Autodesk Inventor Engineer s Handbook

هندبوک مهندسی نرم افزار Autodesk Inventor

انجمن اینونتور ایران

www.irinventor.com

Email: irinventor@chmail.ir irinventor@hotmail.com

Tel: 09352191813 & 021-46088862

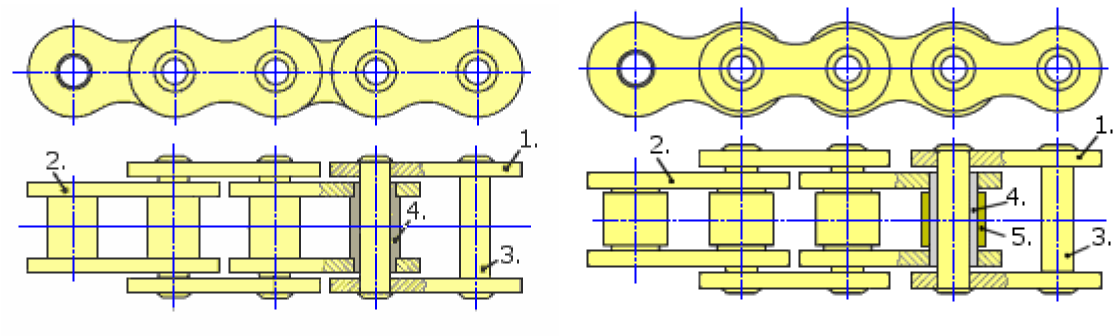
Transmission Mechanism

Roller Chains Calculator

Calculation basics

Chain Construction

The Roller Chains generator is intended to design chain drives with roller and bush chains. The chains can have single strand or multiple strands. The double-pitch chains are also supported. Typical construction of roller and bush chains is shown in the following picture. The main difference is that bush chain does not have a roller.



Bush chain components Roller chain components

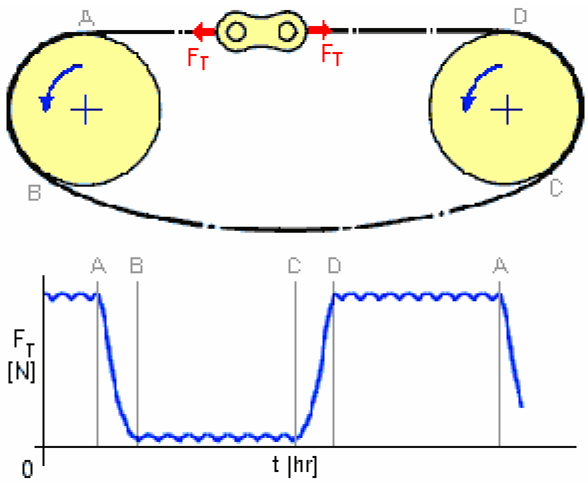
1. outer plate
2. inner plate
3. bearing pins
4. bush
5. roller

If the chain length is an even number of pitches the connection link may be used to connect two ends of the chain together. The chain power capacity is not reduced usually.

If the chain length has an odd number of pitches, an offset link may be used at one end of the chain. Then the connecting link may be used to connect two ends of the chain together. The offset link usually reduces chain power capacity. The amount of power reduction is given by type and construction of the offset link. Consider the reduction of the chain power capacity you need to decrease chain construction factor.

Repeated load tension

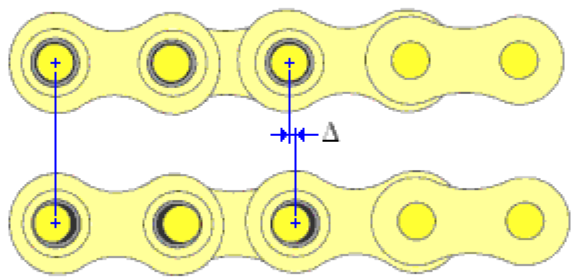
Chains in chain drives are loaded by cyclical tension so the chain is subject of fatigue. The typical load diagram is shown in the following picture. The load diagram may differ for different drive layouts.



Wear

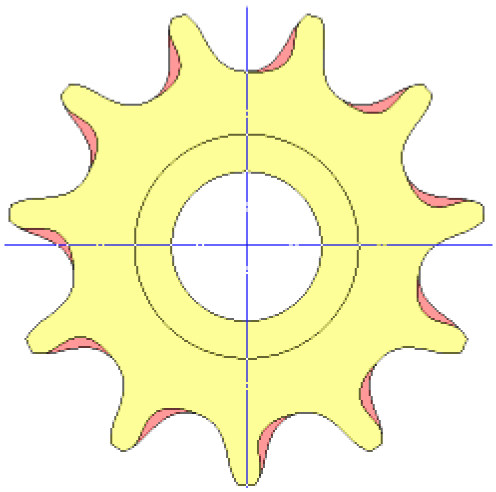
Wear is important in designing roller chain drives. The roller chains are normally most affected by chain joint wear and sprocket wear.

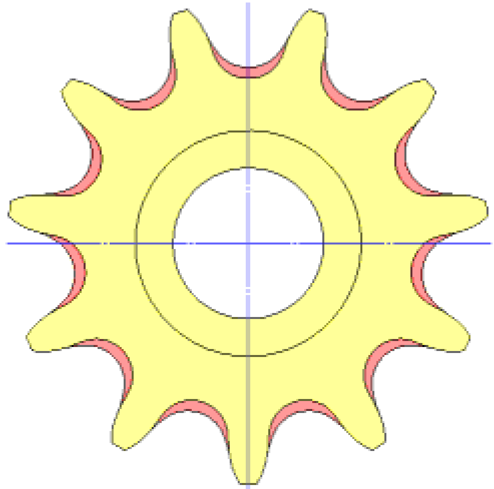
Chain joint wear causes roller chains get longer. Sprockets for roller chains are designed to accept up to 3% (1.5% for double pitch chains) chain elongation from wear. When the chain elongates beyond that point it no longer fits the sprockets and the system will not operate properly. There might be different criteria for chain joint wear for large sprockets or drives with fixed center distance. If the worn chain is about to be replaced, we recommend that you replace the sprockets as well.



Chain joint wear

Sprocket wear is considered as modification of the teeth shape. The teeth begin to take on a hooked shape. For idler sprockets usually wear at the bottom of the tooth space. When the tooth space is worn deeply enough, the chain rollers may bind against the tooth tops as they enter and leave the idler sprocket. The sprocket wear can be source of shock loads in the chain. Reversing the sprocket on the shaft can sometimes extend the life of worn sprocket.



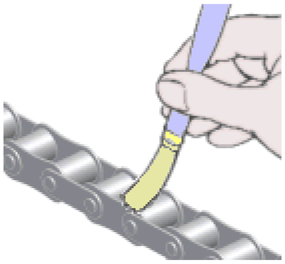


Worn roller chain drive sprocket

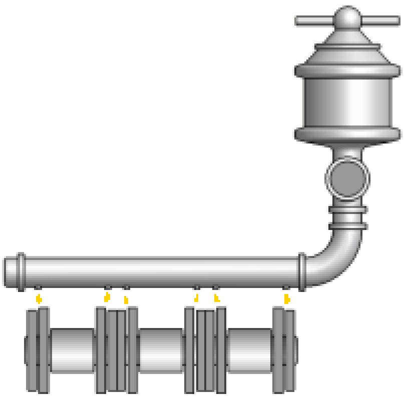
Worn roller chain idler sprocket

Lubrication

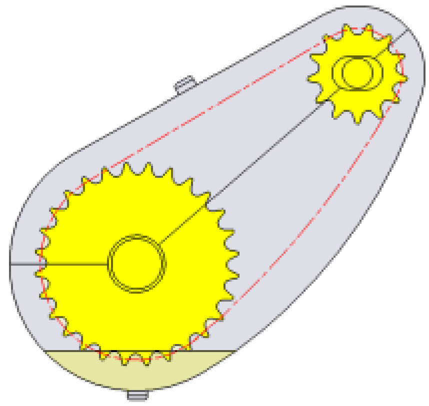
Good lubrication must be provided for the chain drive to obtain longest life from a chain. The effective lubrication is a matter of applying the correct lubricant where it is most needed. The main problem is getting enough clean lubricant to the bearing surfaces of pins, bushings, and rollers.



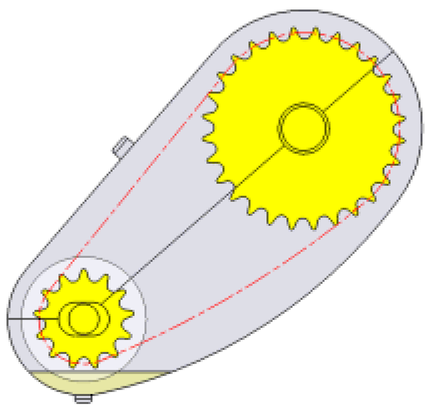
Manual lubrication: Oil is applied copiously with a brush or spout can at least once every 8 hours of operation. The drive is stopped and power to the drive is locked out. Volume and frequency should be sufficient to prevent overheating of the chain or red-brown (rust) discoloration in the chain joints.



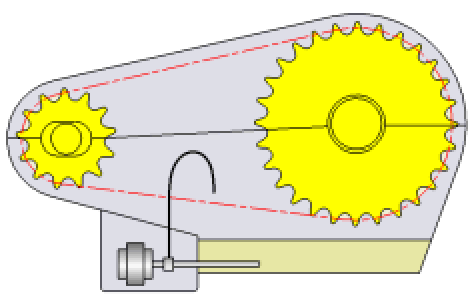
Drip lubrication: Oil continuously dripped onto the upper edges of the link plates, or sidebars, from a drip lubricator. Volume and frequency should be sufficient to prevent red-brown (rust) discoloration of lubricant in the chain joints. Usually the drip rate is from 4 to 20 or more drops per minute. Precaution must be taken against misdirection of the drops by windage.



Bath lubrication: The lower strand of chain runs through a sump of oil in the drive housing. The oil level should reach the pitch line of the chain at its lowest point while operating.



Disc lubrication: The chain operates above the oil level. The disc picks up oil from the sump and deposits it onto the chain, usually with a through. The diameter of the disc should be such as to produce adequate rim speed to pick up oil effectively. Higher speeds can cause oil foaming and overheating.



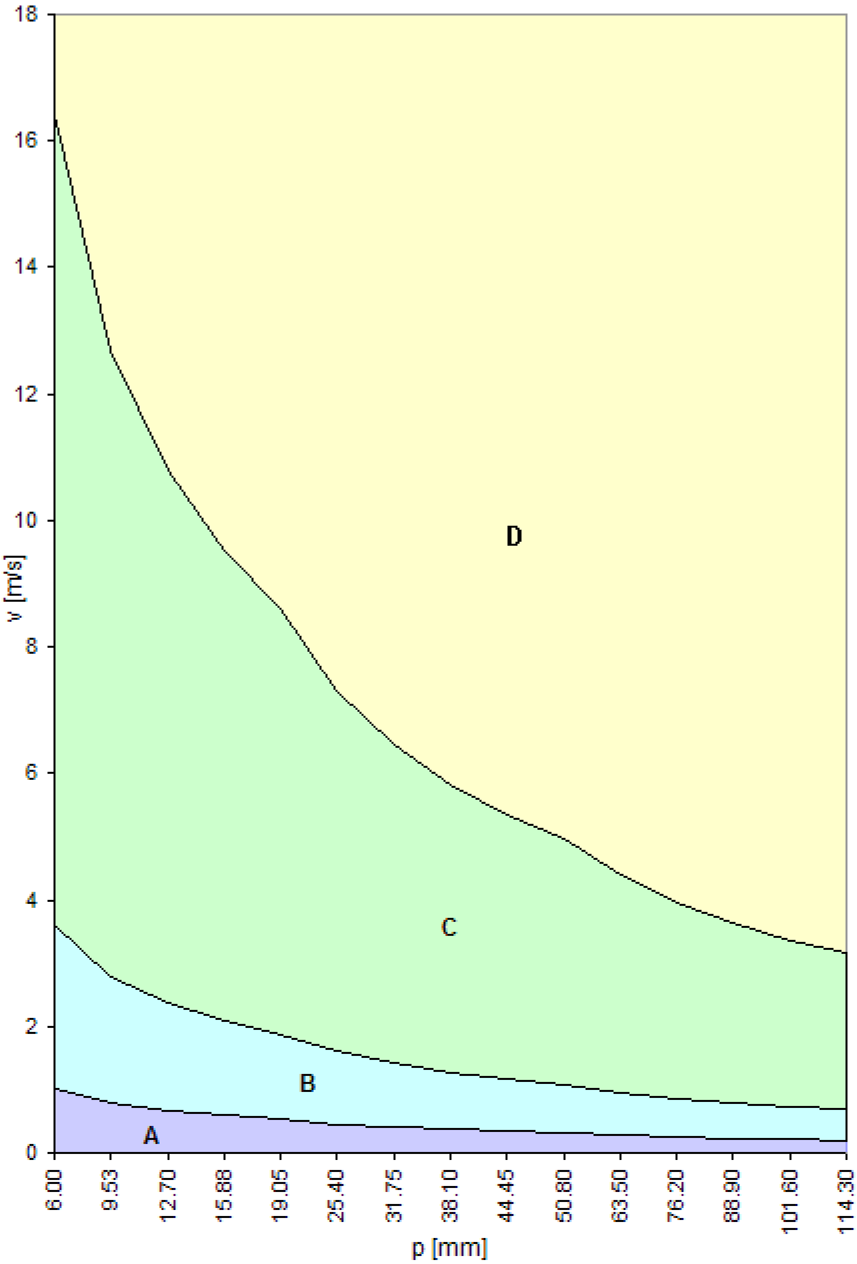
Forced feed lubrication: The lubricant is supplied by a circulating pump capable of supplying chain drive with a continuous stream of oil. The oil should be directed at the slack strand and applied inside the chain loop and evenly across the chain width to ensure that the oil reaches all bearing surfaces. An oil cooler and oil filter can be used if necessary.

Lubricating oil should be free from contaminants, particularly abrasive particles.

Chain drive lubrication oil viscosity class is defined based on ambient temperature as:

Ambient temperature [°C]	$-5 \le t \le +5$	$+5 \le t \le +25$	$+25 \le t \le +45$	$+45 \le t \le +70$
Oil viscosity class	VG 68 (SAE 20)	VG 100 (SAE 30)	VG 150 (SAE 40)	VG 220 (SAE 50)

Adequate recommended lubrication is specified according to the chain size and speed and shown on following chart:



where:

A) Range of manual lubrication. Maximum chain speed is

$v = 2.8\ p^{-0.56}$

B) Range of drip feed lubrication. Maximum chain speed is

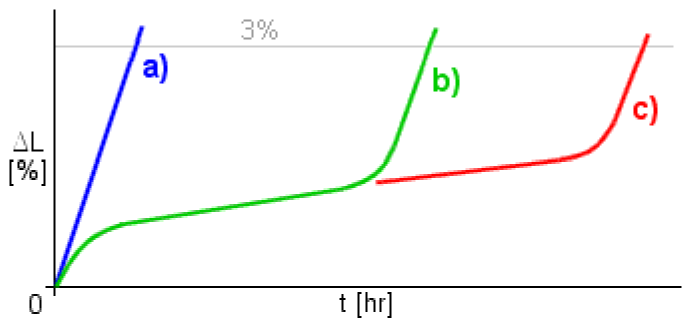
$v = 7\ p^{-0.56}$

C) Range of oil bath or disc lubrication. Maximum chain speed is

$v = 35\ p^{-0.56}$

D) Range of forced feed lubrications

The progress of chain joint wear during chain service life with respect to quality of lubrication is shown in the following picture:



where:

a No lubricated chain in soiled abrasive environment

b Insufficient chain lubrication

c Adequately lubricated chain

Normally at the beginning of the chain service the wear progress rapidly and this stage it is known as initial wear. Initial wear can be minimized by pre-loading the chain what some manufacturers do. The pre-load can increase chain service life.

The chain joint wear then continue and the progress is slow what is known as normal wear. If the chain is adequately lubricated the chain joint wear continue to exhibit normal wear and eventually the chain run out its useful life. At the end of the useful life of a chain, the chain joint wear starts to rapidly progress again.

Ultimate tensile strength of the chain F_U

The ultimate tensile strength of the chain is the highest load that the chain can withstand in a single application before breaking. It is not allowable working load either measuring load. The main value of a specification for minimum ultimate strength is to ensure that the chain was assembled properly. Roller Chain generator uses minimum ultimate tensile strength for determination of safety factors from chain breaking. Also, using additional factors it participates in calculation of expected chain links service life or link plates fatigue.

The default value of ultimate tensile strength comes from standard recommendations for given chain size but you may wish to consult this parameter with specifications provided by your chain manufacturer. The tensile strength may differ for the same size of the chain among different chain producers as well as different materials of the chain.

Specific chain mass m

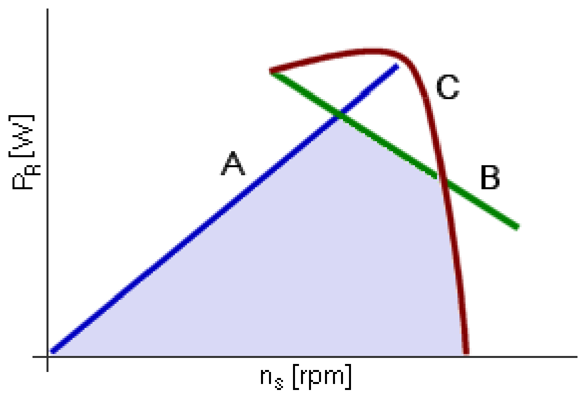
Specific mass of the chain depends on the chain size, construction, and material. The default value is taken from standard recommendations or it is closest value of steel chains produced by chain manufacturers. The specific mass is used for computing centrifugal force as well as for vibration analysis.

Chain construction factor Φ

The chain construction factor describes the actual quality of the chain. It has direct impact to chain power rating as well as chain permissible bearing area pressure. The factor is usually equal to one. It is bigger than one if the chain is made from material with better strength or the quality of the chain is better than mentioned in national standards.

Chain power rating P_R

Chain power rating represents chain capacity rating for specific operating conditions. Normally, the chain capacity is limited by link plate fatigue, roller and bushing impact fatigue and galling between the pin and bushing. See the typical power rating chart on the following image.



where:

A chain drive power capacity limited by link plate fatigue

B chain drive power capacity limited by roller and bush impact fatigue

C chain drive power capacity limited by pin-bush galling

Power correction factors

Chain power rating equations provide valid power capacity for chain drives what work under specific normal operation conditions. If your chain drive works under working condition what does not equal to the normal operation conditions, there is a need to introduce power correction factors as described below.

Normal operation conditions:

1. a chain drive with two sprockets on parallel horizontal shafts;

2. a small sprocket with 19 teeth;
3. a simplex chain without cranked link;
4. a chain length:

   120 pitches for ISO chains,

   100 pitches for ANSI, CSN chains;

5. a speed ratio of 1:3 or 3:1
6. an expected service life:

   15000 hours ISO, ANSI, DIN chains,

   10000 hours CSN chains;

7. an operating temperature between -5ºC and +70ºC;
8. sprockets correctly aligned and chain maintained in correct adjustment;
9. uniform operation without overload, shocks, or frequent starts;
10. clean and adequate lubrication throughout the life of the chain;

Shock factor Y

Service factor takes into account dynamic overloads dependent on the chain drive operating conditions, driver, and driven characteristics. The shock factor is used to determine size of the service factor as well as dynamic factor of safety. The peak loads caused by unexpected shocks and peak overloads can dramatically increase with large moments of inertia of driver or driven machine. By default the Chain generator uses following table to determine shock factor.

Y	*Application*
1.0	Smooth running
1.5	Smooth running with occasional shocks
2.0	Slight shocks, moderate temporary peak overloads
3.0	Moderate shocks, heavy temporary peak overloads
4.0	Heavy shocks, moderate constant peak overloads
5.0	Heavy shocks, heavy constant peak overloads

Service factor f_1

Service factor takes into account dynamic overloads dependent on the chain drive operating conditions and resulting, in particular, from the nature of the driver and driven elements. The valued of factor can be selected directly or using following table.

Driven machine characteristics	*Driver machine characteristics*		
	Smooth running	Slight shocks	Moderate shocks
Smooth running	1.0	1.1	1.3
Moderate shocks	1.4	1.5	1.7
Heavy shocks	1.8	1.9	2.1

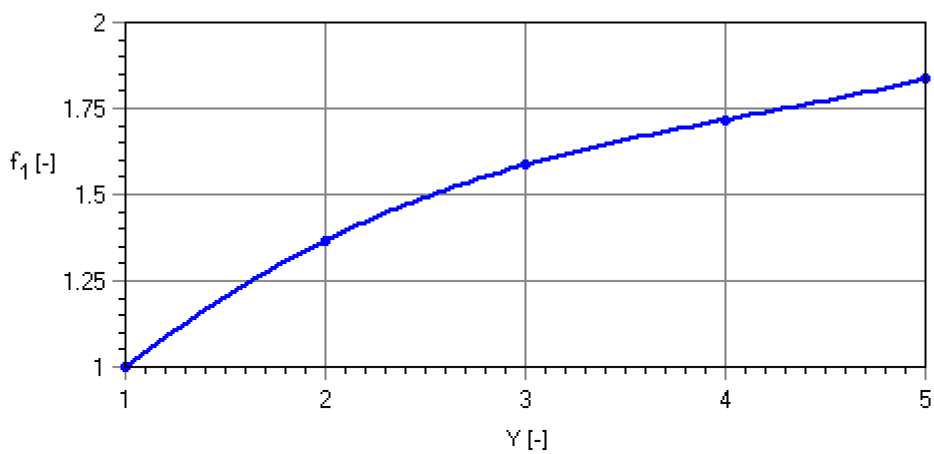
Definitions of characteristics of driver machines

Driver machine characteristics	*Machine type examples*
Smooth running	Electric motors, steam, and gas turbines and internal combustion engines with hydraulic coupling
Slight shocks	Internal combustion engines with six cylinders or more with mechanical coupling, electric motors subjected to frequent starts (more than two per day)
Moderate shocks	Internal combustion engines with less than six cylinders with mechanical coupling

Definitions of characteristics of driven machines

Driven machine characteristics	*Machine type examples*
Smooth running	Centrifugal pumps and compressors, printing machines, uniformly load belt conveyors, paper calendars, escalators, liquid agitators, and mixers, rotary dryers, fans
Moderate shocks	Reciprocating pumps and compressors, with three or more cylinders, concrete mixing machines, non-uniformly loaded conveyors, solid agitators and mixers
Heavy shocks	Excavators; roll and ball mills; rubber processing machines; planers, presses, shears, pumps, compressors with one or two cylinders, oil drilling rings

By default the service factor is determined in accordance with shock factor as shown on the following chart:



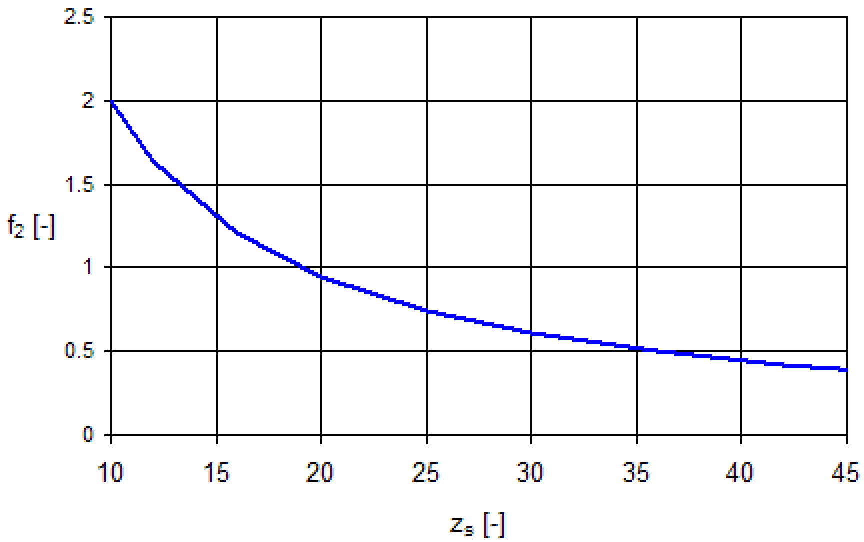
where:

Yshock factor [-]

Sprocket size factor f $_2$

Sprocket size factor takes in an account number of teeth on the small sprocket. The factor is equal to one if power rating equations consider number of teeth of the smallest sprocket in the drive. Size of the smallest sprocket may have specific impact on each portion of chain power rating.

If you customize the chain power rating you need also revise sprocket size factor. If the power rating is specified from power rating tables with respect to number of teeth of the smallest sprocket the factor should remain equal to one. If the power rating provided is obtained from power rating chart where number of teeth of the smallest sprocket is not considered then you may need to adjust size of the factor. The sprocket size factor affects the design power. By default the Chain generator determines size of the factor as



$$f_2 = \left(\frac{19}{z_S}\right)^{1.08}$$

where:

z_sNumber of teeth on the small sprocket [-]

Multiple strand factor f $_3$

Power ratings for single strand chains are given by power rating equations by default. The power ratings for multiple strand chains equal single-strand ratings multiplied by the multiple strand factor. By default, the program uses in-build table as shown in the following table. The multiple strand factor is also used in expected service life analysis.

Chain strands12 3 4 5 6

f_3 [-] 11.72.53.33.94.6

Lubrication factor f_4

The lubrication factor tells the program how much inadequate lubrication impacts chain power capacity as well as service life. If the adequate lubrication is selected the size of the factor is equal to one what does not affect the analysis. If inadequate lubrication has to be used then the factor decreases chain power rating limited by pin-bush galling or the factor increases entire design power. By default the program uses following in-build table to indicate impact of selected lubrication.

Chain speed	*Lubrication factor [-]*			
	Lubrication			
[m/s]	Recommended	Insufficient		No lubrication
		Clean environment	Soiled environment	
up to 4	1	0.6	0.3	0.15
up to 7		0.3	0.15	Inadmissible
up to 12		Inadmissible		
more than 12				

Center distance factor f_5

The minimum center distance is one half the sum of outside sprocket diameters to avoid tooth interference. To ensure adequate wrap on the small sprocket (approximately 120 Degrees) it is suggested to have minimum center distance of the sum of the outside diameter of the large sprocket plus one half the outside diameter of the small sprocket.

It is good practice to set the center distance at 30 to 50 times the chain pitch. The longest practical center distance is about 80 times the chain pitch because chain sag and catenary tension become very large.

The center distance factor corrects the design power and takes into account difference of the actual center distance from the normal. The reason of the center distance factor comes from modification of load tension distribution and its affect to chain fatigue. By default the center distance factor is determined as follows.

$$f_5 = \left(\frac{X_B}{X}\right)^{0.4}$$

where:

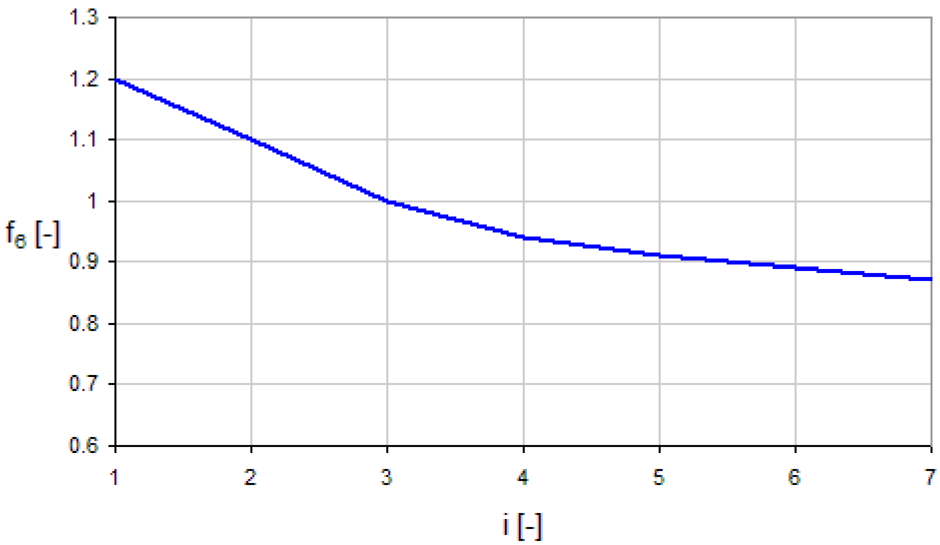
f_5 center distance factor [-]

X_B number of chain links for normal operating conditions [-]

X actual number of chain links in the drive [-]

Ratio factor f_6

The ratio factor corrects the design power and takes into account difference of the transmission ratio from the normal. The reason of the ratio factor comes from modification of load tension distribution and its affect to chain fatigue. By default the ratio factor is determined from the following chart with respect to actual transmission ratio.



Transmission ratio is given from number of teeth of the driver and driven sprocket.

for $z_1 < z_2$ the $i = z_2 / z_1$

for $z_1 > z_2$ the $i = z_1 / z_2$

where:

f_6 ratio factor [-]

i transmission ratio [-]

z_1 number of teeth of the driver sprocket [-]

z_2 number of teeth of the driven sprocket [-]

Service life factor f_7

The service life factor corrects the design power and takes into account difference of the required service life from the normal. By default the ratio factor is determined as

$$f_7 = \left(\frac{L_h}{L_{hB}}\right)^{0.4}$$

where:

f_7 Service life factor [-]

L_h Required service life [hours]

L_{hB} Normal expected service life [hours]

Pressure in chain bearing area analysis

During the chain drive service the fluctuating tensile load acts on contact surfaces between pins and bushings what causes specific size of chain bearing area pressure. If this pressure exceeds permissible pressure in chain bearing area the chain service life might significantly decrease. The amount of the actual pressure in chain bearing area is computed from maximum tension in chain span as follows

$$p_B = \frac{F_{T\max}}{A}$$

where:

p_B Actual pressure in chain bearing area [Pa]

F_{Tmax} Maximum tension in taught chain span [N]

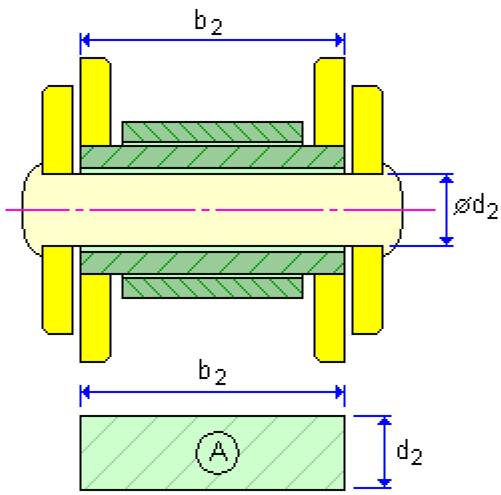
A Chain bearing area [m^2]

Chain bearing area A

Chain bearing area is defined by width of internal chain link and pin diameter. The actual values are defined for each chain within XML data files.

$$A = b_2 d_2$$

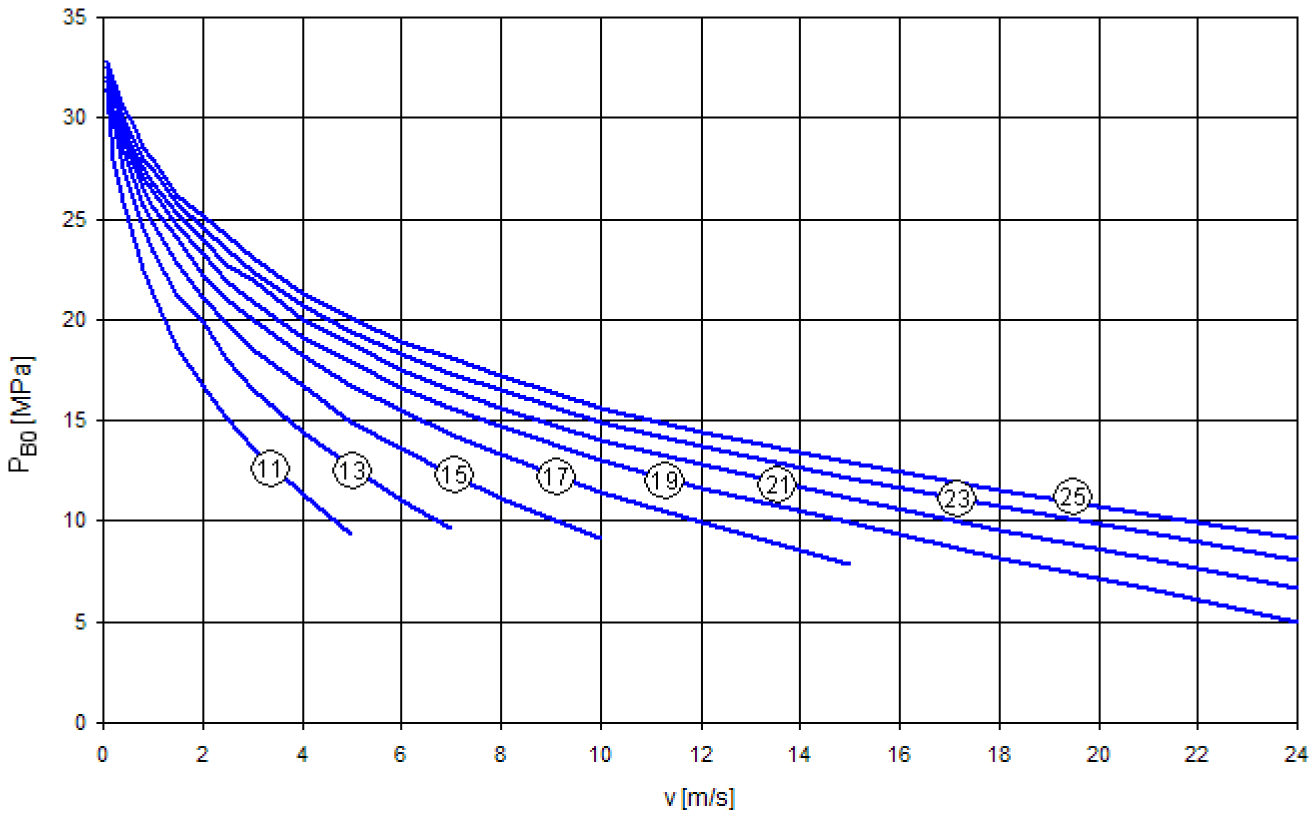
where:
A Chain bearing area [m^2]
b_2 Width of internal chain link [m]
d_2 chain pin diameter [m]



Permissible pressure in chain bearing area p_0

The values of permissible pressure in chain bearing area shown on the chart below apply to normal operating conditions only. For specific operating conditions the value is corrected by specific friction factor λ what results in total permissible pressure. The total permissible pressure is then compared with the actual pressure in chain bearing area.

Permissible pressure specified here can be used for common steel chains. For chains made from different materials, you may need to adjust the permissible pressure accordingly.



The permissible pressure obtained from the previous chart is also corrected by chain construction factor so the amount of permissible pressure is defined as follows

$p_0 = p_{B0}\,\varphi$

where:

p_0 Permissible pressure in chain bearing area for normal operating conditions [Pa]

p_{B0} Specific permissible pressure in chain bearing for normal operating conditions [Pa]

ϕ Chain construction factor [-]

Specific friction factor λ

Specific friction factor corrects the permissible pressure in chain bearing area. The size of the factor depends on how much actual operating conditions differ from normal and it is defined as

$$\lambda = \frac{f_4}{f_1 \cdot f_5 \cdot f_6 \cdot f_7}$$

where:

λ Specific friction factor [-]

f_1 Service factor [-]

f_4 Lubrication factor [-]

f_5 Center distance factor [-]

f_6 Ratio factor [-]

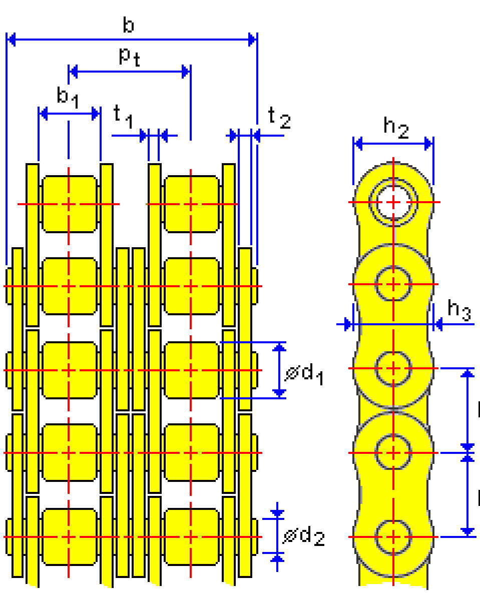
f_7 Service life factor [-]

Geometry design properties

The main properties of bush or roller chain as well as sprockets are described here. The short pitch chains as well as double pitch chains are considered. The chains can have single or multiple strands. All properties are defined within a library of chains.

Roller chain properties

Main chain properties are based on national standard recommendations. Specific size of the chain also defines corresponding dimensions for tooth sprockets as they have to engage properly with the chain.



where:
p pitch
p_t transverse pitch
b maximum width over bearing pins
b_1 minimum width between inner plates
d_1 maximum roller diameter
d_2 maximum bearing pin body diameter
t_1 thickness of inner plates
t_2 thickness of outer plates
h_2 maximum inner plate depth
h_3 maximum outer or intermediate plate depth

Toothed sprocket properties

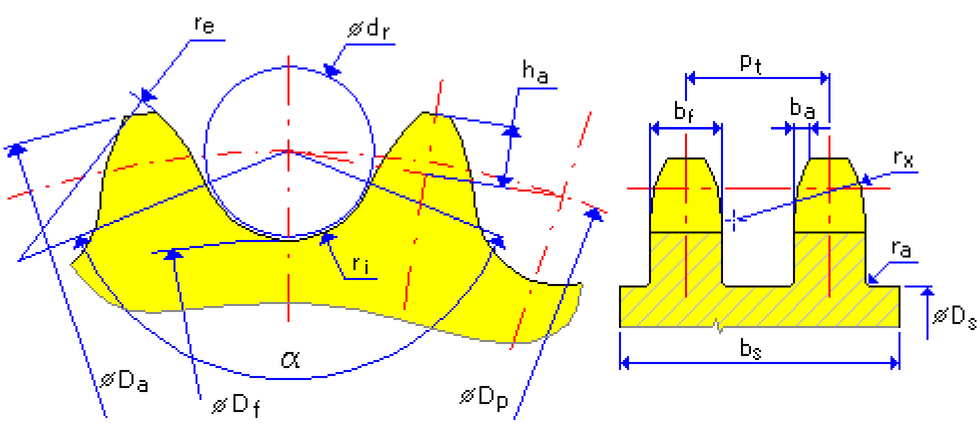
Sprocket dimensional properties are based on a specific chain size as well as national standard recommendations. Not all properties are described here because of complexity. For more details on exact sprocket dimensions see the corresponding chain standards.

Two types of tooth form are considered:

- Theoretical tooth form
- Simplified ISO tooth form

The theoretical tooth form is designed so that the chain rollers ride out towards the tips of the sprocket teeth as the chain wears and elongates. There are many ways how to produce sprocket teeth, and the actual tooth form may not exactly match the theoretical form.

Simplified ISO tooth form is determined by the minimum and maximum tooth gap forms. The actual tooth form, which is provided by cutting or an equivalent method, shall have tooth flanks of a form lying between maximum and minimum flank radii and blending smoothly with the roller seating curve subtending the respective angles. By default the Roller Chain Generator uses minimum tooth gap form recommendations.



$$D_p = \frac{p}{\sin\frac{\pi}{z}}$$

$$D_a = D_p \cdot \cos\frac{\pi}{z} + 2 \cdot h_a$$

$D_f = D_p - 2\, r_i$
$b_s = p_t (k - 1) + b_f$
$b_a = b_{ax}\, p$

$$r_i = \frac{d_r}{2} + SC$$

$$D_g = D_p \cdot \cos\frac{\pi}{z} - h_{max} - 2 \cdot r_a$$

where:

D_P pitch diameter

D_a tip diameter

D_f root diameter

d_r maximum bush or roller diameter

z number of sprocket teeth

p chordal pitch equals to chain pitch

p_t strand transverse pitch

k number of strands

SC seating clearance

r_i roller seating radius

r_e tooth flank radius

α roller-seating angle

h_a height of tooth above pitch polygon

b_f tooth width

b_a tooth side relief

$b_{ax\,f}$ tooth side relief factor

r_x tooth side radius

r_a shroud fillet radius

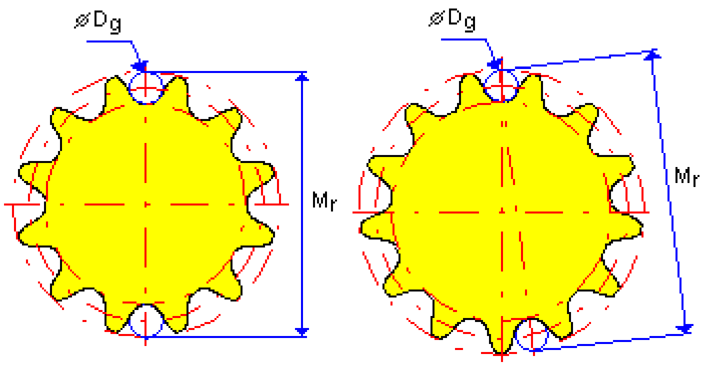
b_s minimum shroud width

D_s maximum shroud diameter

h_{max} maximum plate depth $h_{max} = \max(h_2 ; h_3)$

Measuring toothed sprocket

Even number of teeth Odd number of teeth



$M_r = D_p + 2\,D_g - d_r$

$$M_r = D_p \cdot \cos\frac{\pi}{z} + 2 \cdot D_g - d_r$$

For measuring over pins $D_g = d_r$. For direct measuring $D_g = 0$.

where:

D_p pitch diameter

D_g measuring pin diameter

M_r measurement over pins or direct measurement

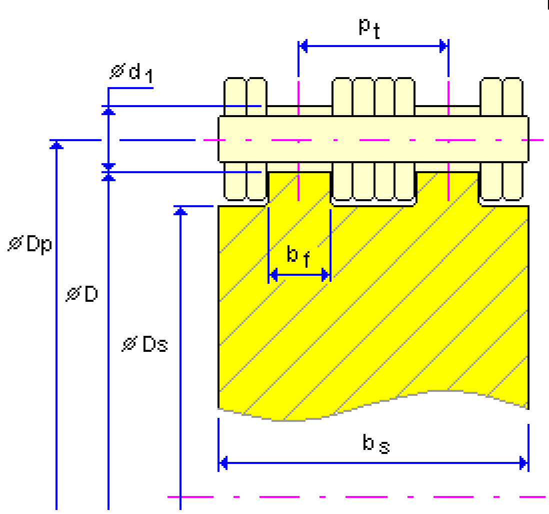
z number of sprocket teeth

d_r maximum chain roller diameter

Flat idler

$D_p = D + d_1$

$b_s = p_t\,(k - 1) + b_f$



$$D_g = D_p \cdot \cos\frac{\pi}{z} - h_{max} - 2 \cdot r_a$$

where:

D_P pitch diameter

D nominal diameter

p chain pitch

p_t strand transverse pitch

k number of strands

b_f strand width

d_1 maximum chain bush or roller diameter

D_s maximum shroud diameter

b_s minimum shroud width

h_{max} maximum plate depth $h_{max} = \max(h2; h3)$

Chain length calculation

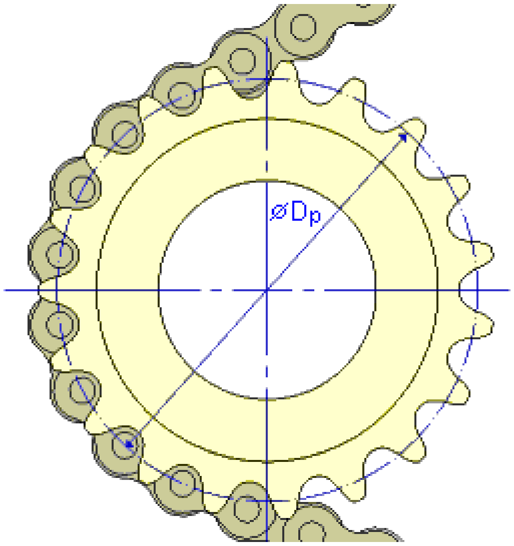
The chain length is given by number of chain links and the chain pitch. The chain drive trajectory is based on individual sprocket position and desired direction of motion.

The algorithm to compute chain length uses sprockets pitch diameters. The pitch diameter for each roller chain drive sprocket or idler is obtained from equations below.

The sliding sprocket position is adjusted accordingly to accomplish desired chain length. The calculation uses linear algebra and iteration solution to find appropriate sliding sprocket position.

When the chain length is computed it is taken onto account that the trajectory is composed from linear segments at chain pitch length and arcs are replaced with actual polygons.

Pitch diameters

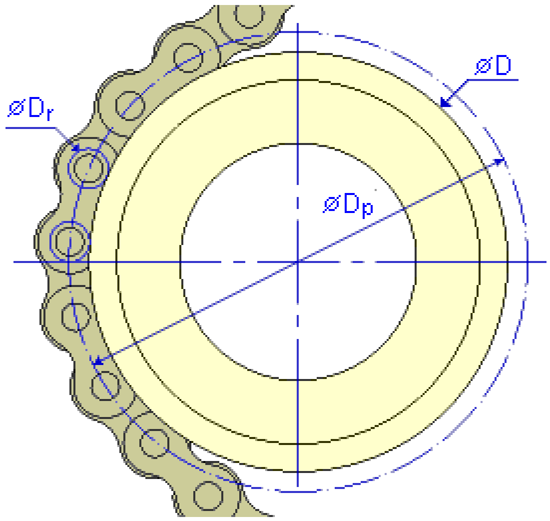


$$D_p = \frac{p}{\sin \frac{\pi}{z}}$$

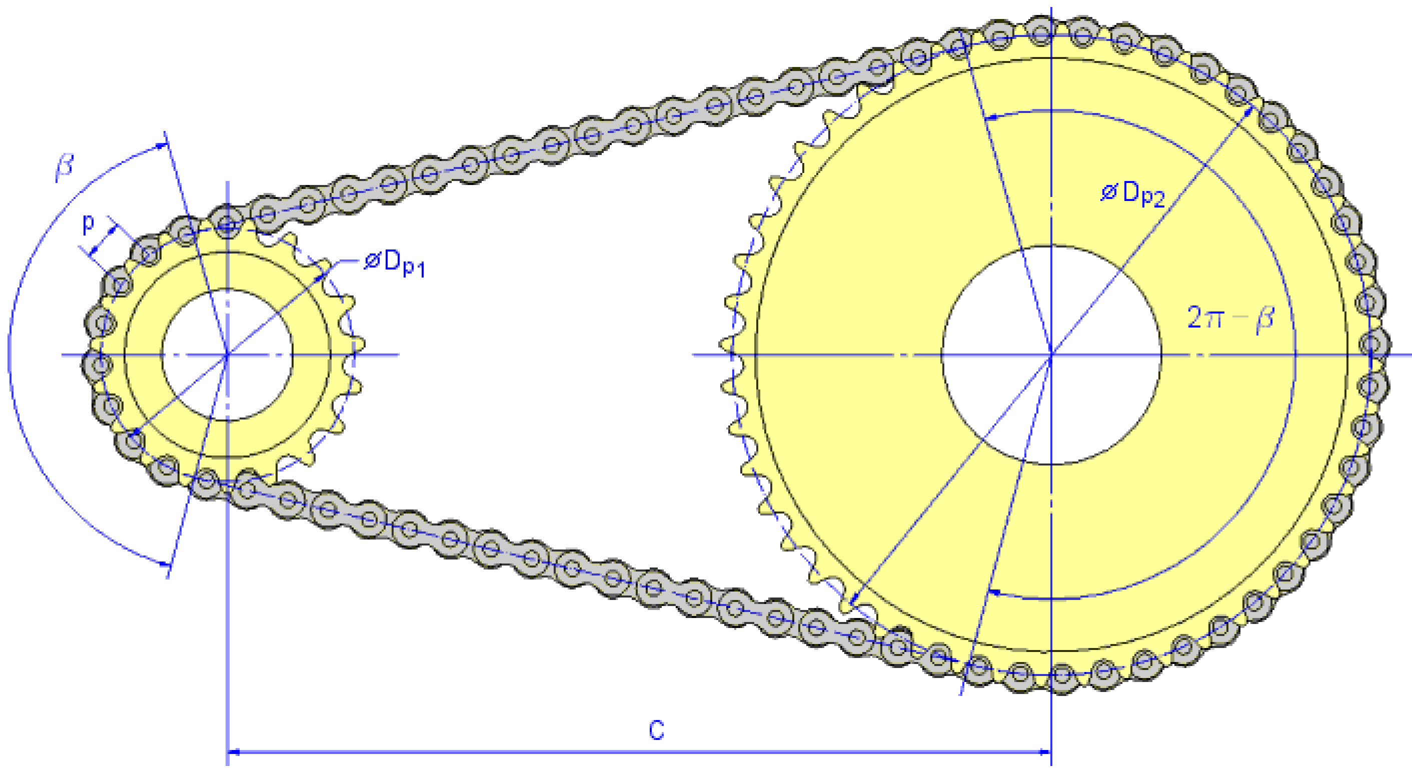
where:
D_p Pitch diameter
p chain pitch

z number of sprocket teeth

$D_p = D + D_r$
where:
D_p Pitch diameter
D Nominal idler diameter
D_r maximum chain roller diameter



Example of chain drive with two sprockets



Required number of chain links for desired center distance

$$X_0 = 2 \cdot \frac{C_0}{p} + \frac{z_1 + z_2}{2} + \frac{p \cdot \left(\frac{|z_2 - z_1|}{2 \cdot \pi} \right)^2}{C_0}$$

The required number of chain links is rounded to closest even or odd number and then the actual center distance is then determined as

$C = F\ p\ [\ 2\ X - (z_1 + z_2)]$

where:

$$F = \frac{1}{4 \cdot \sin\delta \cdot \left(\hat{\delta} + \cot\delta\right)}$$

$$\delta = \arcsin\frac{D_{P2} - D_{P1}}{2C}$$

Contact angle is determined as

$$\beta = 2 \cdot \arccos\frac{D_{P2} - D_{P1}}{2 \cdot C}$$

Number of teeth in contact of the small sprocket

$$z_C = z_1 \cdot \frac{\beta}{360}$$

Meaning of used variables:

C_0 Desired center distance [m]
C Actual center distance [m]
p Chain pitch [m]
z_1 Number of teeth of the driver sprocket [-]
z_2 Number of teeth of the driven sprocket [-]
D_{P1} Pitch diameter of the driver sprocket [m]
D_{P2} Pitch diameter of the driven sprocket [m]
X_0 Required number of chain links [-]
β Contact angle [deg]

Calculation of strength proportions

Power to transmit

$$P = \frac{T \cdot \pi \cdot n}{30}$$

Chain speed

$$v = \frac{D_p \cdot \pi \cdot n}{60}$$

Effective chain pull or tensile load

$$F_p = \frac{P}{v}$$

Centrifugal force

$F_c = m\, v^2$

Maximum tension in taught chain span

$F_{Tmax} = F_p + F_C$

Speed for each driven sprocket or idler

$$n_i = \frac{n_1}{i}$$

Power transmitted by each driven sprocket (for idlers $P_X = 0$ and therefore $P_i = 0$)

$P_i = P\ P_X\ \eta$

Torque what acts on each driven sprocket (for idlers $T_i = 0$)

$$T_i = \frac{30 \cdot P_i}{\pi \cdot n_i}$$

Forces at the tight and slack side of each sprocket are determined. The program defines force on input F_1 and force on output F_2 for each sprocket. These forces are defined with respect to the chain motion. Force on input F_1 is the force in chain span where the chain gets into contact with the given sprocket. Force on output F_2 is the force in chain span where the chain leaves the given sprocket.

Force in chain span at the tight side of the driver sprocket

$F_1 = F_{Tmax}$

Force in chain span at the slack side of the driver sprocket

$F_2 = F_1 - F_p$

Force in chain span at the tight side of each driven sprocket

Force in chain span at the tight side of each driven sprocket is consumed from force where chain gets into contact with logically next sprocket with respect to the chain motion direction.

$F_2 = F_{1(i+1)}$

Force in chain span at slack side of each driven sprocket

$F_1 = F_2 - F_p\ P_X$

Resultant axle load for each sprocket

$$F_R = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos\beta}$$

where:

P	Power [W]
T	Torque [Nm]
n	Speed of the sprocket [rpm]
n_1	Speed of the driver sprocket [rpm]
n_i	Speed of the driven sprocket or idler [rpm]
i	Transmission ratio for driven sprocket or idler [-]
v	Chain speed [m/s]
D_P	Pitch diameter of the sprocket [m]
F_P	Effective chain pull or tensile load [N]
F_C	Centrifugal force [N]
m	Specific chain mass [kg/m]
F_{Tmax}	Maximum tension in taught chain span [N]
P_i	Power transmitted by driven sprocket [W]
P_X	Power ratio factor of driven sprocket [-]
η	Efficiency [-]
T_i	Torque what acts on driven sprocket [Nm]
F_1	Force in the chain span where individual sprocket gets into contact with the chain [N]
F_2	Force in the chain span where individual sprocket leaves the chain [N]
$F_{1(i+1)}$	Force in the chain span where logically next sprocket gets into contact with the chain [N]
F_R	Resultant axle load of each sprocket [N]
β	Contact angle of each sprocket [deg]

Strength check

The Roller Chain Generator uses the following theory to advise users if selected chain works under the specified working conditions.

Static factor of safety from chain breaking is determined for constant load as:

$$S_S = \frac{F_U}{F_{T\max}} \geq S_{S\min}$$

where:

S_{Smin} Minimum allowed static safety factor [-]

F_U Ultimate tensile strength of the chain [N]

F_{Tmax} Maximum tension in taught chain span [N]

Dynamic factor of safety from chain breaking is determined for peak load as:

$$S_D = \frac{F_U}{F_{T\max} \cdot Y} \geq S_{D\min}$$

where:

S_{Dmin} Minimum allowed dynamic safety factor [-]

F_U Ultimate tensile strength of the chain [N]

F_{Tmax} Maximum tension in taught chain span [N]

Y Shock factor [-]

Design power and chain power rating

Chain power rating P_R is consulted with the design power P_D. The chain power rating must be greater than the design power.

$P_D < P_R$

where:

$$P_D = P \cdot f_1 \cdot f_2 \cdot f_5 \cdot f_6 \cdot f_7 \cdot \frac{1}{f_4}$$

$$P_R = P_{RN} \cdot f_3 \cdot \Phi$$

P power to transmit [W]

P_{RN} Single strand chain power rating for normal operation conditions [W]

f_1 Service factor [-]

f_2 Sprocket size factor [-]

f_3 Chain strands factor [-]

f_4 Lubrication factor [-]

f_5 Center distance factor [-]

f_6 Ratio factor [-]

f_7 Service life factor [-]

Φ Chain construction factor [-]

The chain power rating is calculated from empirical power rating equations that are unique for the chain. These equations are mentioned in national standards for the steel chains or they come from ACA (American Chain Association) research. These equations may result in different power capacity than what chain producers publish for their chains.

In general, the power rating equations provide valid a power rating for chain drives that work under specific normal operation conditions. If your chain drive works under a working condition that does not equal the normal operation conditions, the generator automatically adjusts the power rating factors accordingly.

More details on power rating factors can be found in the Calculation basics chapter. They are computed with respect to commonly used practice and normal operation conditions.

The power capacity of the chain drives operating within normal conditions is limited by:

- Link plate fatigue P_{R1}
- Roller and bushing impact fatigue P_{R2}
- Galling between the pin and bushing P_{R3}

Example of chain power rating equations

$$P_{R1} = 0.0044 \cdot z_S \cdot n_S^{0.96} \cdot p^{(3-0.7 \cdot p)}$$

$$P_{R2} = \frac{17 \cdot z_S \cdot p^{0.8}}{n_S^{1.5}}$$

$$P_{R3} = 6.452 \cdot z_S \cdot p^2 \cdot \left(\frac{z_S^3 \cdot n_S^3 \cdot p^5 \cdot (2 + 0.03226 \cdot z_S)}{3.96 \cdot 10^{12}} \right)$$

$P_{RN} = \min (P_{R1} ; P_{R2} ; P_{R3})$

where:

P_{R1} Chain drive power capacity limited by link plate fatigue [hp]

P_{R2} Chain drive power capacity limited by roller and bush impact fatigue [hp]

P_{R3} Chain drive power capacity limited by pin-bush galling [hp]

z_S Number of teeth on the small sprocket [-]

n_S Speed of the small sprocket [rpm]

p Chain pitch [inches]

NoteUsing options in the More section of the Calculation panel, the Chain generator allows a neglect lubrication factor for capacity limited by the link plate fatigue P_{R1} and roller and bush impact fatigue P_{R2}. Only chain power capacity limited by pin-bush galling P_{R3} is then affected by lubrication factor f_4. Then the design power and the resultant chain power capacity are determined as follows:

$P_D = P\ f_1\ f_2\ f_5\ f_6\ f_7$

$P_{RN} = \min (P_{R1} ; P_{R2} ; P_{R3})$

$$P_R = P_{RN} \cdot f_3 \cdot \Phi$$

Chain bearing area pressure

During the chain drive service the fluctuating tensile load acts on contact surfaces between pins and bushings what causes specific size of chain bearing area pressure. If this pressure exceeds the permissible pressure in chain bearing area, the chain service life might significantly decrease and the strength check fails. The following equation is checked to pass the strength check successfully:

$$p_B \leq p_0 \cdot \lambda$$

The amount of the actual pressure in the chain bearing area is computed from maximum tension in chain span as follows:

$$p_B = \frac{F_{T\max}}{A}$$

The permissible pressure in the chain bearing area is determined as

$p_0 = p_{B0} \cdot \varphi$

where:

p_B	Actual pressure in chain bearing area [Pa]
p_{B0}	Specific permissible pressure in chain bearing for normal operating conditions [Pa]
p_0	Permissible pressure in chain bearing area for normal operating conditions [Pa]
F_{Tmax}	Maximum tension in taught chain span [N]
A	Chain bearing area [m^2]
ϕ	Chain construction factor [-]
λ	Specific friction factor [-]

Expected service life analysis

The program checks expected service life for

- given chain elongation t_h
- link plate fatigue impact t_{hL}
- roller and bush impact fatigue t_{hR}

The strength check succeeds if the required service life is equal or less than any expected service life

$$L_h \leq \min\left(t_h; t_{hL}; t_{hR}\right)$$

Expected service life for given chain elongation

The chain elongates throughout the service life because of the wear. The expected service life when the chain elongation reaches 3% is determined from the following empirical equation

$$t_{h3\%} = 2744 \cdot \left(\frac{f_C \cdot f_m \cdot f_k}{p_B} \right)^3 \cdot \frac{X}{v} \cdot \frac{z_1}{\frac{z_1}{z_2}+1} \cdot \frac{p}{\pi \cdot d_2}$$

where:

$t_{h3\%}$ Expected service life for chain elongation of 3% [hr]

f_C Wear factor [-]

f_m Specific chain size factor [-]

f_k Chain speed factor [-]

X Number of chain links [-]

v Chain speed [m/s]

z_1 Number of small sprocket teeth [-]

z_2 Number of small sprocket teeth [-]

p Chain pitch [m]

d_2 Chain pin diameter [m]

p_B Pressure in chain bearing area [N/cm^2]

The chain drive with three or more sprockets is substituted with virtual chain drives consisting from just two sprockets. The resultant service life is determined as follows. The chain bearing area pressure is then specific for taught span in each individual virtual chain drive.

$$t_{h3\%} = \frac{1}{\frac{1}{t_{h1}} + \frac{1}{t_{h2}} + \frac{1}{t_{h3}} + \ldots + \frac{1}{t_{hn}}}$$

where:

$t_{h3\%}$ Expected service life of chain drive for chain elongation of 3% [hr]

t_{h1} ... t_{hn} Expected service life of virtual chain drive for chain elongation of 3% [hr]

The expected service life for specific elongation what differs from 3% is determined as

$$t_h = t_{h3\%} \cdot \frac{\Delta L_{max}}{0.03}$$

where:

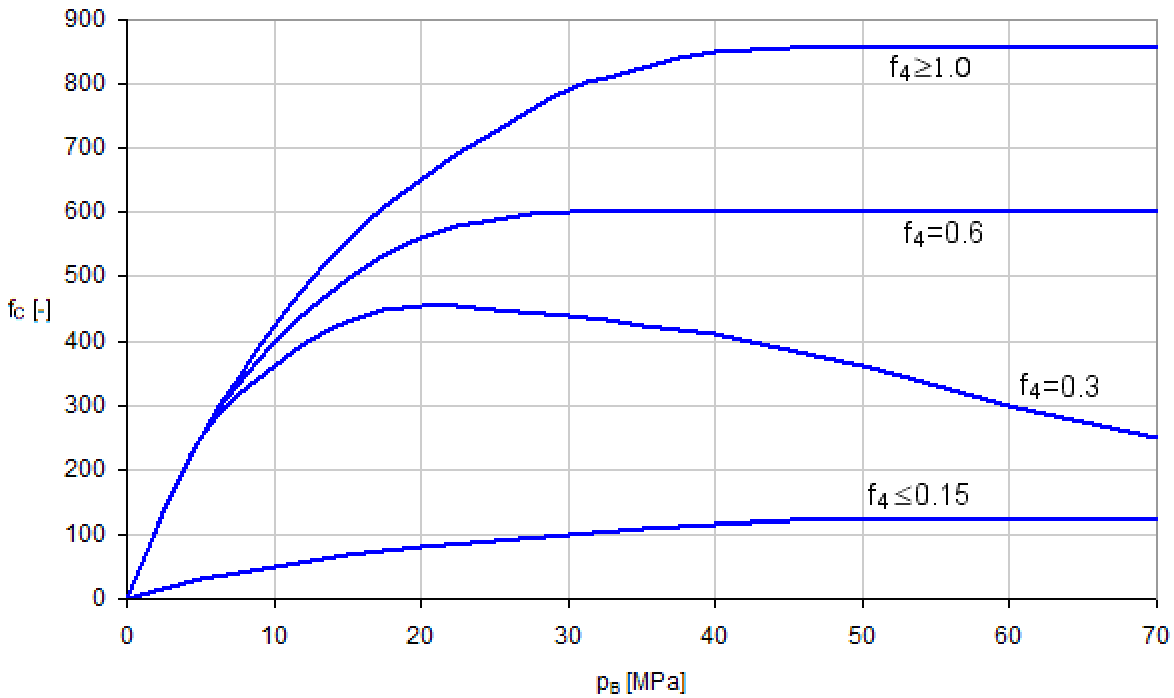
$t_{h3\%}$ Expected service life of chain drive for chain elongation of 3% [hr]

t_h Expected service life of chain drive for given chain elongation [hr]

ΔL_{max} Maximum chain elongation [-]

Wear factor f_C

Wear factor takes into account quality of lubrication and how much it has an impact on the chain wear progress. The size of the wear factor is determined from the following chart with respect to size of lubrication factor f_4 and bearing area pressure p_B .



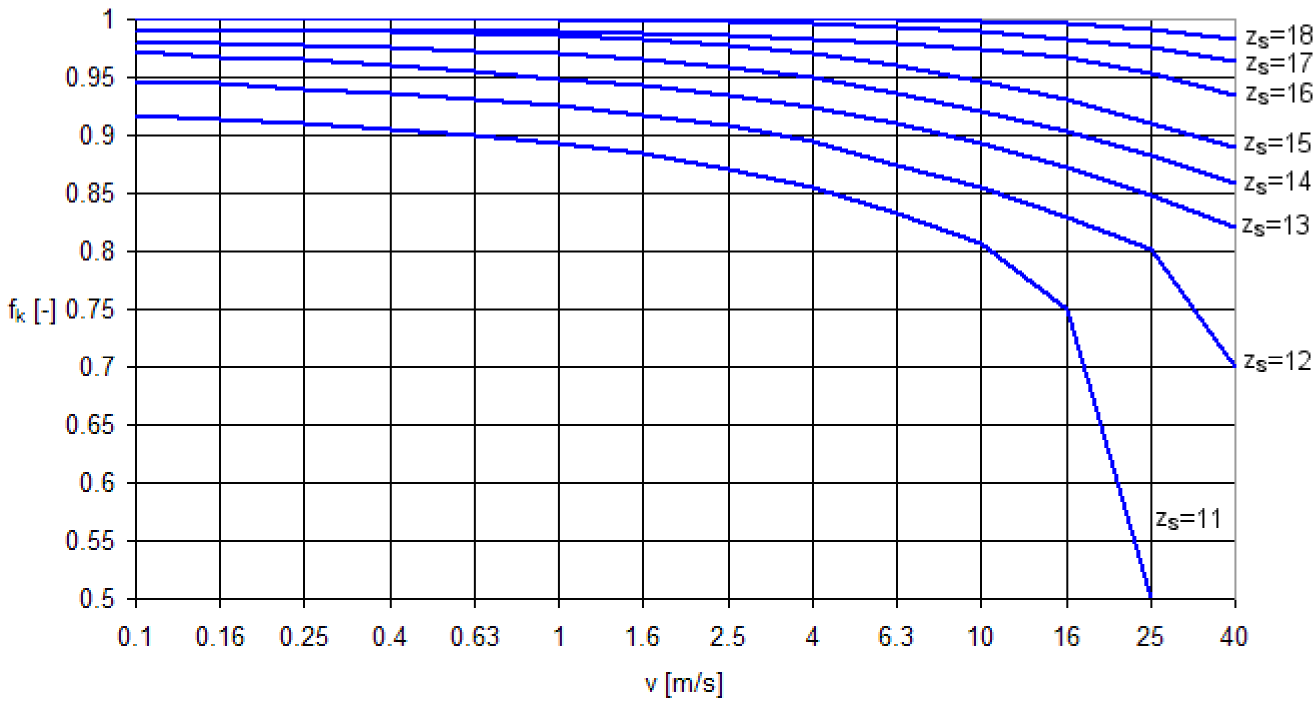
Specific chain size factor f_m

The chain size factor takes into account size of the chain and its impact to the wear progress. The size of the factor is determined from the following table.

Pitch [mm]	4	5	6	6.35	8	9.525	12.7	15.875	19.05	25.4	31.75	38.1	44.45	50.8	63.5	76.2
f_m [-]	1.64	1.57	1.54	1.53	1.49	1.48	1.44	1.39	1.34	1.27	1.23	1.19	1.15	1.11	1.03	0.96

Chain speed factor f_k

Chain speed factor takes into account chain speed v [m/s] for a specific number of teeth of the smallest sprocket z_s [-]. If the smallest sprocket within the drive has 19 or more teeth the factor always equals to one. If smallest sprocket has less than 19 teeth the speed factor is found in the following chart.



Expected service life due to link plates fatigue

The expected service life without link plate fatigue failure is determined from following empirical equation

$$t_{hL} = \frac{X}{n_S} \cdot f_Z \cdot \left(f_Y \cdot \frac{F_U}{F_P \cdot f_1} \right)^{10}$$

where:

t_{hL} Expected service life due to link plate fatigue [hr]

X Number of chain links [-]

n_S Speed of the smallest sprocket [rpm]

f_Z Teeth factor [-]

f_Y Specific chain size factor [-]

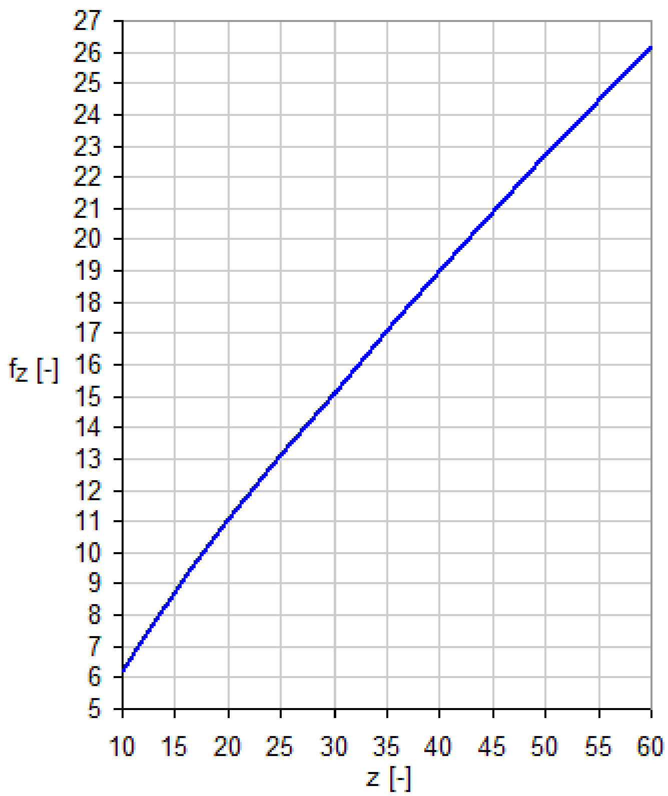
f_1 Service factor [-]

F_U Ultimate tensile strength of the chain [N]

F_P Effective chain pull or tensile load [N]

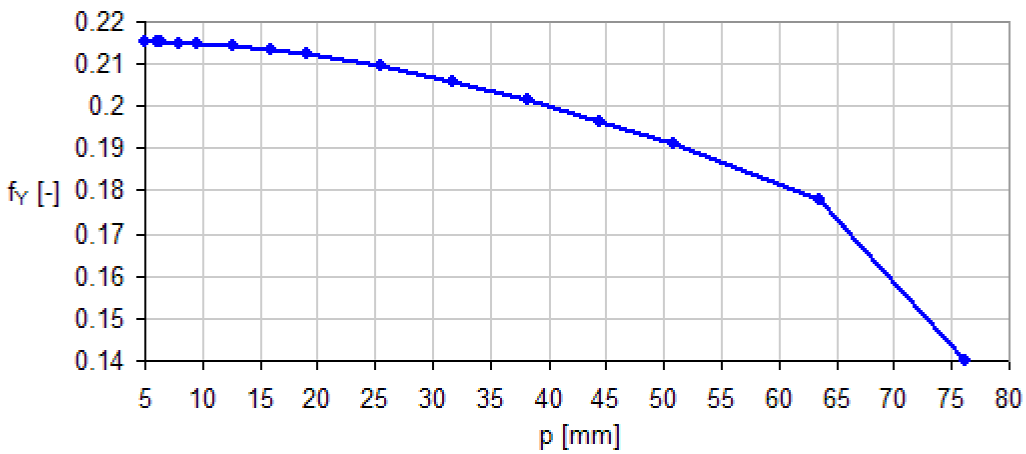
Teeth factor f_Z

The teeth factor takes into account modification of service life caused by size of the smallest sprocket in the chain drive. The size of the factor is defined by following chart.



Specific chain size factor f_Y

The factor takes into account chain size with respect to peak overloads. The size of the factor is defined by following chart.



Expected service life due to roller and bush impact fatigue

Expected service life without roller and bush impact fatigue failure is defined by following empirical equation:

$$t_{hR} = 2.9 \cdot 10^4 \cdot \frac{X \cdot z_S}{n_S} \cdot f_3 \cdot \sqrt[3]{\left[\frac{1}{P \cdot f_1} \cdot \frac{(d_1 - d_2) \cdot b_1}{p}\right]^2}$$

where:

t_{hR} Expected service life due to roller and bush impact fatigue [hr]

X Number of chain links [-]

z_S Number of teeth of the smallest sprocket [-]

n_S Speed of the smallest sprocket [rpm]

f_1 Service factor [-]

f_3 Strand factor [-]

P Power [W]

d_1 Chain roller diameter [m]

d_2 Chain pin diameter [m]

p Chain pitch [m]

Chains vibration

Roller chain can vibrate noticeably when the frequency of an exciting source is close to one of the natural frequencies of the chain. Under certain conditions, the vibration may be so severe that it can damage or destroy the chain or the drive. The major sources of excitement are large cyclic loads, chordal action, and roller-tooth impact.

The natural chain frequencies are computed and transformed to the driver sprocket speeds for better understanding. These speeds are named as critical speeds. If the driver sprocket speed is close to one of the critical speeds the vibrations may occur. The critical interval of driver sprocket speed is determined as follows:

$$n_C \in \langle n - n \cdot \Delta n; n + n \cdot \Delta n \rangle$$

where:

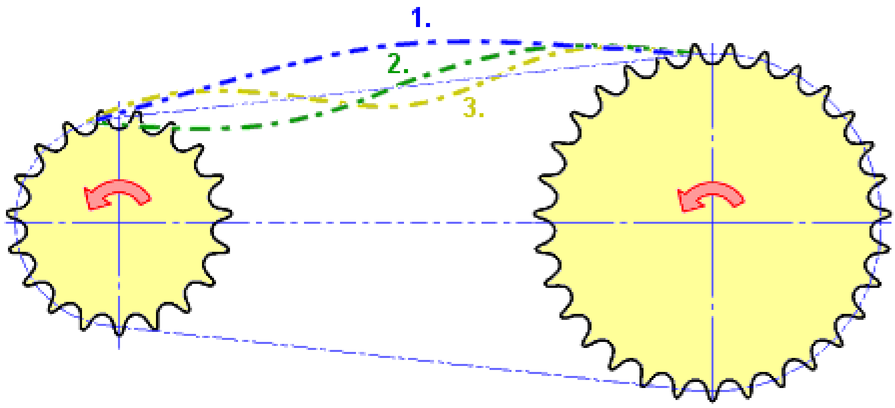
n_C Critical speed of driver sprocket [rpm]

n Actual speed of driver sprocket [rpm]

Δ_n Limit of critical speed [-]

Lateral vibration

In lateral vibration, the chain vibrates up and down (in a horizontal drive) about the chain's axis like a plucked string. It is the most visible, and may be the most common, type of vibration. The natural frequency for lateral vibration is low. The excitation from polygonal effect and large cyclic load may be enough to cause damaging vibration at resonance. The lateral vibration is computed for every span in the chain drive and compared with the current driver sprocket speed.



The critical driver sprocket speed for lateral vibration at each span of the chain drive is given by following equation:

$$n_{cL} = \frac{\lambda \cdot f_{TZ} \cdot 30}{L_T \cdot z_1} \cdot \sqrt{\frac{F_T}{m}}$$

where:

n_{cL} Critical driver sprocket speed from natural frequency of lateral vibration [rpm]

λ An integer represent the harmonic of the vibration [-]

L_T Length of the chain span [m]

F_T Tension in the chain span [N]

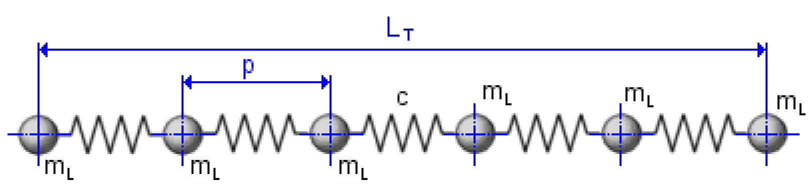
f_{TZ} Teeth factor [-] (f_{TZ} = 1.2 for $z_1 < 18$ otherwise f_{TZ} = 1.1)

z_1 Number of driver sprocket teeth [-]

m Specific chain mass [kg/m]

Wave-type vibration

In wave-type vibration, the chain vibrates axially like an elastic bar that is excited at its ends. Wave-type vibration usually cannot be seen. Wave vibration can increase chain tension quite a lot and cause early chain failure. Damaging wave-type vibration can also occur when the tooth contact frequency matches the second harmonic of the natural frequency of the chain.



The critical driver sprocket speed for wave-type vibration at each span of the chain drive is given by following equation:

$$n_{cW} = \frac{\lambda \cdot 30}{L_T \cdot z_1} \cdot \sqrt{\frac{c \cdot L_T}{m}}$$

where:

n_{cW} Critical driver sprocket speed from natural frequency of wave-type vibration [rpm]

λ An integer represent the harmonic of the vibration [-]

L_T Length of the chain span [m]

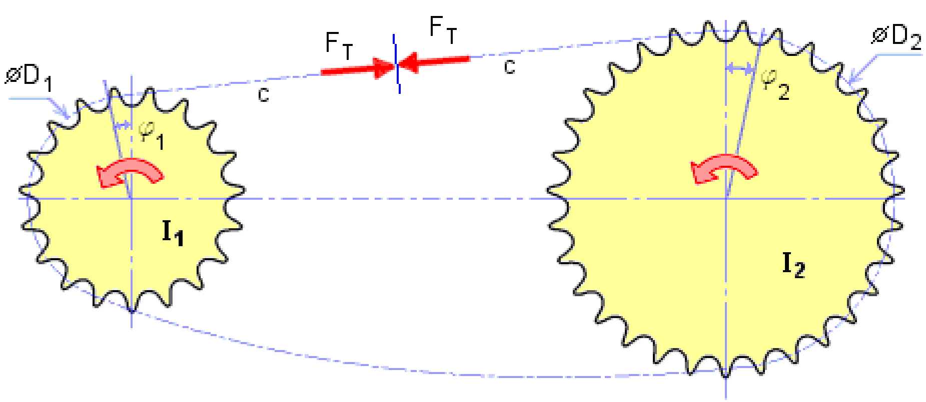
z_1 Number of driver sprocket teeth [-]

c Chain stiffness [N/m]

m Specific chain mass [kg/m]

Axial, or spring-type, vibration

In the axial vibration, the chain acts like a spring connected between to rotors. This type of vibration is not readily seen, but at resonance it may be identified by increased noise. The impulse from a large cyclic load may be enough to cause damaging vibration at resonance.



Critical driver sprocket speed from polygonal effect

$$n_{cAca} = \frac{15}{\pi \cdot z_1} \cdot \sqrt{c \cdot \left(\frac{D_1^2}{I_1} + \frac{D_2^2}{I_2}\right)}$$

Critical driver sprocket speed from circumferential run-out

$$n_{cAcr} = \frac{15}{\pi} \cdot \sqrt{c \cdot \left(\frac{D_1^2}{I_1} + \frac{D_2^2}{I_2} \right)}$$

Critical driver sprocket speed from inaccuracy of chain link pitches

$$n_{cAip} = \frac{7.5}{\pi \cdot z_1} \cdot \sqrt{c \cdot \left(\frac{D_1^2}{I_1} + \frac{D_2^2}{I_2} \right)}$$

where:

n_{cAca} Critical driver sprocket speed from polygonal effect [rpm]

n_{cAcr} Critical driver sprocket speed from circumferential run-out [rpm]

n_{cAip} Critical driver sprocket speed from inaccuracy of chain link pitches [rpm]

z_1 Number of driver sprocket teeth [-]

c Chain stiffness [N/m]

D_1 Pitch diameter of input sprocket [-]

D_2 Pitch diameter of output sprocket [-]

I_1 Rotating moment of inertia related to the input sprocket [kg m^2]

Rotating moment of inertia related to the input sprocket [kg m^2]

I_2

Web: www.irinventor.ir
Email: irinventor@chmail.ir
& irinventor@hotmail.com

Tel: 09352191813 & 021-46088862

Iranian Group INVENTOR