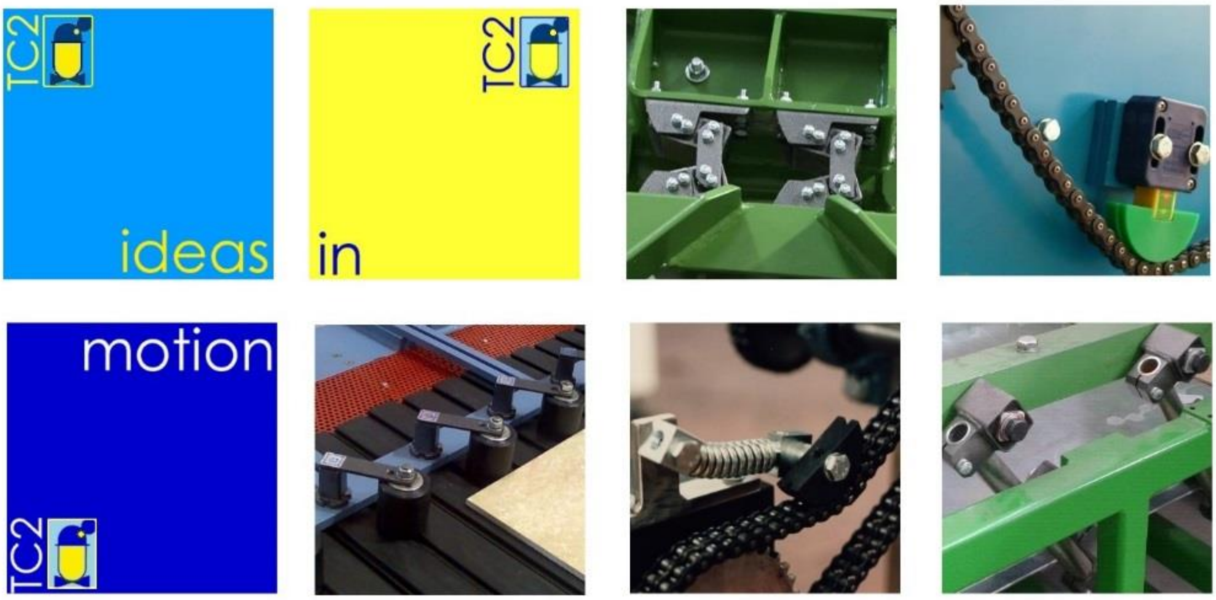
TC2 TECNIDEA CIDUE S.r.l.

...ideas in motion...

QUALITY SYSTEM CERTIFICATION DNV·GL ISO 9001



Giornale – Newspaper – Zeitung N. 05 June 2014

National commitment to international development.

Every day we are bombarded by bad news arising from a bad information; newspapers, magazines, TV and general information media overwhelm us with negative data only, it seem that only bad news makes audience. I believe this is not true, in fact the real world, at least in our sector in the last period, noted a completely opposite trend and almost all companies with we came in contact had a good 2013 and started 2014 well.

Us of **TC2** are satisfied both for 2013 and for 2014; certain that our commitment is always more international taking advantage from benefits of European Community UE grants us.

Blue of Europe is always been our reference color and the 12 yellow stars are directing us daily our membership in this big market that it is even our big country, when every day our employees and collaborators are confronted with our customers and suppliers, they work and create new ideas for a better and united world. Cheers so blue of Europe and its magnificent 12 stars.

The Europe lead to the elimination of the borders of the old small national states creating a new big country that it generated a new large market; thanks to this Tecnidea Cidue could develop and expand itself in new product sectors.

Thanks to the big push of the European Community EU and its liberal politics we could deal with the world markets with more energy, that they have allowed us to get in touch with many new traders and constructors.

All of this energy allowed us to develop our production with new technological and application solutions, but particularly to design and to create new products and new lines of application for countless working solutions.

BLUE is surely the color to which we are linked; BLUE is the color of mechanic, BLUE is even the color of the gowns from the work of workshop but BLUE and YELLOW are also the colors of our city "Verona", in fact here **TC2** has its headquarters. It is in this area that Tecnidea Cidue was born, here it grew up and in this framework that it continues to produce millions of technical mechanical products with a strong propensity to the world market.

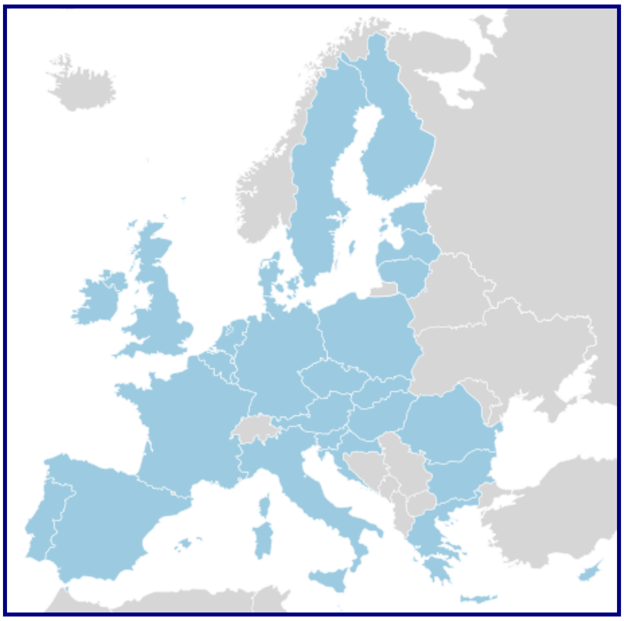
TC2 exports about 70% of its production. This area, the legendary north-east, has supported us in all of these years in our work and it is even thanks to tens of small and medium enterprises that we could get the results obtained.

In fact, this area is mostly composed of master craftsmen who made precision engineering their reason for living and this allowed us to exalt our creativity.

Thanks to collaborators, thanks to suppliers, thanks to Verona, thanks to North/East, thanks Europe, thanks BLUE and YELLOW; you like our flag keep us united and you daily give us the energy to continue hard and serenity our work that it is always in any case projected into the future.

Franco Canova
General Manager

EURPEAN UNION



28 MEMBER COUNTRIES

DISCOVERING TECNIDEA CIDUE…

THE CHAIN. PART II.

DIMENSIONING OF A CHAIN TRANSMISSION: BASIC FORMULA.

Kinematics of a chain transmission.

The links of a chain realize on the sprocket a polygon (picture 8) and make, during the drive, an L-shaped movement.
In this manner the real values change, with a uniform rotation of the sprockets you can have an irregular speed on the section (polygonal effect).
La velocità lineare della catena varia tra i seguenti valori limite:

$$v_{max} = \frac{p \cdot n \cdot \pi}{6 \cdot 10^4 \cdot \sin\frac{\pi}{2}} \quad [m/s]$$

where $\frac{\tau}{2} = \frac{180^\circ}{z}$

$$v_{min} = \frac{p \cdot n \cdot \pi}{6 \cdot 10^4 \cdot tg\frac{\pi}{2}} \quad [m/s]$$

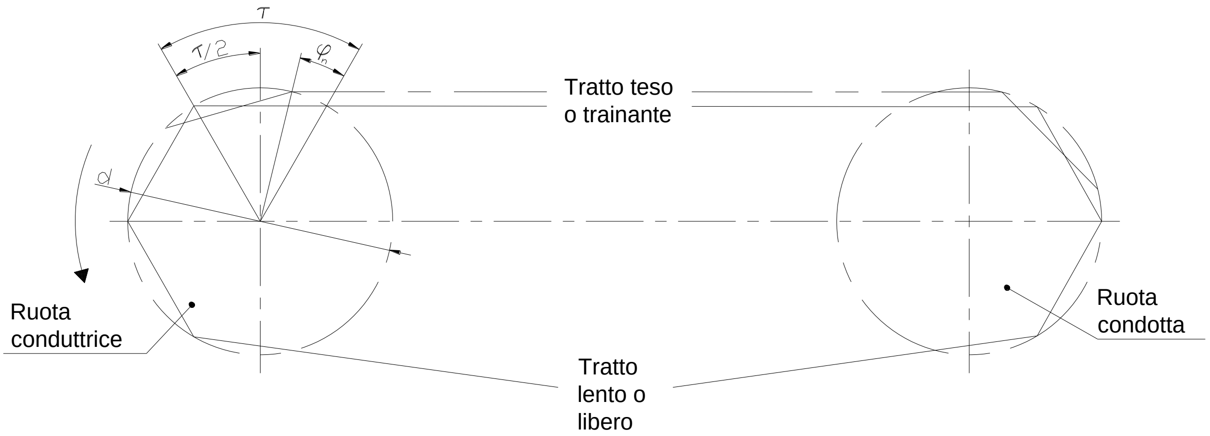
The medium linear speed of the chain, used also for the dimensioning of the chain transmission, is calculated with the following equation:

$$v = \frac{p \cdot n \cdot z}{6 \cdot 10^4} \quad [m/s]$$

The duration of an irregularity period is given by:

$$T = \frac{60}{n \cdot z}$$

By the above mentioned equations it is possible to notice that the irregularity of the chain speed depends by the number of teethe of the sprockets; the higher the number of teethe, the lower the irregularity



Dis. 8

Following the polygonal effect, the chain suffered continuous accelerations and brakings.
The maximum acceleration a_{max} is equivalent to:

$$a_{max} = \frac{2 \cdot 10^4 \cdot v^2}{p \cdot z^2} \quad [m/(s^2)]$$

The maximum braking f_{max} is the opposite of the maximum acceleration so that the medium speed remain constant.

$$f_{max} = -\frac{2 \cdot 10^4 \cdot v^2}{p \cdot z^2} \quad [m/(s^2)]$$

Chain transmission dynamic.

The traction static force of the chain is equal to:

$$F = \frac{P}{v} \; [N]$$

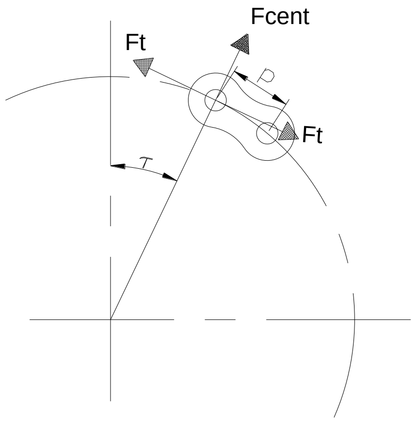
Where:

P [kW] = absorbed power
v [m/s] = Medium linear speed of the chain

We define also the following dimensions that will be used subsequently in the dimensioning:

- Centrifugal traction force: F_t;
- Traction Vertical load: F_{st};
- Pulsating and impact forces deriving by the irregularity that act on the towing section and the towed section.

- The centrifugal traction force F_t (Picture 9) acts as re-action of the radial centrifugal force in 2 sections of chain, depend by the linear weight of the chain [kg/m] and the square of the medium speed of the chain [m/s²].



Dis. 9

- The vertical traction force "F_{st}" (Fig. 10) acts in the towing section and in the free section.
  It depends by the weight of the section "q x L_r" and by the loosening "h_d", besides to the inclination angle δ of the same section.
  In the rightly assembled chain transmission, the loosening must be normally equal to 1-2% and it does not effect on the vertical traction power.
  However, it can reach very high value if the transmission has high centre to centre distances or if the chain is too stretched, also due to the effect of a tensioner.
  Vertical traction forces on the upper and lower gears ($F_{st,s}$ and $F_{st,i}$) are different, except when the inclination angle is null (δ=0).
  These forces are calculated by the following formulae:

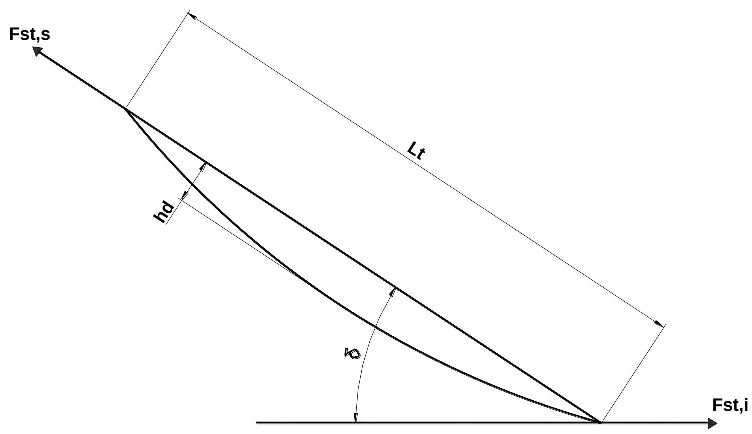
$$F_{st,s} = 10^{-3} \cdot g \cdot q \cdot L_r \cdot (\xi + \sin\delta) \quad [N]$$

$$F_{st,i} = 10^{-3} \cdot g \cdot q \cdot L_r \cdot \xi \quad [N]$$

$$h_r = \frac{h_d}{L_r} \cdot 100 \; [\%]$$

q = weight of the chain per linear meter [N]
h_d = easing of the free section [N]
h_r = relative easing

L_r = Lenght of the chain section [mm]
g = 9,81 [m/s²]
δ = inclination angle of the chain section [mm]
ξ = specific vertical traction force



Dis. 10

In the case of horizontal position of the driving portion and relative easing "$h_r \leq 10$ %", the vertical load of traction can be approximately calculated as:

$$F_{st} = 9{,}81 \cdot \frac{q \cdot L_r^2}{8000 \cdot h_d} \quad [N]$$

Driving groups and/or driven groups generate in chain drives pumping forces and impact due to irregularities.
These forces are considered in calculations through the coefficients of impact.
Due to the polygonal effect, other pumping and impact forces occur, that are considered within the coefficient of the number of the sproket's teeth.
Always due to the polygonal effect, it can happen that during the meshing of the chain on the sprocket, the rollers could bump the teeth causing typical noises of the chain.

- Impact force "F_a" limits the duration of rollers and compasses, accelerating the wear of the teeth.
To calculate impact forces is used the following equation:

$$F_a = 1662 \cdot \sqrt{B_1 \cdot q \cdot p} \cdot \frac{v}{z} \cdot \sin\left(\frac{360°}{z} + y\right) \quad [N]$$

Where:
B_1 = band width of the tooth [mm]
q = weight of the chain per linear meter [kg/m]
p = rate of the chain [mm]
v = linear speed [m/s]
z = N° of teeth
γ = pressure angle [°]
With high speed "v", "F_a" can acquire considerable values.

- Pumping speed "F_p" will be reduced from one tooth to another in the set of teeth of the drive gear as following:

$$\frac{F_{pi}}{F_{pi+1}} = \frac{\sin(\tau + \gamma)}{\sin \gamma}$$

Where:
F_{pi} = strength of the chain link in grip;
F_{pi+1} = strength of the link (i+1) gripping;
z = N° of teeth;
γ = pressure angle;
τ = step angle: (360°)/2;
z_e = N° of teeth in grip.

In the free section, it remains the residual force "F_r", with an opposite direction to the vertical tensile load "F_{st}".
The vertical tensile load should be slightly higher than the residual force, otherwise it could lift the chain from the location of the tooth.
Differently the chain has to be tightened.

$$F_r = F_p \cdot \left[\frac{\sin \gamma}{\sin(\tau + \gamma)}\right]^{ze} \quad [N$$

Geometry of the chain drive.

It's important, for the setting of a chain drive, the correlation between interaxis and the n° of chain links (X), with a specific gait (p) and a specific n° of teeth (z_p, z_c).
Further parameters are the b_1 e b_2 winding angles, that are on the gears and the release of the free section.

Calculation of the chain's length.

The chain's length (considering the n° of its links) can be calculated as following:

$$X_0 = \frac{2 \cdot a_0}{p} + \frac{z_1 + z_2}{2} + \left(\frac{z_2 \cdot z_2}{2 \cdot \pi}\right)^2 \cdot \frac{p}{a_0}$$

The n° of the links which results, must be rounded to a finite number, preferably even.
Odd number of links should be avoided, in as much as it would require the use of a fake link that should reduce the duration of the chain.
The calculation of the development of the chain with 3 or more gears is complex.
According to the n° and the disposition of the gears, case by case, correspondent equations should be arranged.

TC2 NEWS

Calculation of the interaxis

For the setting of the chain driver with a pre-established n° of "X" links, it is necessary an exact calculation to define the interaxis:

$a = [2 \cdot X - (z_1 + z_2)] \cdot p \cdot f_4$ [mm] with: $f_4 = \dfrac{1}{4 \cdot \sin\delta(\hat{\delta} + \cot g\delta)}$

And can be calculated with: $\dfrac{X - z_1}{z_2 - z_1}$ using technical tables.

Choice of the transmission through power diagrams

Power diagrams offer an overview on the field of the roller chains power, according to the norm DIN 8187-8188 and the compasses chains DIN 8154.
The power is graphically pictured according to the n° of the sproket's turns.
These diagrams permit an immediate evaluation of the possible type of chain, according to the maximum power consumption.
As acceptable criterion has been considered the contact wear of the joints and the corresponding extension of the chain itself.
The above-mentioned diagrams, valid for the majority of the chain drive, are trustworthy with a high certainty of working.

Gears: z_p, z_c = 19 teeth;
Length: X = 100 links;
Ratio: i = 1;
Duration: t_h = 15000 [h];
Lubrication: tab.

Lubrication field	1	2	3	4
Type of lubrication	manual	Drip-through	Oil bath	Forced circulation

Ratio that are different than "i = 1" are considerate with a reduction coefficient (f_i):

i	1:1	2:1 , 1:2	3:1 , 1:3	4:1 , 1:4	5:1 , 1:5
f_i	1	0,87	0,82	0,79	0,77

The coefficient (f_i) depends on the n° of the sproket's teeth and on impacts due to the functioning:

z	Uniform functioning	Moderate impacts	Medium impacts	Strong impacts
11	0,55	0,41	0,34	0,32
13	0,66	0,49	0,41	0,39
15	0,77	0,57	0,48	0,45
17	0,88	0,64	0,54	0,51
19	1,00	0,74	0,63	0,59
21	1,11	0,82	0,69	0,65
23	1,23	0,91	0,77	0,72
25	1,35	1,00	0,84	0,79
30	1,64	1,22	1,02	0,97
35	1,93	1,44	1,21	1,14
38	2,11	1,56	1,32	1,24
40	2,24	1,66	1,40	1,32
45	2,54	1,88	1,59	1,49
50	2,84	2,10	1,78	1,67
57	3,28	2,43	2,06	1,93
60	3,46	2,56	2,16	2,04

The elongation caused by the usury of the chain, is directly proportional to the length of the chain itself. If the chain's length is different from X=100 links, the service duration will be different from 15000 [h] too, it will rise or decrease.

Calculation of chain drive's duration.

There are two fundamental criteria to calculate a chain drive's duration:

- Usury resistance:
  The duration of the chain drive is determined by the duration of the chain itself.
  In the chain, that is prone to stress, occurs a swinging movement of the joints, which causes abrasion on linchpins and compasses.
  That entails an increase of games an the resulting extension of the chain.
  To determine the chain's length, we need to know the following data:
  rate of internal links, rate of external links, games forming between linchpins and compasses.
  The maximum allowable usury extension is 3% of the original length of the chain.
  Δ_{lmax} = maximum allowable extension;
  l_0 = original length of the chain;
  l_{costr} = allowable building length of the chain.

$$\Delta l_{max} = l_{costr} - l_0 \leq 3\%(l_0)$$

The friction that occurs in chain's joints is determined by the friction angle, by the compression of the junction surface, and by lubrication.
The friction angle is determined by the swinging angle ($t = 360°/z$) and by the swinging radius $d_2/2$ (with d_2=linchpin-chain diameter).
The chain's duration, with maximum allowable extension on 3% it can be calculated by the following equation:

$$t_h = 2744 \cdot \left(\frac{f_c \cdot f_m \cdot f_k}{p_r} \right)^3 \cdot \frac{X}{v} \cdot \frac{z_p}{\frac{z_p}{z_c} + 1} \cdot \frac{p}{\pi \cdot d_2} \quad [h]$$

If the chain has 3 or more gears, each section's duration $t_{h,i}$ must be calculated individually with its corresponding n° of teeth whereas must be indicated the total n° of the chain's links.

$$t_h = \frac{1}{\Sigma_i \frac{1}{t_{hi}}} \quad [h]$$

The usury of the chain, regardless the running-in period, is proportional to the 2 hours of the chain's operation.
If for particular reasons the maximum allowed extension has to be shorter than 0,03-l_0, then it becomes:

$$t_{hx} = t_h \cdot \frac{\Delta l_x}{\Delta l_{max}}$$

- Resistance of the fatigue operation:
  The fatigue resistance of the components that form a chain drive considerably affect the duration of the chain drive itself.
  The concept of FATIGUE RESISTANCE describes the extent of dynamic loads, with which the components can be stimulated till a defined n° of variations of the load, without breaks.:
  <u>Slabs and linchpins fatigue resistance duration:</u>
  When the linear speed of the chain is smaller than 1 m/s, in conditions with enough lubrication, the duration is influenced mostly by the weariness of plates and of the chain rollers.

$$t = \frac{X}{n} \cdot f_z \cdot \left(f_y \cdot \frac{F_b \cdot y}{F} \right)^{10} \quad [h]$$

<u>Rollers and compasses duration in the field of operational fatigue resistance:</u>
During the wrapping on the sproket, rollers and compasses hit the sproket's tooth profile.
The impact force (F_a) can reach high values due to chain's speed and the number of teeth.
That's the reason why the usury of rollers and compasses can limit the duration of the chain drive, in case of speed equal to 10 m/s or higher and with low n° of teeth.
The chain's duration can be calculated through the following equation:

$$t_h = 2{,}9 \cdot 10^4 \cdot \frac{X \cdot z}{n} \cdot f_v \cdot \sqrt[3]{\left[\frac{y}{P} \cdot \frac{(d_1 - d_2) \cdot b_1}{p} \right]^2} \quad [h]$$

con: $P = \frac{F \cdot v}{1000}$ e $F_d = \frac{F}{y}$

Calculation example.

Setting a chain drive with the following features:

1. EP = 145 [kW] = Engine Power;
2. $n_{.d}$= 800 [1/min] = n° of turns of the driveshaft;
3. $n_{a.c.}$ = 200 [1/min] = n° of turns of the non-drive shaft;
4. i = 4 = ratio;
5. Impact factor: Y = 2 (from tab. 8 I extract: y = 0,73);
6. Lubrication on forced movement and chain tensioners on the free section;
7. Requested duration: t = 20000 [h];
   The architectural area is limited;
   Sproket's obstruction, including the chain's wrapping = 1000 [mm];
8. Interaxis = 1200/1300 [mm]

Procedure:

Ratios factor: i = 4 so f_i = 0,79;
N° of teeth of the engine sproket: z = 19;
Service coefficient (f_1 per z = 19): f_1 = 0,74.

Needed power:

$$P_n = P \cdot \frac{f_i}{f_1} = 154{,}8 \quad [kW]$$

The chosen chain
(according to the power diagram) is:

24 B-3 DIN 8187 (1"1/2 triple chain with rate corresponding a 38.1 mm).

Data:

a. Minimum breaking load: F_b = 425000 [kN];
b. Rate: p = 38,1 [mm];
c. Articulation surfaces: f = 16,63 [cm^2];
d. Chain weight/m: q = 21,0 [kg/m];
e. Internal width: b_1 = 25,4 [mm];
f. Rollers diameter: d_1 = 25,4 [mm];
g. Linchpins diameter: d_2 = 14,63 [mm];
h. Plates height: h = 33,4 [mm];
i. Sproket: z_p = 19 denti;
j. Crown: z_c = 76 denti.

Gear chain tensioner: z_t=17 denti

Calculation of the length of n° of chain's links, according an approximate interaxis (a_0) di a_0 = 1250 [mm]:

$$X_0 = \frac{2 \cdot a_0}{p} + \frac{z_p + z_c}{2} + \left(\frac{z_c - z_p}{2 \cdot \pi}\right)^2 \cdot \frac{p}{a_0} = 115{,}63 \quad [maglie]$$

The obtained value will be approximate to 116 [links].
The exact interaxis, considering 116 links, is:

$$a = \left|2 \cdot X - (z_c + z_p)\right| \cdot p \cdot f_4 = 1256{,}95 \quad [mm]$$

"f_4" determined from the tab.

$$\frac{X - z_p}{z_c - z_p} = 1{,}70$$

coefficient "f_4" = 0,24081.

The value of "a" is rounded to 1257 [mm]
The total obstruction of the crown with wrapped chain can be calculated as:

$$D_c = d_c + h = 921{,}96 + 33{,}4 = 955{,}36 \quad [mm]$$

The crown diameter "d_c" is calculated as following:

$$d_c = \frac{p}{\sin\frac{\tau}{2}} = \frac{p}{\sin\frac{360}{2 \cdot 76}} = 921{,}96 \quad [mm]$$

It can be noticed that "d_c" is minor than the maximum allowable obstruction, estimed at 1000 [mm]. Therefore it has been verified that the chosen chain drive can be inserted in the expected amount of space designed.
Verify the expected duration with an extension of the chain on 3% to usury:

$$t_h = 2744 \cdot \left(\frac{f_c \cdot f_m \cdot f_k}{p_s}\right)^3 \frac{X}{v} \cdot \frac{z_p}{\frac{z_p}{z_c} + 1} \cdot \frac{p}{\pi \cdot d_c} \quad [h]$$

Where:
- f_c = usury factor;
- f_m = rate factor;
- f_k = n° of teeth factor;
- p_s = compression on the junction surface.

The linear speed of the "v" chain is calculated through the following formulae:

$$v = \frac{p \cdot z \cdot n}{60000} = \frac{38{,}1 \cdot 19 \cdot 800}{60000} = 9{,}65 \quad [m/s]$$

To determine "p_s" the following equation has to be used:

$$p_s = \frac{F'}{f}$$

Before determine "p_s" , the total traction force of the chain is needed, and it results from:

$$F' = \frac{F}{y} + F_t$$

Where:

- F = chain's traction force (static);
- F_t = centrifugal force.

$$F = \frac{1000 \cdot P}{v} = 15026 \; [N]$$

$$F_t = q \cdot v^2 = 1955{,}6 \; [N]$$

The total traction force results as:

F' = 22539 [N]

The compression of the junction surface is:

p_s = 1355 [N/(cm^2)]

According to the power diagram, the chosen chain drive is included in the lubrication field n°4.

The expected lubrication (forced circulation) goes in the requested lubrication level.

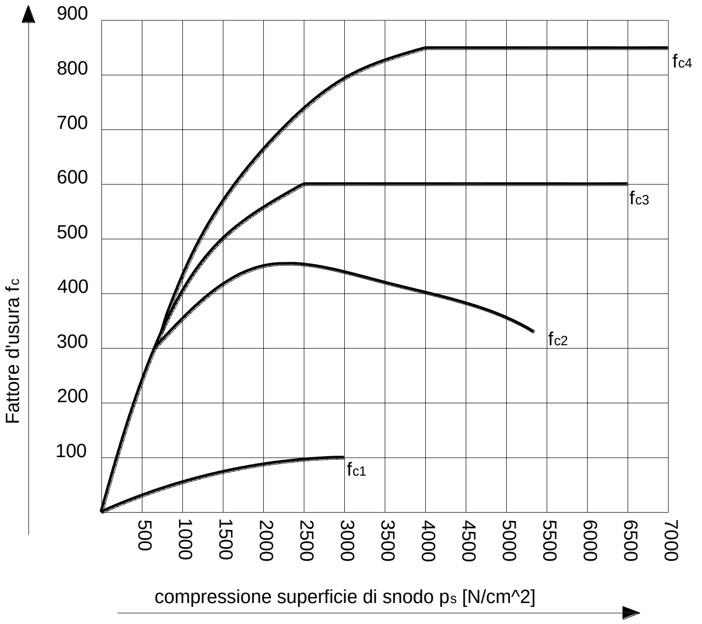
"f_c" , "f_m" , "f_k" factors are coefficients that are empirically obtained and can be acquired with specific diagrams.

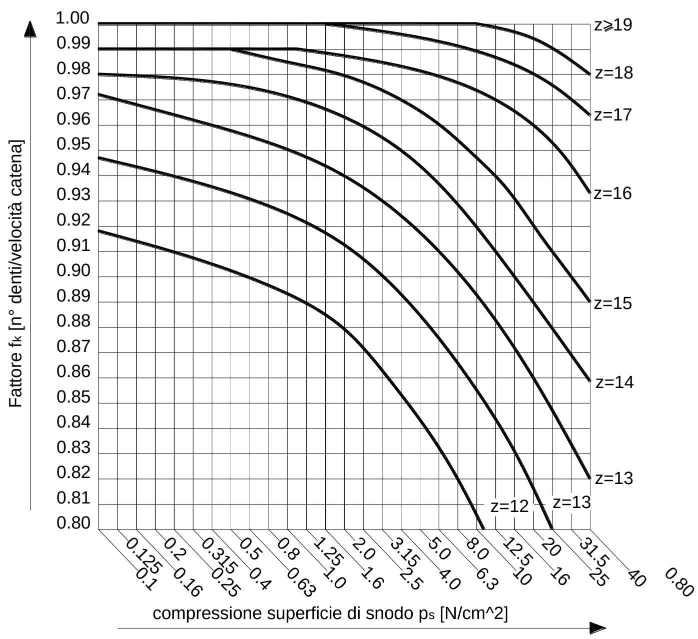
f_m factor (rate chain factor) is obtained with the following tabs:

Chain pitch (mm)	4	5	6	6.35	8	9.525	12.7	15.875
Pitch factor f_m	1.64	1.57	1.54	1.53	1.49	1.48	1.44	1.39

Chain pitch (mm)	19.05	25.4	31.75	38.1	44.45	50.8	63.5
Pitch factor f_m	1.34	1.27	1.23	1.19	1.15	1.11	1.03

f_c factor (usury factor) can be found with the following diagrams:





considering:
f_{c1}= for dry functioning;
f_{c2}= for insufficient lubrication;
f_{c3}= with prescribed lubrication level;
f_{c4}= with lubrication on a higher level than the prescribed one.
f_k factor is obtained by the following diagram and depend on the n° of "z" teeth and the "v" linear speed [m/s].

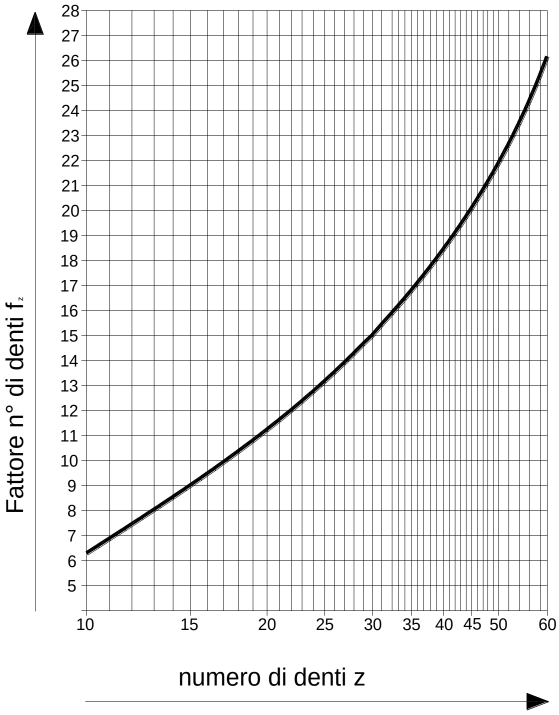
Now calculate the duration according to the fatigue resistance of plates and linchpins:

- Rate factor (fatigue resistance) per p = 38,1 [mm]:
  Tab.: f_y = 0,2014
- N° of teeth factor (fatigue resistance) per z= 19:
  Diagram: f_z = 10,7.

Chain pitch (mm)	5.0	6.0	6.35	8.0	9.525	12.7	15.875
Pitch factor f_y	0.2152	0.2151	0.2151	0.2150	0.2149	0.2145	0.2136

Chain pitch (mm)	19.05	25.4	31.75	38.1	44.45	50.8	63.5
Pitch factor f_y	0.2525	0.2096	0.2058	0.2014	0.1964	0.1909	0.1780

Concernine to f_z factor, should be considered the following diagram:



Expected duration:

$$t_{h1} = \frac{X}{n_{a.m.}} \cdot f_z \cdot \left(f_y \cdot \frac{F_b \cdot y}{F} \right)^{10} = 0{,}145 \cdot 10{,}7 \cdot \left(0{,}2014 \cdot \frac{425000 \cdot 0{,}73}{15026} \right)^{10} = 2399001 \quad [h]$$

Only after these operational hours, plates and pin breakage can happen due fatigue.
We can see now the expected duration of rollers and bushes based on resistance to fatigue.
From the power diagram can be noticed that the n = 800 [turns/min] ordinate crosses the power curve of the chosen chain in the right zone.
It means that the chain's duration is probably limited by the roller breake.
The chain number factor (triple) is obtained by the following tab and is equal to "f_n = 2,5".

N° of rollers	Simplex Chain	Duplex Chain	Triplex chain	Quadruple chain
"f_n" Factor	1,0	1,7	2,5	3,3

The expected duration of the chain can be calculated through the following equotion:

$$t_{hr} = 2{,}9 \cdot 10^4 \cdot \frac{X \cdot z}{n_{a.m.}} \cdot f_n \cdot \sqrt[3]{\left[\frac{y}{p} \cdot \frac{(d_1 - d_2) \cdot b_1}{p} \right]^2} = 2{,}9 \cdot 10^4 \cdot \frac{116 \cdot 19}{800} \cdot 2{,}5 \cdot \sqrt[3]{\left[\frac{0{,}73}{145} \cdot \frac{(25{,}4 - 14{,}63) \cdot 25{,}4}{38{,}1} \right]^2}$$

$t_{hr} = 21736$ [h] hours of the effective duration of the chain.

It can be noticed that the duration of the transmission is influenced by the fatigue resistance of the rollers, that guarantees a chain life higher than 20000 [h] that are requested from the project.

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e Giorgio Canova

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