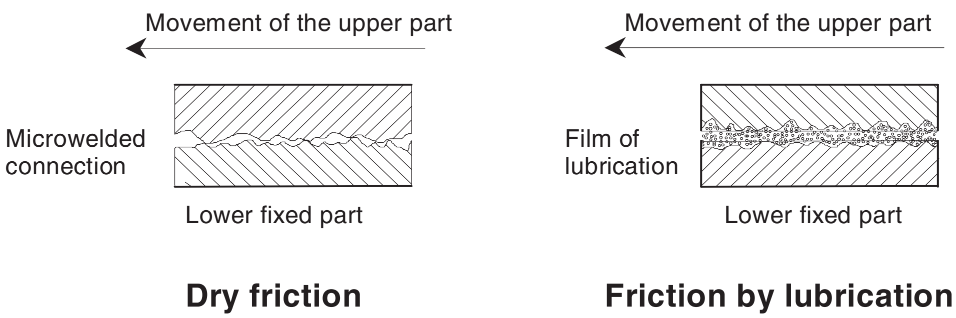
The values of static friction coefficient "μ" for steel on steel, various lubes.

* EP = Extreme Pressure (extreme pressure oil with additives)

b) to lower chain wear and bring energy savings

The relative movement of bushing vs. pin, bushing vs. roller without lubrication leads to progressive abrasion of the respective surfaces. This leads to early wear of the chain, marked increase of friction and thus the need to increase driving power. A suitable lubrication film prevents the metal surfaces from direct contact, thus preventing the above negative tendencies from occurring.

Fig. 6



c) to protect the chain from corrosion

Any metal, unless properly protected, tends to oxidize. Some operating conditions speed up the process:
- high temperature
- high humidity
- presence of aggressive chemical agents

Oxidization is a threat to chain operating life. The presence of suitable lubrication film on chain surface prevents the oxidization and build-up of corrosion. Such protection is advanced by presence of corrosion inhibitors in the lube.

d) to ensure proper chain operation

Every user tries to solve lubrication problems by use of one type of lube. This is often, however, not the best practice, as there are many parameters that influence lube selection. The basic parameter is chain operating temperature. Operating temperatures fall into following categories:
- low temperature, down to -40 deg. C
- common temperature, +15 to +150 deg. C
- high temperature, +150 to +250 deg. C
- very high temperature, over +250 deg. C

Low temperature, down to -40 deg. C
Synthetic lube of the lowest possible viscosity must be used. Lube dispersion in suitable solvent often helps. The solvent must facilitate lube penetration into pins, bushings and rollers (e.g. freon, trichloroethylene).

After the solvent evaporates, a lube film is formed on the surfaces of chain elements. The film exhibits high adhesion and resistance against mechanical factors of chain operating environment.

Common temperature, up to +110 deg. C, transients up to +150 deg. C
These are most common operating conditions. We do not recommend the use of mineral oils. Special chain lubes are the best suited for these applications. These lubes contain additives. The additives prevent lube from splashing and improve its capillarity.

High temperature, +150 to +250 deg. C
Synthetic oils must be used within this temperature range. Synthetic oils possess higher thermal stability than mineral oils and contain a combination of solid graphite-based of disulphide-based pigments. These additives characterize synthetic oils as "emergency mode" lubricants and allow for increase of specific pressure on chain joints. The quality of synthetic oils depends on the quality of tempers and additives that prevent the build-up of deposits on chain.

Very high temperature, over +250 deg. C
Fluid lubrication is not possible within this temperature range. Use suspensions of solid lube in synthetic solution, which evaporates and allows for dry-lubrication. As this procedure involves evaporation, the lube must be supplied to the coolest point of the chain transmission.

8) LUBRICATION SYSTEMS

The selection of proper lube and lubrication system determine the chain transmission's reliability and efficiency. The following lubrication systems are used the most often:

a) occasional manual lubrication by oilcan
b) occasional manual lubrication by soaking the chain in lube
c) intermittent lubrication by drip oiler
d) oil dip
e) lubrication under pressure, oil jets focused onto chain section
f) lubrication by oil spray

Manual lubrication by oilcan should be avoided. This method does not make sure that every chain link is lubricated. Occasional manual lubrication by soaking the chain in lube is more convenient. If the chain stays within the oil bath for longer period of time, the lube can penetrate all inner sockets of a chain link. Small speed and small transmitted power allow to use of drip oiler. Higher speed and transmitted power require the use of oil dip. This method is very reliable. Periodic pulse lubrication or oil spray lubrication are best suited for high speed and high transmitted power.

The selection of proper lube type and lubrication system for a given chain transmission also depends on the value of specific pressure on chain joints.

9) CHAIN CLEANING

The basic prerequisite for correct function and operation of chain transmission is proper lubrication and cleaning. The lubrication is not as much effective unless the chain has been cleaned prior to lubrication. All dirt and deposits must be removed. Suitable solvents (chlorine- or fluorine-based) can be utilized. Both types of solvents remove kerosene- or diesel-based contamination. The metal surfaces of the chain must be clean and damp after cleaning. This cannot be accomplished by use of kerosene- or diesel-based solvents. Such solvent leave an oily film on metal surfaces. The lube tends to "float" on such surfaces. Moreover, kerosene remains can induce or speed up the lube oxidization processes at high temperature. The chain is always to be cleaned cold. Apply the solvent with a brush. When the machine cannot be turned off, we recommend to apply the lube thinned with a detergent (1:1).

10) TENSILE STRENGTH, BREAKING STRENGTH

The tensile strength, expressed in Newton, represents the chain load that breaks it. Its values, listed in catalogues, are test results acquired at a temperature of ca. 20 deg. C. ISO 1977 norms require the minimal measured breaking strength to be at least 95% of the value listed in the catalogue for a particular chain type.

11) SELECTION OF CHAIN TYPE BASED ON TRACTION FORCE

The traction force represents the force necessary to move the chain, mechanical elements connected to it and transported load. The computation proceeds in two phases:

1. preliminary phase - the chain model, weight and friction coefficient are determined
2. verification phase - weight and friction coefficient of real chain are put in equation

The following factors influence the determination of the traction force:

a) weight of transported material
b) weight of chain and additional supporting elements (slats - light carrying elements - crossbar supports - joints etc.)
c) friction coefficient
d) corrective coefficient for type of operation, depends on load and operating hours per day
e) velocity correction coefficient
f) corrective coefficient when the chain is taken by the sprocket (small number of teeth)
g) safety coefficient and specific pressure

a) weight of transported material

see paragraph 2

b) weight of chain and additional supporting elements (slats - light carrying elements - crossbar supports - joints etc.)

Preliminary phase utilizes the approximate weight of whole chain branch as chain weight. Verification phase utilizes the real weight of chain.

c) friction coefficient

The friction coefficient determines the force necessary to overcome friction of two parts that touch each other and are in relative motion. When the chain touches its guideway, then the slide, slippage friction occurs - "fr".
The following table lists the values of slide, slippage friction coefficient:

Table 2

Bodies in contact	slide friction coefficient "fr", dry surface	slide friction coefficient "fr", lubricated surface
Steel chains on hard wooden guideways	0,44	0,29
Steel chains on steel guideways	0,30	0,20
Steel chains on chopped, uneven or rusted guideways	0,35	0,25
Steel chains on high density polyethylene guideways	0,18	0,05

When chains are fitted with protruding rollers, the combination of slide and roll friction occurs - "fv".
Preliminary phase utilises a value of fv = 0,2 ,

$$fv = C \times \frac{r}{R} + \frac{b}{R} \text{ (m)}$$

where:
C = coefficient of slide, slippage friction between bushing and roller (listed in table on next page)

Table Nr. 3

Bodies in contact	no lubrication "C"	lubricated "C"
Steel roller, steel bushing	0,25	0,15
Steel roller, bronze bushing	-	0,13
Nylon roller, steel bushing	0,15	0,10

r = outer diameter of bushing (mm)
R = outer diameter of roller (mm)
b = empirical coefficient of roll friction depending on materials of contact areas and their finish
b = 0,5 for steel roller on even steel guideway
b = 1,0 for steel roller on rugged steel guideway

Important:
The static friction coefficient is 1.5 to 3 times higher than the dynamic one, thus the roller diameter should be at least 2.5 times greater than that of the bushing to ensure proper roll, not slide.

d) corrective coefficient for type of operation Fs

This correctional coefficient applies to traction force, depending on characteristics and environment where the conveyor is operated. The following table lists the values of Fs in the most common applications:

Table Nr.4

Operating conditions	Fs
Load balance - centered load - uneven load distribution	1,0 1,5
Load characteristics - even load: overload occurrence less than 5% - small fluctuations: overload occurrence 5 to 20 % - great fluctuations: overload occurrence 20 to 40 %	1,0 1,2 1,5
Frequency of start/stop under load - less frequent than 5 times a day - 5 times a day to twice an hour - more frequent than twice an hour	1,0 1,2 1,5
Operating environment - relatively clean - medium dusty or contaminated - damp, very contaminated or corrosive	1,0 1,2 1,3
Workhours per day - up to 10 - up to 24	1,0 1,2

Total value of Fs is a multiply of all partial values of Fs corresponding to individual elements representing chain's operating conditions.

e) velocity correction coefficient Fv
This correctional coefficient applies to traction force, depending on chainvelocity and its ratio to the number of teeth of the driving and driven sprocket - see the following table.

Table 5

Chainvelocity m/min.	Number of teeth of the sprocket 6	7 – 8	9 - 10	11 - 12	13 - 16	17 - 20	21 - 24
3,0	0,9	0,8	0,8	0,8	0,7	0,7	0,7
7,5	1,0	0,9	0,8	0,8	0,8	0,8	0,7
15,0	1,4	1,0	0,9	0,9	0,8	0,8	0,8
30,0	2,0	1,3	1,1	1,0	0,9	0,9	0,8
60,0	4,4	2,0	1,4	1,2	1,1	1,0	0,9

f) corrective coefficient when the chain is taken by the sprocket (small number of teeth) Fa

This correctional coefficient applies to traction force. Traction force is escalated by additional friction as the chain is taken by driving and driven sprocket.

Fa = 1,05 for sprockets mounted on bronze bushings
Fa = 1,03 for sprockets mounted on bearings

"Fa" is close to 1 and its influence is thus neglected in our calculations.

g) safety coefficient and specific pressure

The permissible strain exerted to chain construction materials must be considered for the chain to be properly dimensioned. Even the operating load, i.e. 2/3 of breaking strength, strains the chain material beyond the point of "set" (0.2% of tensibility). We therefore recommend the breaking strength to be 8 times the maximum traction force. This ratio is the "safety coefficient".

In some cases, the real load of chains operated within harsh conditions cannot be pre-calculated. Thus, a sizable safety coefficient must be applied. Our technical department will be glad to assist you with the determination of proper safety coefficients.

We recommend to check specific pressures on chain joints, bushings vs. rollers and pins vs. bushings. This is especially important for uneven load, where the calculation of traction force is not sufficient for proper selection of chain type. If the values of specific pressure do not fit within permissible limits (see Tables 6 and 7), a chain of greater area of working surfaces must be selected. This lowers the specific pressure on chain joints.

Specific pressure calculation:

a) specific pressure on roller $= \dfrac{P}{L \,.\, Dr}$ (MPa)

a) specific pressure on pin $= \dfrac{T}{Lb \,.\, Dp}$ (MPa)

where

P = load of each roller (N)
T = effective traction force (N)
L = roller hole length (mm)
Lb = total length of bushing (mm)
Dr = roller hole diameter (mm)
Dp = outer diameter of pin (mm)

Maximum permissible specific pressures:

Table 6

Materials in contact *Bushing*	*Pin*	***Maximum specific pressure MPa***
Cemented steel	Cemented steel	25,0
Cemented steel	Treated steel	21,0
Iron	Cemented steel	17,5
Stainless steel	Stainless steel	12,0
Bronze	Cemented steel	10,0

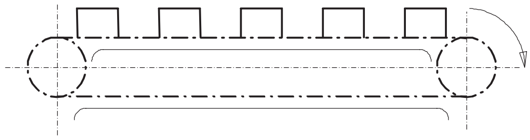
Tabelle Nr. 7

Materials in contact *Roller*	*Bushing*	***Maximum specific pressure MPa***
Cemented steel	Cemented steel	10,0
Treated steel	Cemented steel	10,0
Iron	Cemented steel	7,0
Bronze	Cemented steel	6,0
Polyethylene	Cemented steel	1,0
Stainless steel	Stainless steel	4,0
Iron	Bronze	2,8

Formula for traction force of chain conveyor

a) horizontal conveyor - slide guideway

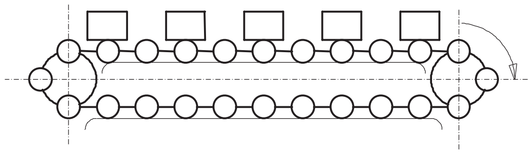
Fig. 7



$$T = 9{,}81 \cdot \frac{(P + P1) \cdot fr \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

b) horizontal conveyor - idle pulleys

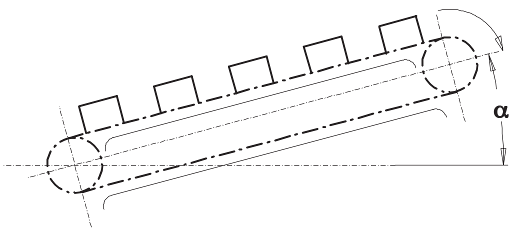
Fig. 8



$$T = 9{,}81 \cdot \frac{(P + P1) \cdot fv \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

c) inclined conveyor - slide guideway

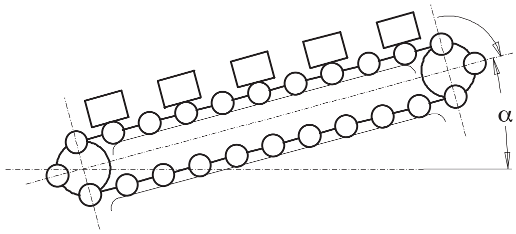
Fig. 9



$$T = 9{,}81 \cdot \frac{[\cos\alpha \cdot (P + P1) \cdot fr + \sin\alpha \cdot P1] \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

d) inclined conveyor - idle pulleys

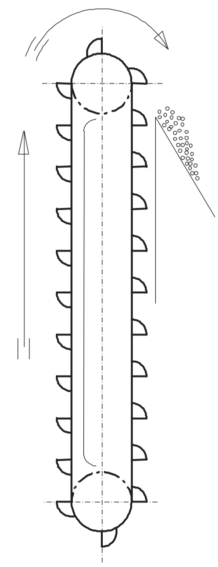
Fig. 10



$$T = 9{,}81 \cdot \frac{[\cos\alpha \cdot (P + P1) \cdot fv + \sin\alpha \cdot P1] \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

e) vertical conveyor

Fig. 11



$$T = 9{,}81 \cdot \frac{(P/2 + P1) \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

All methods depicted above must be utilized in case of conveyors that mix the above principles.

Traction force computation for slat conveyors

We utilize the following additional factors, along with all parameters specified above:

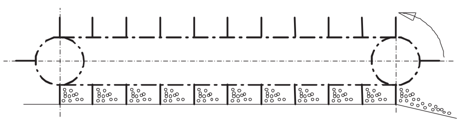
fm = friction coefficient; transported material vs. conveyor conduit - see Table 8
L = loaded length of conveyor (m)
Q = material flow (T/h)
H = conveyor conduit height (m)
B = conveyor conduit width (m)
β = conveyor repletion coefficient - 0,5 -0,6
γ = specific weight of transported material (T/m^3) - see Table 8
v = chainvelocity (m/sec.)

Table 8

Transported material	** *specific weight of transported material γ (tons/cubic meter)*	** *friction coefficient fm*
Gat	0,45	0,7
Corn (wheat)	0,75	0,4
Corn (maize)	0,80	0,4
Barley (bere)	0,45	0,7
Rye	0,65	0,4
Rice	0,75	0,4
Linen seed	0,70	0,4
Dry malt	0,40	0,4
Wheat flour	0,70	0,4
Maize flour	0,65	0,4
Refined sugar (powdered)	0,80	0,5
Cement	1,00	0,9
Anthracite coal	0,70 - 0,90	0,4
Coke	0,50	0,7
Dry clay	1,60	0,7
Ashes	0,60	0,6
Cement chippings	1,30	0,8

a) horizontal slat conveyor - slide guideway

Fig. 12



$$T = 9{,}81 \cdot \frac{(P \cdot fr + P1 \cdot fm) \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

or

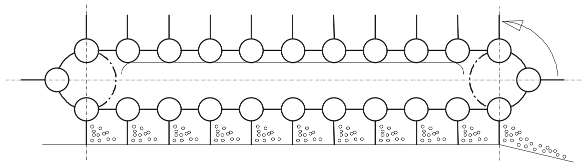
$$T = 9{,}81 \cdot \frac{\left(P \cdot fr + L \cdot \frac{Q}{3{,}6 \cdot v} \cdot fm\right)}{\text{number of chains}} \cdot Fs \cdot Fv \quad (N),$$

where $P1 = H \cdot B \cdot L \cdot \beta \cdot \gamma \cdot 1000$ (kg)

or $P1 = L \cdot \frac{Q}{3{,}6 \cdot v}$ (T) , ***where*** $Q = H \cdot B \cdot L \cdot \beta \cdot \gamma \cdot v \cdot 3600$ (T/h)

b) horizontal slat conveyor - slats and idle pulleys

Fig. 13



$$T = 9{,}81 \cdot \frac{(P \cdot fv) + (P1 \cdot fm) \cdot Fs \cdot Fv}{\text{number of chains}} \quad (N)$$

or
$$T = 9{,}81 \cdot \frac{\left[P \cdot fv + L \cdot \left(\frac{Q}{3{,}6 \cdot v} \cdot fm \right) \cdot Fs \cdot Fv \right]}{\text{number of chains}} \quad (N),$$

where $P1 = H \cdot B \cdot L \cdot \beta \cdot \gamma \cdot 1000$ (kg)

or $P1 = L \cdot \frac{Q}{3{,}6 \cdot v}$ (T) , ***where*** $Q = H \cdot B \cdot L \cdot \beta \cdot \gamma \cdot v \cdot 3600$ (T/h)

Calculation of necessary shaft power

We recommend the following method of determination of total traction force exerted on chain conveyor:

$$Mt = \frac{9550 \cdot N}{n} = \frac{T \cdot dp}{2000} \quad (Nm)$$

$$N = \frac{T \cdot v}{1000} = \frac{Mt \cdot n}{9550} \quad (kW)$$

Mt = shaft torque (Nm)
N = necessary shaft power (kW)
n = revs of driving wheel (1/min.)
T = traction force (N)
dp = pitch diameter of driving wheel (m)
v = chainvelocity (m/sec.)

The above mentioned equations can be combined to:

$$T = \frac{1000 \cdot N}{v} = \frac{2000 \cdot Mt}{dp} \quad (N)$$

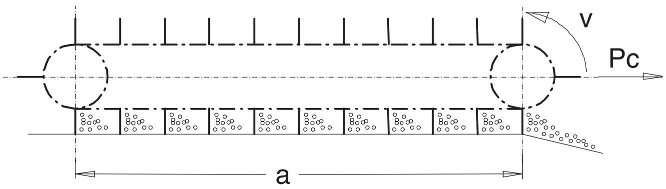
The necessary shaft power calculated using the above formula is influenced by efficiency of real chain conveyor as a whole

12) EXAMPLE OF CALCULATION AND SELECTION OF CONVEYOR CHAIN

A suitable type of chain has to be selected for horizontal slat conveyor:

Given values:

Transported material	brown coal
Conveyor length	30 m
Flow	40 T/h
Conveyor conduit width	350 mm
Conveyor conduit height	250 mm
Number of chains	1
Number of teeth of the sprocket	9 (pre-selected)
Load distribution	even



1. Material flow ***Q***

Q = 40 T/h

2. Chainvelocity ***v***

we use the formula: Q = H . B . β . γ. v . 3600 (T/h)

$$\mathbf{v} = \frac{Q}{H \,.\, B \,.\, \beta \,.\, \gamma \,.\, 3600} = \frac{40}{0{,}25 \,.\, 0{,}35 \,.\, 0{,}6 \,.\, 0{,}7 \,.\, 3600} = \mathbf{0{,}3\ m/sec}.$$

β = 0,6 (pre-selected)
γ = 0,7 (found in Table 8)

3. Weight of transported material ***P1***

$$\mathbf{P1} = a \,.\, \frac{Q}{3{,}6 \,.\, v} = 30 \,.\, \frac{40}{3{,}6 \,.\, 0{,}3} = \mathbf{1\ 111\ kg\ .\ .\ .\ 10\ 900\ N}$$

4. Selection of suitable chain

Selected chain must resist the weight of transported material multiplied with safety coefficient (k = 7), thus its break strength must be:

F_B = P1 . k = 10 900 . 7 = **76 300 N**

Corresponding type of chain according to **DIN 8167 (ISO 1977) is MRC 80 x 125.**

*5. Chain weight **P***

The selected chain's weight per meter is **q = 4,5 kg/m**; its pitch is **p = 125 mm** and the assumed number of teeth of the sprockets is **Z = 9**. Total chain weight is:

$$\mathbf{d_t} = \frac{p}{\sin\frac{180°}{Z}} = \frac{125}{\sin\frac{180}{9}} = \mathbf{367{,}6\ mm\ .\ .\ .\ 0{,}368\ m}$$

$$\mathbf{L} = 2\ .\ a + \pi.\mathbf{d_t} = 2\ .\ 30 + \pi\ .\ 0{,}368 = \mathbf{61{,}16\ m}$$

$$\mathbf{P} = \mathbf{L}\ .\ q = 61{,}16\ .\ 4{,}5 = \mathbf{275\ kg}$$

*6. Selection of friction coefficient **fr***

The chain slides on even steel guideway. The estimated reading of Table 2 is **fr = 0,3**.

*7. Correction coefficient for type of operation **Fs***

See Table 4:

Load balance - centered load	Fs = 1,0
Load characteristics - small fluctuations	Fs = 1,2
Frequency of start/stop under load	Fs = 1,2
Operating environment - medium dusty	Fs = 1,2
Workhours per day	Fs = 1,0
Total friction coefficient	**Fs = 1,73**

*8. Determination of velocity correction coefficient **Fv***

Chainvelocity **v = 0,3 m/sec.** see Table 5; for **Z = 9** the resulting value **Fv = 0,8**.

*9. Friction coefficient **fm***

The friction coefficient "fm" describes the influence of friction of transported material vs. conveyor conduit.
See Table 8 - for given material, **fm = 0,7.**

*10. Computation of traction force **T***

$$\mathbf{T} = 9{,}81\ .\ \frac{(P\ .\ fr + P1\ .\ fm)\ .\ Fs\ .\ Fv}{\text{number of chains}} = 9{,}81\ .\ \frac{(275\ .\ 0{,}3 + 1\ 111\ .\ 0{,}7)\ .\ 1{,}73\ .\ 0{,}8}{1} = \mathbf{11\ 679\ N}$$

*11. Computation of necessary shaft power **N***

$$\mathbf{N} = \frac{T\ .\ v}{1000} = \frac{11\ 679\ .\ 0{,}3}{1000} = \mathbf{3{,}5\ kW}$$

12. Specific pressure on chain joints p_t

$$\mathbf{p_t} = \frac{T}{f} = \frac{11\ 679}{468} = \mathbf{24{,}96\ MPa} < \mathbf{25\ MPa}\ \text{(see Table 6)}$$

f = 468 mm² according to the chain catalogue.
Computed specific pressure is lower than maximum permissible pressure listed in Table 6.

The selected chain fulfils requirements.

13) SPROCKETS

The design and actual condition of the sprocket influences the chain operating life the most. Generally speaking, the following recommendations should be followed:

a) sprockets of the biggest possible diameter should be employed to lower the pressure on chain joints and the polygonal effect.

b) driving wheels should be located at conveyor end, especially with complicated conveyor designs (e.g. conveyors that include fermentation or dry-up compartments).

c) if the conveyor employs two or more interconnected chains, their driving sprockets must not be interlocked

The following formula can be used to determine the pitch, outer and root diameters of a sprocket:

$$\mathbf{Dp} = \frac{P}{\sin\frac{180^\circ}{Z}} \quad \mathbf{(mm)}$$

$$\mathbf{De} = Dp + (0{,}55 \div 0{,}8) \,.\, D \quad \mathbf{(mm)}$$

$$\mathbf{Di} = Dp - D \quad \mathbf{(mm)}$$

where:

Dp = pitch diameter (mm)
De = outer diameter (mm)
Di = root diameter (mm)

P = chainpitch (mm)
Z = number of teeth
D = diameter of chain roller (mm)

If put in the following equation:

$$\mathbf{Dp} = \frac{P}{\sin\frac{180^\circ}{Z}} \quad \textbf{for sin } \mathbf{\frac{180^\circ}{Z}} \text{ is filled in } \mathbf{\frac{1}{n}},$$

If "$\frac{1}{n}$" (see the values of "n" in Table 9 below) is put in place of " $\sin\frac{180}{Z}$ " in the above formula for Dp, we get the following:

$$\mathbf{Dp} = \frac{P}{\frac{1}{n}} = P \,.\, n \quad \mathbf{(mm)}$$

Table 9

Number of teeth	"n"	Number of teeth	"n"	Number of teeth	"n"
6	2,000	21	6,709	36	11,474
7	2,305	22	7,027	37	11,792
8	2,613	23	7,344	38	12,110
9	2,924	24	7,661	39	12,428
10	3,236	25	7,979	40	12,746
11	3,549	26	8,296	41	13,063
12	3,864	27	8,614	42	13,382
13	4,179	28	8,931	43	13,700
14	4,494	29	9,249	44	14,018
15	4,810	30	9,567	45	14,336
16	5,126	31	9,885	46	14,654
17	5,442	32	10,202	47	14,972
18	5,759	33	10,520	48	15,290
19	6,076	34	10,838	49	15,608
20	6,392	35	11,156	50	15,926

14) CONCLUSION

The use of chain transmissions presents the user with many different problems and aspects. All such factors must be considered and solved during the design of particular chain transmission. The information presented above cannot fully explain all problems pertaining to this area. Its purpose is to highlight the basic problems and to help avoid elementary errors.