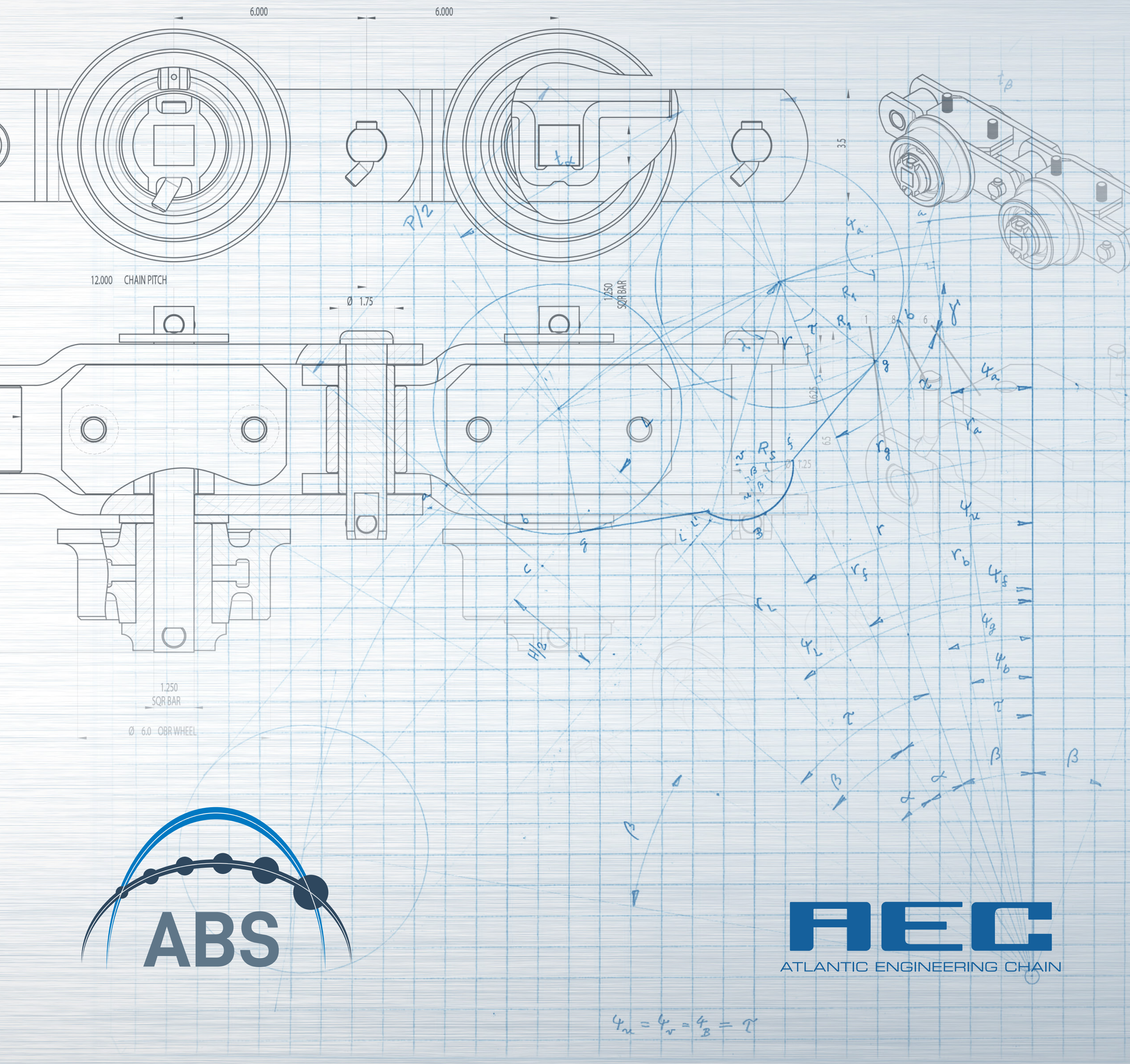
TECHNICAL SECTION

Power Transmission Chain Drives | Conveyor Traction Chains



ABS

AEC
ATLANTIC ENGINEERING CHAIN

AEC
ATLANTIC ENGINEERING CHAIN

AEC Chains emerge as a new highly engineered chain in the market and are designed to ensure the highest quality and performances. We manufacture many types of chains including Drop Forged Rivetless, Steel Roller Conveyor, Cast Combination, Welded Steel chains and others. Every AEC Chain Component is manufactured using the highest quality of steels or alloy steels.

Our chains receive a proper heat treatment in order to achieve the right balance between resistances to shock loading forces and long wear life. Our main goal is to offer our customer a personalized solution which ensures long-lasting and cost effective chains for the heavy industry applications.

The United States of America

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Int. Cl.: 6

Prior U.S. Cls.: 2, 12, 13, 14, 23, 25 and 50

Reg. No. 3,663,342
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AEC-USA
ATLANTIC ENGINEERING CHAIN

ATLANTIC BEARING SERVICES (FLORIDA LIMITED LIABILITY COMPANY)
8208 NW 30TH TERRACE
MIAMI, FL 33122

FOR: METAL ENGINEERING CHAINS, NAMELY, DROP FORGED CHAINS, RIVETLESS CHAINS, BAGASSE CARRIER CHAINS, CONVEYOR CHAINS, CAST COMBINATION CHAINS, WELDED STEEL CHAINS, REDDLER TYPE DRAGGING CHAINS, WIDE STEEL DRAG CHAIN, 900 CLASS PINTEL CHAINS, CANE CARRIER SLAT CHAINS, STANDARD PINTEL CHAINS, IN CLASS 6 (U.S. CLS. 2, 12, 13, 14, 23, 25 AND 50).

FIRST USE 3-18-2008; IN COMMERCE 3-31-2008.

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1. Power Transmission Chain Drives

1.1. Chain Transmissions and Their Chains

Atlantic Engineering Chain (AEC) manufactures and supplies short pitch precision roller chains, according to the standard American series (A series) of ANSI/ASME B29.1 and ISO 606. An exploded view of the structure of these chains is shown in Figure 1-1. The parts of the chain are: 1) Outer plate, 2) Inner plate, 3) Pin, 4) Bushing, and 5) Roller.

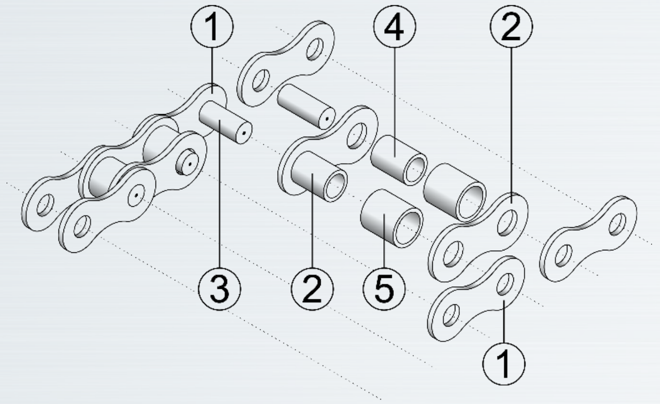

Figure 1-1. Parts of a short pitch precision roller chain

Figure 1-2. Riveted chain on a small sprocket

An inner link comprises two bushings press-fitted into the holes of two inner plates; before the assembly, a roller is slide-fitted into the outside of each bushing. On the other hand, an outer link is composed of two pins press-fitted into the holes of two outer plates; before the assembly, each pin slide-fits inside the bushing of an inner link. This way, a short chain of three links is make up. The continuation of this assembly process allows making a chain strand with any number of odd links, where both its ends are inner links. To lock in place the parts of the chain, pins ends are usually riveted, as shown in Figure 1-2. In large pitches, pins ends are locked by cotter pins instead.

To close any odd-linked strand into a continuous even-linked chain, the ends of the former may be joined with a standard additional outer link, the so-called standard connecting link. This is the strongest type of continuous chain, thanks to its fully homogeneous structure. The making of such a continuous chain is better done in the original factory, or in a workshop with appropriate tools and trained personnel. In the field, or in an unprepared workshop, it is difficult to press-fit the closing plate of a standard outer link on its pins. For these cases, a special connecting link is available, with a detachable plate whose holes slide-fit into the pins. This plate is locked in its place with a spring fastener, Figure 1-3, or by cotter pins in large pitches. When a spring fastener is used, the chain should move only in one direction: with the closed end of the fastener pointing forward. A detachable connecting link is easy to close and open, but the chain loses about 20 % of its plate fatigue strength with respect to the homogeneous chain.

If, due to unavoidable circumstances, an odd-linked continuous chain is needed, a strand with an even number of links is closed with a cranked link, as shown in Figure 1-4. Usually, the pin of this kind of link slide fits into the plates. Such a chain should move only in one direction: with the narrow section pointing forward. Besides, a chain with a cranked link loses at least 35 % of its plate fatigue strength with respect to the homogeneous chain. To tackle this problem, a special two-pitch cranked link is available. It consists of a cranked link and a normal (parallel-plate) link connected with a press-fit pin. This two-pitch cranked link may be used to close an odd-linked strand to give an odd-linked closed chain. This chain will be as strong as the connecting links used to join the two-pitch cranked link with the rest of the chain.

Figure 1-3. Connecting link with spring fastener

Figure 1-4. Cranked link with cotter pin

Pin/bushing pairs are articulations that allow every link of the chain to rotate a wide angle with respect to its immediate neighbors. Thanks to these rotations, the chain may wrap around each sprocket in the transmission, and the resulting meshing allows the transmission of motion and force. The rollers slide over the bushings, making rotational pairs. When a sprocket tooth engages with the chain, the involved roller rolls its outer surface over the tooth profile. At the same time, the inner surface of the roller slides on its bushing over a considerable support area. These actions reduce friction in the sprocket/chain meshing. In consequence, a well-installed and lubricated short pitch precision roller chain transmission has an energy efficiency between 97.5 % and 98.5 % at full load.

This class of transmission chains has the widest range of types, sizes, operating speeds and transmitted powers. Hence, they are used in many different industrial applications. Setting up a chain transmission involves a chain, a lubrication system, as well as sprockets mounted on appropriate shafts. Sometimes, accessories such as tighteners and dampers are also included. Most complex—and usually weakest—element in the transmission is the chain itself. A well-selected chain may give years of reliable service with minimum total costs, and is the first step in the design of a new power transmission or the checkup or reengineering of an existent one.

As all machine elements, chain selection must be based on its load capacity, which is limited by several mechanisms of failure. There are two basic types of failure: catastrophic and gradual. A *catastrophic failure* is a sudden, difficult to predict event that ends the working life of the element. In contrast, a *gradual failure* arrives after a long period of slow and visible deterioration, easily monitored by condition-based maintenance. Short pitch precision roller chains exhibit three modes of catastrophic failure, as follows:

- Fracture of plates, due to pulsating traction fatigue;
- Fracture of rollers and bushes, due to impact fatigue;
- Seize of pin and bush mating surfaces, due to adhesive wear (galling).

In the two first modes of catastrophic failure, the steel of the involved parts undergoes variable loads, and develops tiny fatigue cracks in the weakest points of its structure. The growth of the cracks reduces, without visible symptoms, the strength of the plates, until a sudden fragile fracture breaks the plate or the roller/bushing pair, and the whole chain itself. The third mode of catastrophic failure involves the breakdown of the lubricant film between the roller/bushing rotating pair, due to a combination of high load and speed. Then, thin oxide films that usually cover the mating steel surfaces are wiped off. Cold weld points quickly arise between the two parts, which are immediately broken. The transfer of material from one part to the other and back destroys the surfaces and generates intense heat, noise, and smoke. After a very short time, pin/bushing pairs seize, forcing the chain to break.

If catastrophic failure modes are absent, short pitch precision roller chains exhibit a mode of gradual failure: abrasive wear. Due to this physical process, the sliding surfaces of pin/bushing pairs lose material and the clearance between them increases continually. The process may proceed very slowly if the environment is clean and lubrication is appropriate, but it never stops. A contaminated environment and an improper lubrication may increase sharply the rate of wear, usually measured by lost volume of material in unit sliding length. Because of abrasive wear, chain pitch elongates with time, and pitch diameters on the sprockets get bigger and bigger. When the elongation reaches a certain point, the chain begins to jump some of the large sprocket teeth, and the transmission no longer operates properly. In addition, due to the resulting jerks, the weakened chain may be broken, if the transmission is not timely stopped.

Many years of systematic experimental research in test beds, sponsored by manufacturers' organizations like the American Chain Association (ACA) have made possible to determine reliably the rating (load capacity), of power transmission chains working in a non-contaminated environment under appropriate lubrication. In every experiment, a chain is make to transmit a mechanical power P_C while running at a linear speed v. The rating of the chain is the set of values (P_C, v) that do not give rise to galling or fatigue, the latter within a probable working life of 15,000 hours. As galling is a very fast process, it is not linked to any working life; it just happens as soon as the corresponding power/speed limit is exceeded.

In a power/speed plane with logarithmic coordinates, the load capacity or rating for a given chain is represented by the grayed region inside the jagged boundary shown in Figure 1-5. This graphical representation is known as the rating chart of the given chain. Said region is delimited by three almost straight-line segments as follows. Over the left inclined segment, the chain fails due to plate fracture. Over the right inclined segment, the chain fails due to bush/roller pair fracture. To the right of the vertical segment, the chain fails due to pin/bushing pair galling. Within the normal working region, the chain on the test bed undergoes just abrasive wear.

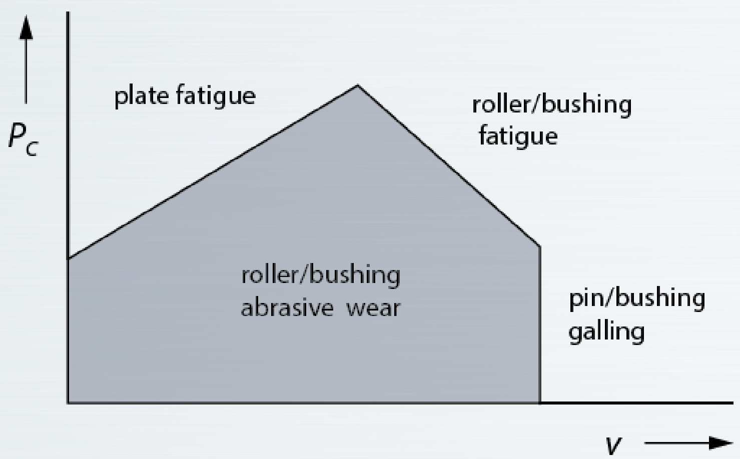

Figure 1-5. Typical rating chart of a transmission roller chain, where P_C and v are chain transmitted power and speed.

Rating charts give a valuable qualitative portray of the factors that shape the normal working zone of transmission chains. In addition, said diagrams may express numerical data on the rating. In these cases, the horizontal axis of the chart does not show chain speed, but the rotational frequency of the small sprocket used in the test bed. An ACA rating chart for the ANSI/ASME B29.1 A series of short pitch precision transmission chains is given in Figure 1-6. This chart covers AEC-USA full line of chains in this class. The number of teeth on the small sprocket for this chart is 25. The power is given in hp, while the rotational frequency scale is given in min^{-1}. ANSI chain number is given next to each individual chart.

Nevertheless, rating charts lack precision as sources of data on chain load capacity. Even with a full-page size, the resolution of a chart is poor, and the reading of the logarithmic scale is always tricky and error prone. Therefore, organizations like the ACA have traditionally published their detailed rating data in table format.

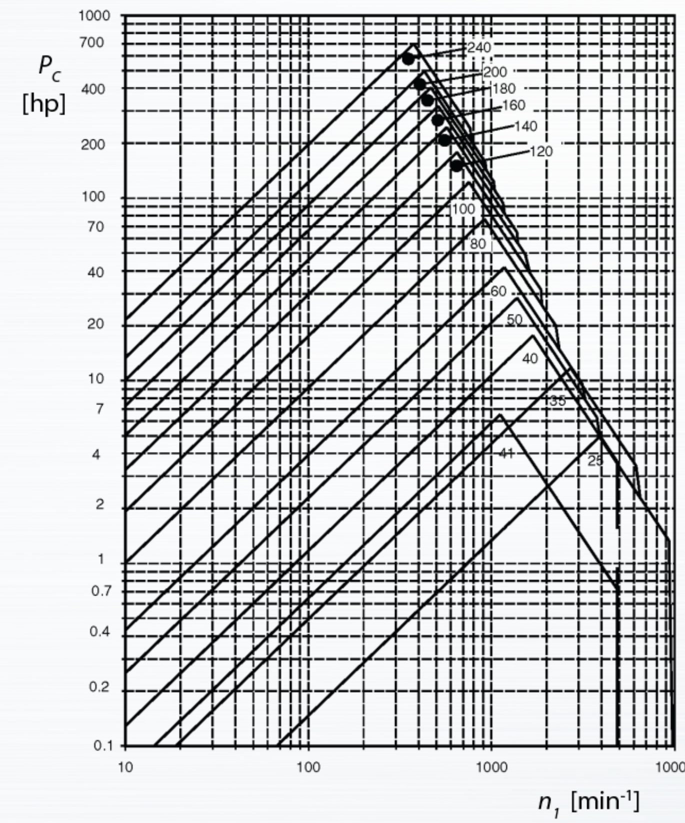

Figure 1-6. Rating chart for the ANSI/ASME B29.1 standard (A) series transmission chains (25 teeth small sprocket)

These tables give the power that may be transmitted by a given chain on the test bed, for a wide range of numbers of teeth on small sprocket, and a wide interval of closely spaced values of its rotational frequency. In addition to the rating values, the limits of application of the different chain lubrication systems are also shown in the tables by jagged line borders. The only calculation needed to use the rating tables in practical calculations is interpolation within set values of the rotational frequency. Despite their good qualities, rating tables are extensive, and take a lot of space in catalog technical sections.

Since a few years ago, the results of chain rating tests are also published as mathematical models, or simply models. In current engineering, models are sets of algebraic or numerical relations. These relations, when computed, behave in a way similar to the physical objects they represent. Currently available rating models are compact and may express the chain rating data as precisely as the tables, without any interpolation needed.

1.2. Selection of Transmission Chains

In this subsection, AEC introduces a mathematical model for short pitch precision roller chain selection. This model includes the current ACA chain rating and lubrication model, as an alternative to the low precision of rating charts and the long extension of the rating tables. The ACA model is linked here with a standard model of transmission chain kinematics and statics. The whole model is called *AEC-Power*. This model may be applied by any on-site industrial technical office equipped at least with a basic engineering or scientific calculator.

In the following pages, variables and relations of the *AEC-Power* model are given. Later, based on the same model, a procedure for chain selection in a typical practical situation is shown and applied in an example. Although short pitch precision roller chains may work at high speeds up to 20 m/s and even 30 m/s, many applications involve medium speeds ranging from 4 to 7 m/s, or low speeds, below 4 m/s. The chain selection procedure included in this subsection covers transmissions in all speed ranges expressed above, with two sprockets, horizontal shafts, center distance inclinations up to 60° and single- or multiple-strand chains from the A series of ANSI/ASME B29.1 standard. If your application involves more sprockets, or a life different from 15,000 hours, or a chain length sensibly different from 100 pitches, or inclinations nearer the vertical, or speeds close to the maximum, please refer to AEC-USA Technical Department.

Some model variables are described as "desired." The values of these variables should be careful set by the problem solver, since the choice may change substantially the results. Namely, they are:

- Desired transmission ratio i_d:
  Values from 1 to 3, give compact transmissions; from 3 to 7, become bulkier and heavier.
- Desired teeth number in small sprocket z_{1d}:
  It takes values from 11 to 25, in order of increasing chain rating and sprocket diameters.
- Desired relative center distance (in pitches) λ_d:
  It may take values as extreme as 20 and 80, although the range from 30 to 50 gives better transmissions.
- Desired center distance a_d:
  It should be specified only if strictly necessary, leaving the relative value free.

The variables of the *AEC-Power* model are given in Table 1-1. For each variable, the table shows: symbol, unit of measurement, and name. Symbols are alphabetically ordered. In calculations, all variables must be expressed in basic units of the international system (SI), since they greatly simplify formulas and computations. However, staying aware of long-standing traditions, certain variables will be expressed also using other unit systems, always within parentheses.

Table 1-1. Variables of the *AEC-Power* model

Symbol	*Unit*	*Name*
a	m	Center distance
$ANSIchainNo$		ANSI/ASME B29.1 standard chain number
a_d	m	Desired center distance
$ConnLink$		Connection link closing the chain (standard, detachable, cranked)
D_1	m	Pitch diameter of small sprocket
D_2	m	Pitch diameter of large sprocket
D_{f1}	m	Foot diameter of small sprocket
D_{f2}	m	Foot diameter of large sprocket
D_{a1}	m	Tip diameter of small sprocket
D_{a2}	m	Tip diameter of large sprocket
D_{H1max}	m	Maximum hub diameter of small sprocket
D_{H2max}	m	Maximum hub diameter of large sprocket
d_1	m	Chain roller diameter
F_R	N	Resultant force on the loaded chain strand
F_C	N	Tension force at the ends of chain catenary
F_t	N	Nominal chain pull force
F_S	N	Force on both sprocket shafts
F_v	N	Centrifugal force on the chain
f	m	Sag of unloaded chain strand
f_{rel}		Relative sag of unloaded chain strand
g	m/s^2	Acceleration of gravity
i		Transmission speed ratio
Δi		Slack in transmission speed ratio
i_d		Desired transmission speed ratio
K_A		Application factor
K_C		Factor of connecting link
K_G		Rating factor of pin/bushing galling
K_{GA}		Rating factor of lubrication type A
K_{GB}		Rating factor of lubrication type B
K_N		Multi-strand chain factor
K_P		Rating factor of plate fatigue
K_R		Rating factor of roller/bushing impact fatigue
l	m	Length of taught chain strand
$Load$		Type of load imposed by driven machine
$LubType$		Minimum lubrication type needed (A, B, C)
$Machine$		Type of driven machine (Class & Subclass)
n_1	Hz (min^{-1})	Rotational frequency of small (driver) sprocket
n_{1r}		Relative rotational frequency of small sprocket
n_{1b}	Hz (min^{-1})	Basic rotational frequency of small sprocket
n_2	Hz (min^{-1})	Rotational frequency of large (driven) sprocket
n_{2d}	Hz (min^{-1})	Desired rotational frequency of large sprocket
N		Number of chain strands (from 1 to 10)
p	m (in)	Chain pitch
p_r		Relative chain pitch
p_b	m (in)	Basic chain pitch
p_0	m (in)	Unrounded value of chain pitch
P_P	W	Chain rating limited by plate fatigue
P_R	W	Chain rating limited by roller/bushing impact fatigue
P_G	W	Chain rating limited by pin/bushing galling

6

Symbol	Unit	Name
P_{GA}	W	Chain rating limited by lubrication type A
P_{GB}	W	Chain rating limited by lubrication type B
P_C	W	Mechanical power allowed by chain catastrophic failure modes
P_{C0}	W	Initial value for P_C
P_1	W	Mechanical power at small (driver) sprocket shaft
P_2	W	Mechanical power at large (driven) sprocket shaft
$PrimeMover$		Type of prime mover
q	kg/m	Chain mass per unit length
s	m	Length of chain catenary on unloaded strand
T_1	N·m	Torque on small (driver) sprocket shaft
T_2	N·m	Torque of large (driven) sprocket shaft
v	m/s	Chain speed
X		Relative length of chain (in pitches)
X_0		Unrounded relative length of chain (in pitches)
z_{1d}		Desired teeth number on small sprocket
z_1		Teeth number on small sprocket
z_2		Teeth number on large sprocket
δ	rad (…°)	Center distance inclination with the horizontal (from 0 to 60°)
ε_{max}	%	Relative elongation of chain allowed by large sprocket
φ	rad	Unloaded chain branch chord inclination with the center distance
η		Energy efficiency of transmission
λ		Relative center distance (in pitches)
λ_d		Desired relative center distance (in pitches)
$\Delta\lambda$		Slack in center distance in pitches
τ_1	rad	Angular pitch of small sprocket
τ_2	rad	Angular pitch of large sprocket
ψ	rad	Unloaded chain branch chord inclination with the horizontal
Λ	m	Chain length

The algebraic relations included in the *AEC-Power* model are given in Table 1-2. For each relation, the following information is shown: relation, number, and comment. Units of measurement are SI basic ones in all cases, including the ACA rating and lubrication model, originally developed for customary units of length, rotational frequency, and power. The values of constants included in some relations are usually given in the related Comments field.

Table 1-2. Relations of the *AEC-Power* model

Relation	Number	Comments
$P_{C0} = K_{Amin} \cdot P_1$	(1)	
$K_A = P_C/P_1$	(2)	
$P_C = \min(P_P, P_R, P_G)$	(3)	
$p_0 = p_b \sqrt[3]{P_{C0}/(K_N \cdot K_C \cdot 3.281 \cdot z_1 \cdot n_{1r}^{0.96})}$	(4)	
$P_P = K_N \cdot K_C \cdot K_P \cdot z_1 \cdot n_{1r}^{0.96} \cdot p_r^{(3.0-0.07 \cdot p_r)}$	(5)	
if $(ANSIchainNo = 41)$ then $K_P = 1.805$ W if $(ANSIchainNo \neq 41)$ then $K_P = 3.281$ W	(6)	
$P_R = K_N \cdot K_C \cdot K_R \cdot z_1^{0.5} \cdot n_{1r}^{-1.5} \cdot p_r^{0.8}$	(7)	
if $(p = 0.00635$ or $p = 0.009525)$ then $K_R = 1.27 \cdot 10^7$ W if $(p \neq 0.00635$ and $p \neq 0.009525)$ then $K_R = 2.16 \cdot 10^7$ W	(8)	
$P_G = K_N \cdot (\, K_G \cdot z_1 \cdot p_r^2 - K_{G0} \cdot z_1^3 \cdot n_{1r}^3 \cdot p_r^5 \cdot (2 + 0.0323 \cdot z_1)\,)$	(9)	$K_G = 4811$ W $K_{G0} = 1.883 \cdot 10^{-10}$ W
$P_{GA} = K_N \cdot (\, K_{GA} \cdot z_1 \cdot p_r^2 - K_{G0} \cdot z_1^3 \cdot n_{1r}^3 \cdot p_r^5 \cdot (2 + 0.0323 \cdot z_1)\,)$	(10)	$K_{GA} = 240.6$ W
$P_{GB} = K_N \cdot (\, K_{GB} \cdot z_1 \cdot p_r^2 - K_{G0} \cdot z_1^3 \cdot n_{1r}^3 \cdot p_r^5 \cdot (2 + 0.0323 \cdot z_1)\,)$	(11)	$K_{GB} = 2406$ W
if $(P_C \leq P_{GA})$ then $LubType = A$ if $(P_C > P_{GA}$ and $P_C \leq P_{GB})$ then $LubType = B$ if $(P_C > P_{GB})$ then $LubType = C$	(12)	
$K_N = \text{Table3}(N)$	(13)	See Table 3
$p = \text{nearest}(p_0)$	(14)	In chain tables. Get also chain number
$i_d = n_1/n_{2d}$	(15)	
$i = z_2/z_1$	(16)	
$i_d + \Delta i = z_2/z_1$	(17)	
$\tau_1 = 2 \cdot \pi/z_1$	(18)	
$\tau_2 = 2 \cdot \pi/z_2$	(19)	
$D_1 = p/sin(\tau_1/2)$	(20)	
$D_2 = p/\ sin(\tau_2/2)$	(21)	
$Load = Table\ 5(Machine)$	(22)	See Table 5
$K_{Amin} = Table\ 4(PrimeMover, Load)$	(23)	See Table 4
if $(ConnLink = \text{standard})$ $then$ $K_C = 1$ else if $(ConnLink = \text{detachable})$ $then$ $K_C = 0.8$ else if $(ConnLink = \text{cranked})$ $then$ $K_C = 0.65$	(24)	
$\lambda_d = \lambda + \Delta\lambda$	(25)	
$\lambda = a/p$	(26)	
$X_0 = 2 \cdot \lambda_d + ((z_1 + z_2)/2) + ((z_2 - z_1)^2)\ /\ (4 \cdot \pi^2 \cdot \lambda_d)\ )$	(27)	

Relation	*Number*	*Comments*
$X = \mathrm{round}(X_0)$	(28)	To next even integer, avoiding the need for a cranked connecting link
$a = (p/4) \cdot \left[X - ((z_1 + z_2)/2) + \sqrt{(X - ((z_1 + z_2)/2))^2 - 2 \cdot ((z_2 - z_1)/\pi)^2}\right]$	(29)	
$\Lambda = X \cdot p$	(30)	
$v = \pi \cdot n_1 \cdot D_1$	(31)	
$F_t = P_1/v$	(32)	
$T_1 = F_t \cdot d_1/2$	(33)	
$T_2 = T_1 \cdot i \cdot \eta$	(34)	$\eta \approx 0.98$
$F_v = q \cdot v^2$	(35)	
$\varphi = \sin^{-1}[(d_2 - d_1)/(2 \cdot a)]$	(36)	
$l = a \cdot \cos(\varphi)$	(37)	
$q = q(p)$	(38)	In chain tables
$f = f_{rel} \cdot l$	(39)	$f_{rel} = 0.02$
$s = l \cdot \sqrt{\frac{16}{3} \cdot f_{rel}^2 + 1}$	(40)	
$F_C = (q \cdot g/12) \cdot \left[\left(s^2/(8 \cdot f) + (f/2)\right)\right]$	(41)	
$F_R = K_A \cdot F_t + F_v + F_C$	(42)	
$F_S = K_A \cdot F_t + 2 \cdot F_C$	(43)	
$\varepsilon_{max} = 200/z_2$	(44)	
$i = n_1/n_2$	(45)	
$p = p_r \cdot p_b$	(46)	$p_b = 0.0254\ \mathrm{m}$
$n_1 = n_{1r} \cdot n_{1b}$	(47)	$n_{1b} = (1/60)\ \mathrm{Hz}$
$\lambda_d = a_d/p$	(48)	
$P_2 = \eta \cdot P_1$	(49)	$\eta \approx 0.98$
$D_{f1} = D_1 - d1$	(50)	
$D_{f2} = D_2 - d1$	(51)	
$D_{a1} = D_1 \cdot \cos(\tau_1/2) + 0.6 \cdot p$	(52)	
$D_{a2} = D_2 \cdot \cos(\tau_2/2) + 0.6 \cdot p$	(53)	
$D_{H1max} = D_1 - (p + 0.00076)$	(54)	
$D_{H2max} = D_2 - (p + 0.00076)$	(55)	

The following three tables are part of the AEC-Power model. Table 1-3 gives the values of the multi-strand factor for chains from one to ten strands. Table 1-5 gives the ACA minimum service factor of the chain transmission in function of the type of prime mover, and the type of load imposed by the driven machine. The information on the ACA type of load of the driven machine, needed in Table 1-5, may be found in Table 1-4 for a wide selection of driven machines.

Table 1-3. Multi-strand chain factors for roller chain transmissions, according to industry standard practice

Number of chain strands, N	Multi-strand chain factor, K_N	Number of chain strands, N	Multi-strand chain factor, K_N
1	1	5	3.9
2	1.7	6	4.6
3	2.5	8	6.2
4	3.3	10	7.5

Table 1-4. Types of loads imposed by driven machines, according to ACA

Driven machine Class	*Type of load*		
	Smooth load	*Moderate shock load*	*Heavy shock load*
	Subclass 1	*Subclass 2*	*Subclass 3*
1. Agitators	Pure liquid		
2. Beaters		All	
3. Blowers	Centrifugal		
4. Boat propellers			All
5. Bucket elevators	Uniformly loaded or fed	Not uniformly loaded or fed	
6. Clay working machinery		Pug mills	Brick presses, briquetting machines
7. Compressors		Centrifugal, reciprocating (3+ cylinders)	Reciprocating (1, 2 cylinders)
8. Conveyors	Uniformly loaded or fed	Heavy duty, not uniformly loaded	Reciprocating and shaker
9. Cranes and hoists		Medium duty, skip hoists (travel & trolley motion)	Heavy duty, logging, and rotary drilling
10. Dredgers		Cable, reel, and conveyor drives	Cutter head, jig, and screen drives
11. Feeders	Rotary table	Apron, screw, rotary vane	Reciprocating, shaker
12. Generators	All		
13. Machine tools	Drills, grinders, lathes	Boring mills, milling machines, hobs, shapers	Punch presses, shears, plate planers, cold formers
14. Mills		Ball, pebble, and tube	Draw benches, hammer, rolling, wire drawing
15. Paper machinery		Pulp grinders	Calenders, mixers, sheeters

16. Printing presses			All
17. Pumps	Centrifugal	Reciprocating (3+ cylinders)	Reciprocating (1, 2 cylinders)
18. Screens	Rotary, uniformly fed		
19. Textile machinery		Calenders, mangles, nappers	Carding machinery
20. Woodworking machines		All	

Table 1-5. Minimum service factors for roller chain transmissions, according to ACA

Type of load imposed by driven machine	Type of prime mover		
	Reciprocating engine with hydraulic transmission	Electric motor or turbine	Reciprocating engine with mechanical transmission
	Minimum service factor, K_{Amin}		
Smooth	1.0	1.0	1.2
Moderate shock	1.2	1.3	1.4
Heavy shock	1.4	1.5	1.7

Short pitch precision roller chain selections for speed reducing transmissions are everyday problems in industrial drive engineering. Problems in this class may involve design of a new transmission, or the checking/reengineering of an existing one. An engineering problem is well posed when both data (known values) and query (unknowns) are clearly identified. Each compatible data/query combination defines a different problem.

To solve a problem is to find the values of the variables listed in the query. A problem is usually solved using an algorithm, this is, an ordered set of relations from a model of the system, where the unknown values are calculated from the data and the previously calculated unknowns.

A typical example problem in chain selection is posed as follows, to illustrate the application of the *AEC-Power* mathematical model given above.

Problem Data:

1. Driven machine: $Machine$ = belt conveyor, heavy duty.
2. Prime mover: $PrimeMover$ = geared motor.
3. Power needed by the machine: P_2 = 11,300 W.
4. Rotational frequency of the high-speed (driver) shaft: n_1= 2.88 Hz (173 min^{-1}).
5. Desired rotational frequency of the low-speed (driven) shaft: n_{2d} = 0.983 Hz (59 min^{-1}).
6. Inclination of the center distance with respect to the horizontal: $\delta = \pi/6$ rad (30°);
7. A chain with a detachable connecting link is to be used.
8. Desired relative center distance: λ_d = 40.
9. Desired teeth number in small sprocket: z_{1d} = 25

Problem Query:

1. Minimum chain application factor: K_{Amin}
2. Chain application factor: K_A
3. Number of teeth in small sprocket: z_1
4. Number of teeth in large sprocket: z_2
5. Chain pitch: p in m (in);
6. Relative chain length: X
7. Center distance: a in m;
8. Rotational frequency of small sprocket: n_2 in Hz (min^{-1})
9. Power to be taken from the prime mover: P_1 in W
10. Torque on small sprocket: T_1 in N·m
11. Torque on large sprocket: T_2 in N·m
12. Radial force on both shafts: F_S in N
13. Type of lubrication of the chain: $LubType$
14. Maximum relative allowable wear elongation of the chain: ε_{max} in %

To solve the above-posed problem, please apply the algorithm developed in Table 1-6. Following the steps of the algorithm, you will find that in each step the concerned relation has only one unknown. The value of this unknown may be directly obtained from the relation with an appropriate calculator. If checking in step 20 is not satisfied, return to step 13 and try with next chain size pitch.

Table 1-6. Algorithm to solve posed example problem

Step	*Relation*	*Unknown*	*Solution*	*Comments*
1	(15)	i_d	2.932	
2	here	z_1, z_2	2.932 ≈ 293/100 ≈ 73/25 = z_2/z_1	Just fits desired z_1
3	(16)	i	2.920	
4	(17)	Δi	-0.0122	
5	(22)	$Load$	Moderate shock	Class 8, Subclass 2
6	(23)	K_{Amin}	1.3	
7	(49)	P_1	11,500 W	
8	(24)	K_C	0.8	
9	(1)	P_{C0}	15,000 W	
10	(13)	K_N	1	
11	(47)	n_{1r}	173	
12	(4)	p_0	~ 0.0298 m (1.173 in)	
13	(14)	p	0.03175 m (1.250 in) ANSI chain number: 100	In AEC chain catalog
14	(46)	p_r	1.25	
15	(6)	K_P	3.281 W	
16	(5)	P_P	17,700 W	
17	(8)	K_R	2.16·10^7 W	

2. Conveyor Traction Chains

Atlantic Engineering Chain (AEC) manufactures and supplies a variety of conveyor chains, also known as engineered or engineering chains. These chains are used as traction elements in heavy-duty conveyors of cane sugar mills, palm oil mills, mines, and other industries. Environment and function of conveyor chains differ a great deal from those of transmission chains, covered in the first part of this Technical Section. Instead of lubricant oil, conveyor chains are often immersed in dust, mud, fibers, or liquids that are normal constituents of the carried product. By nature, conveyors may be long, and the weight of the chains becomes a significant part of the total mass in movement. Therefore, engineering chains should be strong, light, and inexpensive at the same time.

2.1. Chains with Rollers or Bushings

In very small conveyors working in clean environments, a modified single-pitch roller transmission chain is sometimes used, but most industrial conveyor chains are double-pitched. The use of double-pitch links reduces the mass and cost of conveyor chains, decreasing the number of articulations per unit length. On the positive side, double pitches allow more space for plate attachments. These attachments are bolted to slats, flights, vanes, scrapers, aprons, or buckets that carry or push the conveyed product along the conveyor trace.

Regrettably, double pitches make sprockets get bigger. This may be partially compensated dropping sprocket teeth numbers, in some cases down to six. However, sprockets with small teeth numbers display a strong polygonal effect. Such effect excites vibrations that increase stresses in conveyor chain and drive. Striving to compensate this phenomenon, chain speeds are set low: seldom over 1.0 m/s, often below 0.50 m/s, or even 0.050 m/s. However, very low speeds of sliding—under 0.035 m/s—may also induce vibrations due to stick-slip. This phenomenon is characterized by a back and forth transition between static and kinetic friction, when elastic strain of the chain may put some points of it shortly in a zero-speed condition. Surprisingly, stick-slip may happen earlier in the bush/roller sliding pairs of large roller chains, because such pairs slide at a speed well below of chain speed.

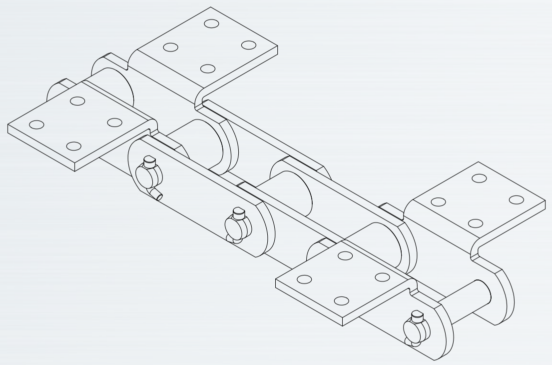

Figure 2-1. Straight-plates chain with small rollers

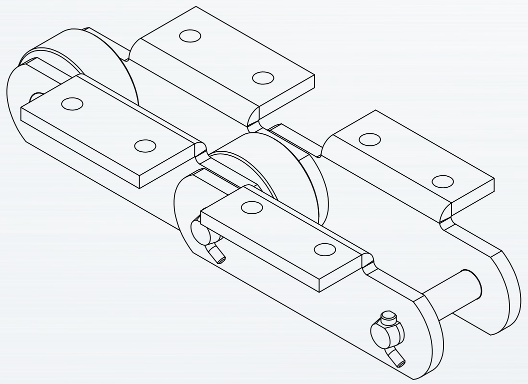

Figure 2-2. Straight-plates chain with large rollers

Many conveyor chains are straight-plate roller chains according to ANSI/ASME B29.4, as shown in Figure 2-1. These chains use the same bushings and pins as ANSI/ASME B29.1 drive chains, with double length plates. There is an option with small diameter rollers, Figure 2-1. These small rollers allow chain plates to slide on a wide support rail. An alternative option provides large rollers, Figure 2-2, whose diameters exceed plate height. This way, chain may roll on a narrow support rail of the conveyor. This reduces force and power necessary to pull the chain, although the chain itself becomes somewhat heavier. Large rollers may be plain (as small rollers do) or may have a flange similar to railroad car wheels.

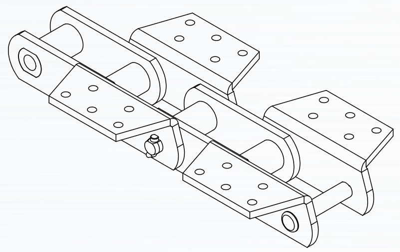

Figure 2-3. Straight-plates rollerless (bushed) chain

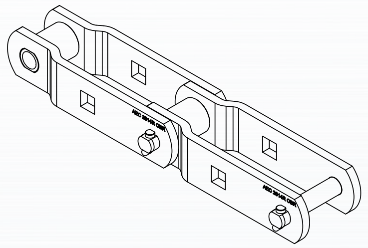

Figure 2-4. Cranked-plates rollerless chain for outboard rollers

When the conveyed product is a fine and abrasive dust, like cement, or milled coal, it may be stuck in the roller/bushing pair gap. Consequently, the roller stops rotating, and its function is nullified. In these cases, bushed (rollerless) chains as shown in Figure 2-3 are a rational solution, due to its lower cost. The bushings of these chains have the same outer diameter as the rollers of ANSI/ASME B29.1 drive chains, so standard sprocket teeth may mesh normally with them.

As an alternative, large rollers with sealed sliding or rolling bearings may be applied. Large rollers inside the chain require special sprockets, but outboard rollers allow the use of standard sprockets. Figure 2-4 shows a chain intended to carry outboard rollers, whose installation is depicted in the exploded view of Figure 2-5. These chains are applied in sugar mill conveyors where the cane, just arrived from the fields, comes along with abrasive soil dust. It is easy to see that the chain in Figure 2-4 has cranked plates. This kind of chain may have any number of links, even or odd, because it is assembled from a single type of link: no special connecting links are needed.

A new type of conveyor chain is the hollow-pin chain, Figure 2-6, as per ANSI/ASME B29.27 standard. This chain is lightweight and strong, because plates do not have attachments or holes, beyond articulation ones. Parts to be fastened to the chain may be held by bolts passing through the hollow pins. Hollow pins have the same outer diameter as the bushings in ANSI/ASME B29.1 series chains. Pins are locked to outside plates by riveting, elastic clips, or cotter pins. The bushings have the same outer diameter as the rollers in ANSI/ASME B29.1 series chains. Therefore, “small roller” variants of these chains are in fact bushed chains. Large roller variants of hollow-pin chains have rollers with the same outer diameters as ANSI/ASME B29.4 large roller series chains. When maximum strength is asked from ANSI/ASME B29.27 chains, the standard covers a solid-pin variant, shown in Figure 2-7. These pins may have outboard extensions—plain or threaded—of maximum diameter. The remaining parts are identical to the hollow-pin variant.

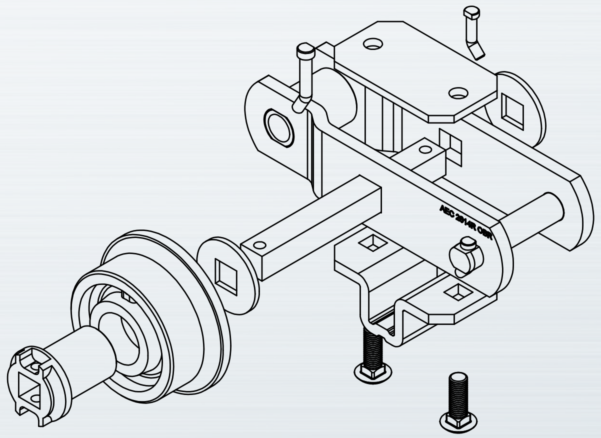

Figure 2-5. Large flanged outboard roller installed on a chain link

Figure 2-6. Hollow-pin roller chain

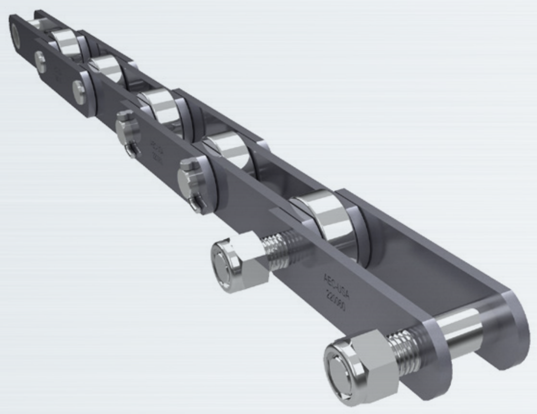

Figure 2-7. ANSI/ASME B29.27 chain with solid pins

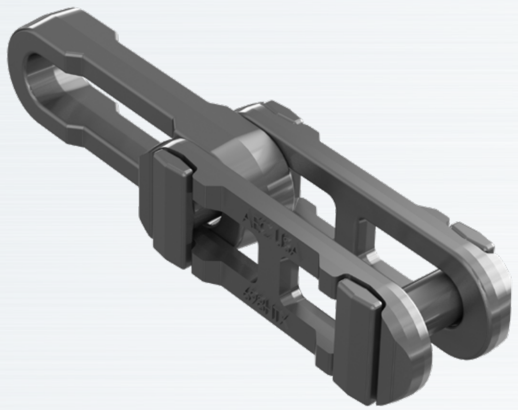

Figure 2-8. Dropforged rivetless chain

More types of roller or bushed chains may be found in the pages of this catalog, which solve specific chain conveyor problems in sugar mill, mining, and other industries. All chains belonging to this first group share a set of common characteristics, due to the cylindrical form of their rollers and bushings, which mesh with sprocket teeth.

2.2. Chains without Rollers or Bushings

A second group of conveyor chains in this catalog cannot be included in the above-mentioned group, because driver sprocket teeth do not mesh with rollers or bushings, but with the link plates. Link plates in these chains have partial or complete external cylindrical surfaces, often with a slight crowning. These external cylindrical surfaces mesh with driver sprocket teeth, to transfer movement from the latter to the chain. These chains do not have rollers or bushings, only pins that never get in contact with sprocket teeth.

One example from this group is the dropforged rivetless chain, Figure 2-8, as per ANSI/ASME B29.22 standard. An outer link of this chain has two outer plates and two pins, while an inner link has a single hollow inner plate. As shown in Figure 2-8, plate ends have external semi-cylindrical surfaces with crowning. Big radii of curvature and crowning of these surfaces, allow a well-localized contact with sprocket teeth, despite their modest dimensional precision, assuring a satisfactory abrasive wear life. In addition, chain/sprocket mesh is fully open, and not prone to the packing of dust or fibers.

Driver sprocket for this type of chain is usually a single sprocket with teeth that mesh with the ends of the inner links. Accordingly, this sprocket only meshes with every other chain link. Two other driver sprocket designs exist; however, they are seldom applied due to their complexity. Certain applications do not drive this type of chain using sprockets, but so-called caterpillar drives: auxiliary power transmission chains with special dogs that engage voids between consecutive outer plates of the chain. Dropforged rivetless chains are strong and light, and may be assembled or dismantled without tools. The introduction of these chains in 1919 revolutionized automobile assembly lines, and is the best solution still today. Currently, these chains are used also in many other applications, from wash and feed cane conveyors and overhead big bag sugar conveyors in sugar mills, to heavy carbon anode handling in aluminum industry plants.

Other chains that belong to the second group are 900 Class Pintle Chains, Figure 2-9, widely used in intermediate carrier slat conveyors of sugar mills, which move crushed sugar cane from one roller mill to next in the tandem row. Made of bronze, the links of these chains have lateral cylindrical barrels. The sides of these barrels mesh with a double sprocket teeth, in an open gear pair where juice soaked bagasse fibers cannot build-up and clog the mechanism. Pins are made of carbon or stainless steel. To prolong the life of the costly bronze links, their pinholes are usually protected with bushings. These steel bushes may be replaced when worn, allowing the bronze links to be used more than once. However, in no way this is a bushed chain, because such protective bushings never take part in the mesh with sprocket teeth.

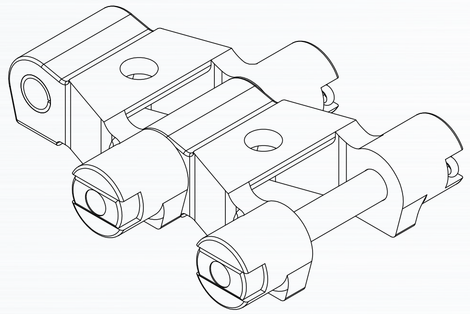

Figure 2-9. 900 class pintle chain

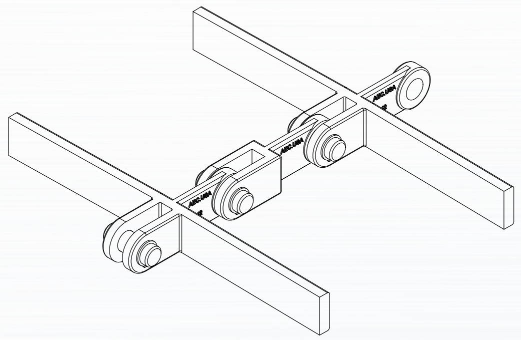

Figure 2-10. Forked link of a Redler chain

Redler chains are made of forked links, similar to the one shown in Figure 2-10. These links have a single plate, forked in the front end, and with a yoke at the back end. The holes of the fork articulate with the yoke of the preceding link through a short pin, usually locked by an elastic ring or a safety nut. Flights of different configurations may be welded to the links or bolted to the pins. Made of hardened steel, these chains are key components of the so-called *en-masse* drag conveyors. They are able to transport a wide variety of dry bulk products, in hermetically closed casings, along combined horizontal, inclined, and vertical traces. This chain is driven by double sprockets, which mesh with flat or semi cylindrical back-of-the-fork surfaces of every link.

2.3. Chain Conveyors and Elevators

In chain conveyors, one or more traction chains pull the load to move it from one point to other. However, this basic function may be done in different ways. According with the aims of this section, chain conveyors may be divided into two groups, as follows.

1. Conveyors where the traction chains carry the load;
2. Conveyors where the traction chains do not carry the load.

In the first group, conveyor load is carried on troughs, trays, buckets, or hangers, bolted to and moving together with, the chains. In the second group, the load is pushed by flights attached to the traction chains, which force it to slide in a stationary trough or other support media. Table 2-1 shows five typical examples of the two groups of conveyors, adding details about the type of load handled and the load supporting and pushing elements.

Table 2-1. Basic groups of chain conveyors, its characteristics, and examples

Conveyor group	*Traction chains carry the load*	*Type of load*	*Load support*	*Load pusher*	*Examples*
1	Yes	Bulk	Mobile troughs or trays	None	Apron conveyors
			Buckets		Bucket elevators
		Unit	Hangers		Hanging conveyors
2	No	Bulk	Fixed troughs	Flights	Drag conveyors
			Load floors		Scraper conveyors

Figure 2-11. Apron conveyor carrying hot ore in a mining plant

Figure 2-12. Drag conveyor carrying bagasse

As the apron conveyor shown in Figure 2-11, conveyors of the first group do not slide the load over its support. In this case, the ore is supported by a mobile trough. The bottom and walls of the trough are made of steel slats and side plates, bolted on the two traction chains located beneath the trough. Observe that these chains have outboard rollers that run on steel rails. Therefore, the friction between ore and its support is static, which does not develop resistive power. Only the friction between the traction chains and their support rails should be taken into account to determine the opposition to movement of the conveyor.

Conveyors of the second group slide the load on its support, as the drag conveyor shown in Figure 2-12. In this case, conveyor flights attached to the traction chains push the bagasse along a fixed trough. Consequently, the friction between both media develops resistive power, which should be computed, and added to the produced by the friction of traction chains against their support rails. As a result, the calculation of the opposition to movement of the conveyor is done with different formulas. Traction chains of the conveyor shown have internal large rollers that run on steel rails.

Scraper conveyors, like the one shown in Figure 2-13, belong to the second group because the load is not carried by the traction chains. Instead, conveyor flights scrap directly on the heap of material to be conveyed. In this case, the gypsum scrapped by the conveyor flights slides over the layers of gypsum located below it.

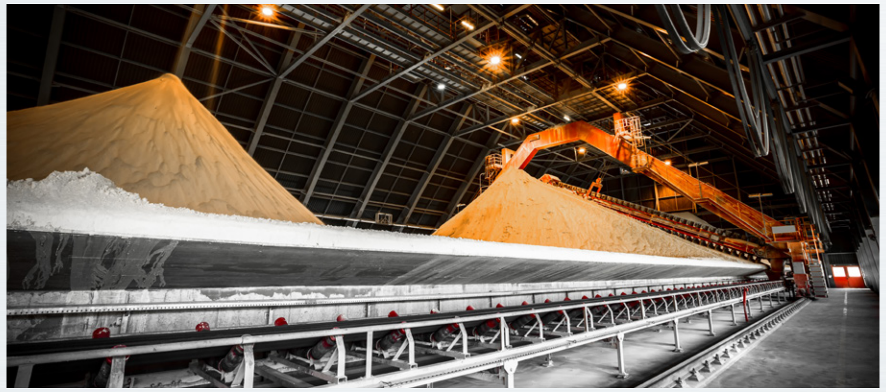

Figure 2-13. Scraper conveyor reclaiming gypsum in a storage hall

Many conveyors have a single, straight trace, with a given inclination, as shown in Figure 2-14. It is easy to see that a single trace may be horizontal, inclined, or vertical. Vertical conveyors are also called elevators. The general case is the inclined conveyor, which reduces to one of the two particular cases when the inclination is null or maximum. The total length of a conveyor is usually given by the center distance a between traction chain sprockets. Besides, an inclined conveyor has a horizontal length L and a height H, as seen in Figure 2-14. A horizontal conveyor has no height, and an elevator has no horizontal length. In the same figure, you may observe that horizontal and inclined conveyors need support for the traction chains (illustrated as a line with curved ends) both in the upper and in the lower branches. Elevator chains usually do not need said support. Very short horizontal or nearly horizontal conveyors may not have support for the lower branch, which simply hangs from its sprockets, taking a natural catenary shape.

A number of conveyors have a complex trace, composed of two or more straight segments, with different inclinations. Nevertheless, they may be decomposed into single segments to simplify their analysis.

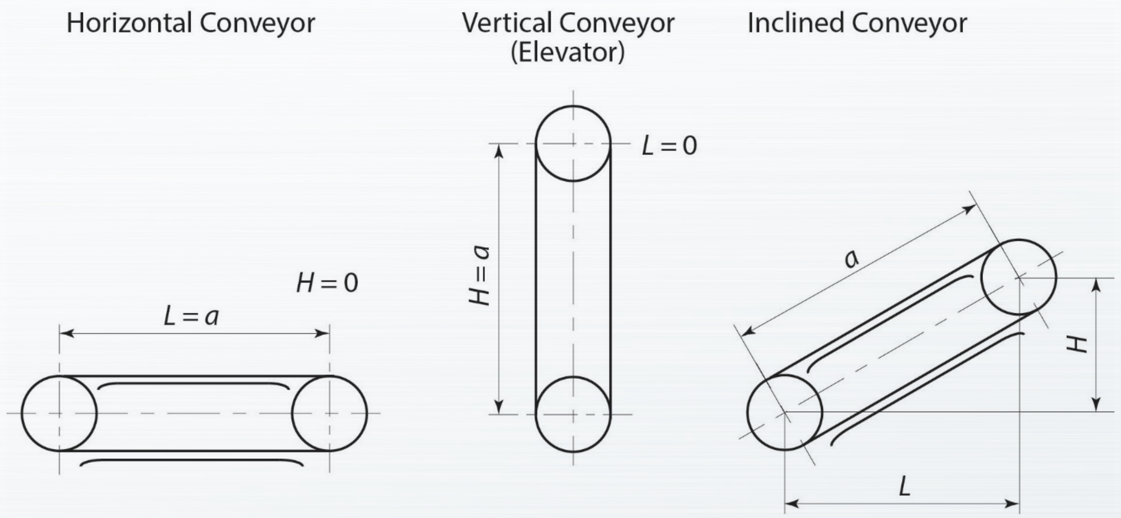

Figure 2-14. Inclination cases of single, straight trace conveyors

2.4. Capacity of Chain Conveyors

Single, straight trace conveyors have two branches. Usually, one branch is loaded and the other is unloaded, so the capacity of the loaded branch will be the conveyor capacity. However, there are conveyors where both branches may be loaded, totally, or only in sectors of its length.

The capacity of a loaded branch is usually expressed by its mass capacity, given by Equation 2-1.

$$C_m = q_L \cdot v \qquad \text{Equation 2-1}$$

Where:

C_m is the mass capacity of the loaded branch [kg/s].
q_L is the load mass per unit length of the loaded branch [kg/m].
v is the speed of the conveyor chain [m/s].

Sometimes, the capacity of a loaded branch is expressed by its volume capacity, given by Equation 2-2.

$$C_V = C_m / \rho \qquad \text{Equation 2-2}$$

Where:

C_V is the volume capacity of the loaded branch [m^3/s].
C_m is the mass capacity of the loaded branch [kg/s].
ρ is the apparent density of the load [kg/m^3].

2.5. Resistance to motion in conveyor branches

To develop intended capacity, a conveyor needs to move their loaded and unloaded branches, against or in favor of the weight forces and the friction forces acting on the load, chains, and other parts. This subsection gives formulas for the calculation of the force of resistance to motion, R, of conveyor branches under different cases of loading. This force should be provided by the traction chains at the pulling end of the involved branch. The pulling end of a branch is the one pointed by the direction of its speed of movement. If $R > 0$, this means that the branch increases tension of the preceding sectors of the traction chains. On the contrary, If $R < 0$, this means that the branch decreases tension of the preceding sectors of the traction chains.

2.5.1 Generalities about resistance to motion

The friction factor f_C for chains that slide against their support rails is given in Table 2-2.

Table 2-2. Friction factor for sliding chains

Rail material	f_C	
	With bad lubrication	*With good lubrication*
Steel	0.30 — 0.50	0.25
Polyamide or High Density Polyethylene	0.40	0.15
Hardwood	0.50	0.30

The friction factor f_C for chains that roll against their support rails is given in Table 2-3.

Table 2-3. Friction factor for rolling chains

Roller/bushing bearing	f_C	
	With bad lubrication	*With good lubrication*
Sliding bearing	0.08 — 0.13	0.06 – 0.10
Rolling bearing	0.035 — 0.045	0.020 – 0.030

The apparent densities ρ and angles of repose ϕ_r of a number of bulk load materials, as well as the friction factors f_L of the same sliding on steel, are given in Table 2-4.

Table 2-4. Apparent density, repose angle, and friction factor against steel of load materials

Load material	ρ [kg/m^3]	ϕ_r [°]	f_L [1]
Alumina	676 — 826	50	0.36
Ashes, dry	570 — 650	45 — 48	0.50
Ashes, wet	730 — 810	48 — 55	0.60
Bagasse, moist	200	51	0.35 — 0.45
Bauxite	1 200 — 1 360	31 — 33	0.65
Cement, clinker	1 209 - 1 590	30	0.70
Cement, Portland	1 150 – 1 540	20	0.65
Coke	500	30	0.55
Cooper ore	2 510 — 2 830	35	0.53
Gravel, dry	1 520	35 — 40	0.45
Gravel, with sand	2 000	25 — 30	0.60
Iron ore	3 610	35	0.64
Sand, moist	1 960	35	0.85
Sand, wet	2 080	25	0.60
Sugar cane, knifed	240 — 288	45	0.40
Sugar, granulated dry	801	30 — 35	0.60
Sugar, raw	960	34 — 40	0.45
Wood chips, dry	240 - 520	45	0.40
Zinc ore	2 560	35	0.45

A measure of how heavily loaded a conveyor is, may be given by the load mass per unit length of its loaded branch. In some conveyors, the load is continuous and laterally unbounded. In these cases, the load mass per unit length of loaded branch is given by Equation 2-3.

$$q_L = A \cdot \varphi_A \cdot \rho \qquad \textbf{Equation 2-3}$$

Where:

q_L is the load mass per unit length of the loaded branch [kg/m].
A is the nominal transversal area of the continuous laterally unbounded load flow [m^2].
φ_A is the mean transversal area filling factor of the continuous load flow [1].
ρ is the apparent density of the load [kg/m^3].

In some conveyors, the load is supported by a trough, whose lateral retaining walls avoid material spills beyond their limits. If the load is continuous and laterally bounded by retaining walls, the load mass per unit length of loaded branch is given by Equation 2-4.

$$q_L = B \cdot W \cdot \varphi_A \cdot \rho \qquad \textbf{Equation 2-4}$$

Where:

q_L is the load mass per unit length of the loaded branch [kg/m].
B is the width of the load trough [m].
W is the height of the walls of the load trough [m].
φ_A is the mean transversal area filling factor of the load trough [1].
ρ is the apparent density of the load [kg/m³].

If the load is discontinuous, the load mass per unit length of loaded branch is given by Equation 2-5.

$$q_L = (V/l_V) \cdot \varphi_V \cdot \rho$$ **Equation 2-5**

Where:

q_L is the load mass per unit length of the loaded branch [kg/m].
V is the volume of each load container [m³].
l_V is the pitch of the load containers along the length of the loaded branch [m].
φ_V is the mean volume filling factor of load containers [1].
ρ is the apparent density of the load [kg/m³].

The resistance to motion of an unloaded conveyor branch moving downwards is given by Equation 2-6.

$$R_U = q_C \cdot g \cdot (L \cdot f_C - H)$$ **Equation 2-6**

Where:

R_U is the force of resistance to motion of the unloaded branch [N].
q_C is the chain mass per unit length [kg/m], including attachments and load support, pushing or scraping elements.
L is the horizontal length of conveyor [m].
g is the acceleration of gravity, 9.81 m/s².
f_C is the friction factor of the chain running on its support rail [1], given in Table 2-1 or Table 2-2.
H is the conveyor height [m].

In certain conveyors of both groups, the conveyed load may slide against two stationary retaining sidewalls that avoid material spills. A parameter that characterizes the opposition to motion due to this sliding, is the equivalent mass sliding on sidewalls per unit length of the loaded branch, given by Equation 2-7.

$$q_w = w^2 \cdot \lambda \cdot \rho$$ **Equation 2-7**

Where:

q_w is the equivalent mass sliding on sidewalls, per unit length of the loaded branch [kg/m].
w is the depth of load material against the sidewalls [m].
λ is the horizontal pressure factor [1].
ρ is the apparent density of the load [kg/m³].

The value of q_w may be null due to two reasons: There is no load material pressing against existent stationary retaining sidewalls ($w = 0$), or there are no stationary retaining sidewalls in the conveyor.

The horizontal pressure factor is given by Equation 2-8.

$$\lambda = 1 - \sin \phi_r$$ **Equation 2-8**

Where:

λ is the horizontal pressure factor [1].
ϕ_r is the angle of repose of the load material [°], given in Table 2-4.

2.5.2 Loaded branches of first group conveyors

The resistance to motion of a loaded conveyor branch moving upwards is given by Equation 2-9.

$$R_{LI} = \left((q_L + q_C) \cdot f_C + q_w \cdot f_L\right) \cdot L \cdot g + (q_L + q_C) \cdot H \cdot g$$ **Equation 2-9**

Where:

R_{LI} is the force of resistance to motion of the loaded branch [N].
q_L is the load mass per unit length of the loaded branch [kg/m], given by Equation 2-3 or Equation 2-5.
q_C is the chain mass per unit length [kg/m], including attachments and load support elements.
q_w is the equivalent mass sliding on sidewalls, per unit length of the loaded branch [kg/m], given by Equation 2-7.
L is the horizontal length of conveyor [m].
g is the acceleration of gravity, 9.81 m/s².
f_C is the friction factor of the chain running on its support rail [1], given in Table 2-2 or Table 2-3.
H is the conveyor height [m].

2.5.3 Loaded branches of second group conveyors

The resistance to motion of a loaded conveyor branch moving upwards is given by Equation 2-10.

$$R_{LII} = \left((q_L + q_w) \cdot f_L + q_C \cdot f_C\right) \cdot L \cdot g + (q_L + q_C) \cdot H \cdot g$$ **Equation 2-10**

Where:

R_{LII} is the force of resistance to motion of the loaded branch [N].
q_L is the load mass per unit length of the loaded branch [kg/m], given by Equation 2-4.
q_w is the equivalent mass sliding on sidewalls, per unit length of the loaded branch [kg/m], given by Equation 2-7.
q_C is the chain mass per unit length [kg/m], including attachments and load pushing or scraping elements.
L is the horizontal length of conveyor [m].
f_L is the friction factor of the load sliding on trough material [1], given in Table 2-4.
g is the acceleration of gravity, 9.81 m/s².
f_C is the friction factor of the chain running on its support rail [1], given in Table 2-2 or Table 2-3.
H is the conveyor height [m].

2.6. Chain Pull, Driving Force, and Driving Power

Two essential variables in a conveyor are:

1. The maximum tensile force acting on the chain, T_{max} [N], also known as chain pull.
2. The force exerted by the chain on the drive sprocket, F_D [N], also known as driving force.

Although their values may be close, and even equal sometimes, these forces represent two different concepts that should not be confused about each other. The value of the chain pull is necessary to select a chain with the appropriate strength to do its work as traction element of the conveyor. The driving force value is needed to determine the mechanical power that the conveyor will demand from its drive, through the drive sprocket and its shaft. In any case, Equation 2-11 holds.

$$T_{max} \geq F_D$$ **Equation 2-11**

Detailed calculation procedures of chain pull, driving force, take-up force, and driving power for two typical chain conveyors follow on

2.6.1 Apron Conveyor

A sketch of an upward inclined apron conveyor working under load is given in Figure 2-15. The drive sprocket is always located at the head of the conveyor, where the loaded branch ends. Load is charged at the tail and discharged at the head of the conveyor. This layout assures the best distribution of tensions along the traction chain. Tail sprocket should include a constant force take-up device, to maintain an optimum chain engagement with its sprockets.

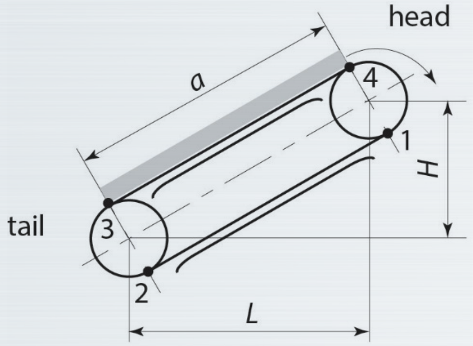

Figure 2-15. Inclined apron conveyor working under load

There are four characteristic points along the chain, shown and numbered in Figure 2-15:

1. Here the chain exits the drive sprocket. This point, or point 2, is where the lowest tension of the chain takes place.

2. Here the chain enters the tail sprocket.

3. Here the chain exits the tail sprocket.

4. Here the chain enters the drive sprocket. This is usually the point of highest tension of the chain.

Since the apron conveyor is a conveyor of the first group, it is possible to pose the following equations:

$$\text{if } R_U \geq 0 \text{ then } T_1 > 0 \text{ else if } R_U < 0 \text{ then } T_1 > |R_U|$$ **Equation 2-12**

$$T_2 = T_1 + R_U$$ **Equation 2-13**

$$T_3 = T_2 \cdot K_S$$ **Equation 2-14**

$$T_4 = T_3 + R_{LI}$$ **Equation 2-15**

$$F_D = T_4 \cdot K_S - T_1$$ **Equation 2-16**

Where:

T_1 to T_4 are the chain tensions in points 1 to 4 [N].
R_U is the resistance to movement of the unloaded branch [N], calculated from Equation 2-6.
R_{LI} is the resistance to movement of the loaded branch [N], calculated from Equation 2-9.
K_S is the factor of resistance to movement of chain on sprockets [1], generally between 1.05 and 1.08.
F_D is the driving force [N].

Once the value of T_1 is chosen according to Equation 2-12, the other tensions and the driving force are obtained from the remaining four equations. The biggest chain tension takes place in point 4. Therefore,

$$T_{max} = T_4$$ **Equation 2-17**

The take-up force at the tail sprocket is given by Equation 2-18.

$$F_T = T_2 + T_3$$ **Equation 2-18**

Where:

F_T is the take-up force at tail sprocket [N]
T_2 and T_3 are the chain tensions at points 2 and 3 [N].

2.6.2 Drag conveyor

A horizontal drag conveyor working under load is sketched in Figure 2-16. The drive sprocket is always located at the head of the conveyor, where the loaded branch ends. As previously said, the pulling end of a branch is the one pointed by the direction of the speed of movement. Load is charged at the tail and discharged at the head of the conveyor. This layout assures the best distribution of tensions along the traction chain. Tail sprocket should include a constant force take-up device, to maintain an optimum chain engagement with its sprockets.

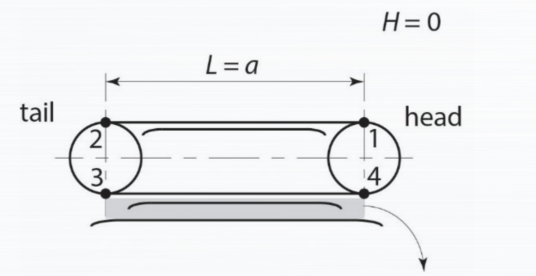

Figure 2-16. Horizontal drag conveyor working under load

There are four characteristic points along the chain, shown and numbered in Figure 2-16:

1. Here the chain exits the drive sprocket. This point, or point 2, is where the lowest tension of the chain takes place.
2. Here the chain enters the tail sprocket.
3. Here the chain exits the tail sprocket.
4. Here the chain enters the drive sprocket. This is usually the point of highest tension of the chain.

Since the drag conveyor is a conveyor of the second group, it is possible to pose the following equations:

$$\text{if } R_U \geq 0 \text{ then } T_1 > 0 \text{ else if } R_U < 0 \text{ then } T_1 > |R_U|$$ **Equation 2-19**

$$T_2 = T_1 + R_U$$ **Equation 2-20**

$$T_3 = T_2 \cdot K_S$$ **Equation 2-21**

$$T_4 = T_3 + R_{LII}$$ **Equation 2-22**

$$F_D = T_4 \cdot K_S - T_1$$ **Equation 2-23**

Where:

T_1 to T_4 are the chain tensions in points 1 to 4 [N].
R_U is the resistance to movement of the unloaded branch [N], calculated from Equation 2-6.
R_{LII} is the resistance to movement of the loaded branch [N], calculated from Equation 2-10.
K_S is the factor of resistance to movement of chain on sprockets [1], generally between 1.05 and 1.08.
F_D is the driving force [N].

Once the value of T_1 is chosen according to Equation 2-19, the other tensions and the driving force are obtained from the remaining four equations. The biggest chain tension takes place in point 4. Therefore,

$$T_{max} = T_4$$ **Equation 2-24**

The take-up force at the tail sprocket is given by Equation 2-25.

$$F_T = T_2 + T_3$$ **Equation 2-25**

Where:

F_T is the take-up force at tail sprocket [N]
T_2 and T_3 are the chain tensions at points 2 and 3 [N].

In drag conveyors, chain tension in all points of loaded branch should be high enough for the flights to stay close to vertical. If tension near point 3 is not high enough, it may be necessary to increase the value of tension T_1 chosen in Equation 2-19 to obtain appropriate values for T_3 and the remaining tensions.

2.6.3 Driving Power

Once the chain pull is known, it is easy to determine the mechanical power drawn by the conveyor from its drive. This power is given by Equation 2-26.

$$P_D = K_A \cdot F_D \cdot v$$ **Equation 2-26**

Where:

P_D is the mechanical power drawn by the conveyor from its drive [W].
F_D is the driving force of the conveyor [N], as given in subsections 2.5.1 and 2.5.2.
v is the speed of the chain in the conveyor [m/s].
K_A is application service factor [1], as given in the first part of this technical section.

2.7. Traction Chain Selection

Conveyor chains usually work in a highly contaminated environment, where lubrication oil is mixed with the conveyed material, or the former is in fact substituted by other less proper substances as water or sugar cane juice, for example. In addition, many chain conveyors are used inside furnaces, where the temperature is high. All these factors make the selection of conveyor chains a non-established technical science. This explains why a formal selection procedure, similar to the existent for power transmission chains, has not yet been standardized.

The de-facto standard is based on the ultimate tensile strength (UTS) of the chain, using a so-called safety factor according to very general guidelines. The UTS of a chain, measured in standard tensile test machines, is an objective measure of the static load the chain may endure before breaking. However, well before breaking, the chain elongates plastically, and becomes useless. Therefore, the allowable working force of a chain should be many times less than the UTS, applying a safety factor not less than seven. To cope with the dynamic behavior typical of conveyor chain operation, a service factor is also applied in the selection calculation, based above all on practical experience.

A number of sources suggest checking the chain for pressure (*p*) and pressure-speed product (*pv*) on the bush/roller pair. The basis of this check lies in the fact that in a reasonably clean environment, a well lubricated standard engineering steel chain may work satisfactorily during 15 000 hours under a bearing pressure of 1 800 000 Pa, if the sliding speed in the pair does not exceed 0.15 m/s. However, under real plant conditions of operation, it is difficult to make a sound prognosis of the working life under any given *p* and *pv* values. Consequently, many manufacturers cite, but do not enforce the bearing pressure and speed check, and rely mainly on the UTS safety factor method. This is the state of the art nowadays.

The maximum tensile force on the chain, obtained as explained in subsection 2.6, is linked to the ultimate tensile strength of the selected chain by Equation 2-27.

$$T_{max} \leq K_N \cdot Q_{mean} / (K_A \cdot S_F)$$ **Equation 2-27**

Where:

T_{max} is the maximum tensile force acting on conveyor chains [N], as obtained in subsection 2.5.
Q_{mean} is the mean ultimate tensile strength of the selected chain [N], recorded in chain tables.
K_A is service factor of the application [1], as given in the first part of this technical section.
K_N is the multi-strand factor [1], as given in the first part of this technical section.
S_F is the safety factor for strength of the chain [1], as given in Table 2-5.

The multi-strand factor takes into account that the maximum tensile force may be exerted sometimes by a sole chain, more frequently shared by two, and by three or more in big conveyors.

When the application needs an especial level of reliability, the minimum ultimate tensile strength Q_{min} is applied in Equation 2-27, instead of the mean value. Regarding the safety factor, a value linked to the speed of the chain is a rational approach, as given in Table 2-5. Within the ranges of S_F given, the higher values should be adopted when more reliability is asked for the application, or when the number of teeth of sprockets is close to the minimum.

Table 2-5. Safety factor for strength of conveyor chains

Chain speed v [m/s]	Safety factor S_F [1]
≤ 0.30	7
0.30 — 0.50	7 — 8
0.50 — 0.65	8 — 10
0.65 — 0.80	9 — 13
0.80 — 1.00	10 — 15
> 1.00	12 — 20

2.8. Units of Measurement

In the Technical Section, all variables are expressed using the basic units of the International System of units, also known as SI. This makes the equations simpler, and avoids the confusions usually arisen when customary units or different multiples and submultiples of SI units are mixed in a single application. Nevertheless, in chain engineering practice, customary units are needed, and will be needed for a long time to assure best professional communication at all levels of activity: from plant & workshop to the enterprise.

How to proceed rationally in this important subject? Our suggestion is as follows:

1. Take all the data in customary units, and convert it to the basic SI units.
2. Make all the calculations in the basic SI units, following this Technical Section.
3. Take all the calculation results needed, and convert them to the customary units.

Steps 1 and 3 require tables of conversion factors. For the convenience of our customers, a group of selected unit conversions linked to the contents of this Technical Section is given to three significant digits and alphabetically ordered, in Table 2-6. To make the conversions straightforward, all units in the table appear once in its first column.

Table 2-6. Unit conversions

To convert from	*To*	*Multiply by*
Cubic feet [ft^3]	Cubic metres [m^3]	0.028 3
Cubic feet per minute [ft^3/min]	Cubic metres per second [m^3/s]	0.000 472
Cubic metres [m^3]	Cubic feet [ft^3]	35.3
Cubic metres per second [m^3/s]	Cubic feet per minute [ft^3/min]	2 120

To convert from	*To*	*Multiply by*
Feet [ft]	Metres [m]	0.305
Feet per minute [ft/min]	Metres per second [m/s]	0.005 08
Feet squared [ft^2]	Metres squared [m^2]	0.092 9
Horsepower [hp]	Watt [W]	746
Inches [in]	Metres [m]	0.025 4
Inches squared [in^2]	Metres squared [m^2]	0.000 645
Kilograms [kg]	Pounds [lb]	2.20
Kilograms force [kgf]	Newtons [N]	9.81
Kilograms per cubic meter [kg/m^3]	Pounds per cubic feet [lb/ft^3]	0.062 4
Kilograms per metre [kg/m]	Pounds per foot [lb/ft]	0.672
Kilograms per second [kg/s]	Pounds per minute [lb/min]	132
	Tonnes per hour [t/h]	3.60
Metres [m]	Inches [in]	39.4
	Feet [ft]	3.28
Metres per minute [m/min]	Metres per second [m/s]	0.016 7
Metres per second [m/s]	Feet per minute [ft/min]	197
	Metres per minute [m/min]	60.0
Metres squared [m^2]	Feet squared [ft^2]	10.8
	Inches squared [in^2]	1 550
Newtons [N]	Pounds force [lbf]	0.225
	Kilograms force [kgf]	0.102
Newtons by meter [N·m]	Pounds force by inch [lbf·in]	8.85
Pounds [lb]	Kilograms [kg]	0.454
Pounds force [lbf]	Newtons [N]	4.45
Pounds force by inch [lbf·in]	Newtons by meter [N·m]	0.113
Pounds per cubic feet [lb/ft^3]	Kilograms per cubic meter [kg/m^3]	16.0
Pounds per foot [lb/ft]	Kilograms per metre [kg/m]	1.49
Pounds per minute [lb/min]	Kilograms per second [kg/s]	0.007 56

To convert from	*To*	*Multiply by*
Tonnes per hour [t/h]	Kilograms per second [kg/s]	0.278
Watt [W]	Horsepower [hp]	0.001 34

AEC hopes that the closing Technical Section of the AEC Catalog on Engineering Chains will be useful for you, our distinguished user. In case of extremely hot or cold temperatures, or the presence of corrosive substances, or a special class of conveyor, or any other design, operational or maintenance concerns, AEC Technical Department will be glad to assist you in the best chain selection for your conveyor.

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