CALCULATION AND DESIGN OF CHAIN TRANSMISSIONS

Project and calculation of the chain transmission follows standards CSN 01 4809 and DIN 8195.
Chain selection and the first chain project are coming from Diagram 1 and 2, where the curves define each chain efficiency. They are valid only for the chain transmission where the driving wheel is with $z_1 = 19$ teeth and the driven wheel $z_2 = 57$ teeth. The next condition is axis distance a = 40 p (100 p according to DIN standard), $\rho = 1$, $\varphi = 1$, $\mu = 1$, planned durability 10 000 (15 000 according to DIN standard) running hours and the allowed chain prolongation by the wear influence is 2 % (3% according to DIN standard) of the basic chain length.
The running conditions influencing the chain are various in practice, that´s why it is necessary to make recalculation, to compare it with the ideal condition. We modify the value of transformed power P to the value:

$$P_D = \frac{P}{\mu\varphi\chi}$$

χ = coefficient of power from Table 1
φ = coefficient of type

$\varphi = 1$ for the chains mentioned in Diagrams 1 and 2
$\varphi = 0{,}8$ for the chains not mentioned in Diagrams 1 and 2
$\varphi = 1{,}5$ for the long pitch chains according to standard CSN 02 3315 (DIN 8181)
μ = coefficient of the lubrication that you can find it in Table 2 (for details see chapter " Chain lubrication ").

With this recalculated power and with the help of Diagrams 1 or 2 (depending on the chosen chain type) we determine the size of the chain.
Now we devide pre-projected axis distance with the chain pitch and with the help of Table 3 we select the next axis distance correction coefficient. With this coefficient we devide the power P_D and we get the corrigated power P_{1D}.
With the help of the P_{1D} we check or, if it is necessary, correct the selected chain in accordance with Diagrams 1 or 2.

Table 1 Power coefficient χ

Transmission ratio *i*	Shock coefficient ***Y = 1*** Numbers of teeths z_1 of small wheel					Shock coefficient ***Y = 2*** Numbers of teeths z_1 of small wheel					Shock coefficient ***Y = 3*** Numbers of teeths z_1 of small wheel					Shock coefficient ***Y = 4*** Numbers of teeths z_1 of small wheel				
	13	17	19	21	≧ 25	13	17	19	21	≧ 25	13	17	19	21	≧ 25	13	17	19	21	≧ 25
1	(0,39)	0,73	0,83	0,93	1,11	(0,28)	0,53	0,60	0,67	0.81	(0,24)	0,42	0,52	0,58	0,70	(0,21)	0,34	0,43	0,53	0,64
2	0,50	0,82	0,93	1,04	1,26	(0,36)	0,60	0,68	0,76	0,92	(0,30)	0,50	0,59	0,66	0,80	(0,26)	0,44	0,52	0,61	0,73
3	0,57	0,88	1,00	1,12	1,36	0,42	0,65	0,73	0,82	0,99	(0,35)	0,55	0,63	0,71	0,86	(0,29)	0,51	0,58	0,65	0,79
5	0,64	0,96	1,09	1,22	1,49	0,47	0,71	0,80	0,89	1,09	0,40	0,61	0,69	0,77	0,94	(0,33)	0,57	0,63	0,71	0,86
≧ 7	0,67	1,02	1,15	1,30	1,59	0,49	0,75	0,85	0,95	1,16	0,42	0,64	0,73	0,82	1,00	(0,35)	0,59	0,67	0,75	0,92

Running conditions for the values in bars are not suggested.

If table 1 is used it is necessary to count with the index 1 for the small wheel, without regard if it is a driving or a driven wheel.
For the transmissions $i < 1$ is the value χ (λ) subtracted from the upset value $\frac{1}{i}$

Table 2 Lubrication coefficients μ

<table>
<tr><th rowspan="3">Power divides</th><th rowspan="3">Chain speed in ms^{-1}</th><th colspan="4">Lubrication coefficients μ for</th><th colspan="2" rowspan="2">Lubrification types</th></tr>
<tr><th rowspan="2">perfect lubrica-tion</th><th colspan="2" rowspan="2">insufficient lubrication without with dirt (m)</th><th rowspan="2">no lubrica-tion</th></tr>
<tr><th>useful</th><th>acceptable</th></tr>
<tr><td>I</td><td>up to 4</td><td rowspan="4">1</td><td>0,6</td><td>0,3</td><td>0,15</td><td>Light drip lubrication, 4 to 14 drops in 1 min.</td><td>Fat lubrication. Hand lubrication.</td></tr>
<tr><td>II</td><td>up to 7</td><td>0,3</td><td>0,15</td><td>not accep-table</td><td>Dipped lubrication by wetting in oil bath.</td><td>Drip lubrication, about 20 drops in 1 min.</td></tr>
<tr><td rowspan="2">III</td><td>up to 12</td><td colspan="3" rowspan="2">not acceptable</td><td>Force - feed lubrication</td><td>Oil bath with spattering disk.</td></tr>
<tr><td>up to 12</td><td>Oil mist lubrication, force - feed lubrication with jet for small drip creation. Oil cooling if necessary, modificate!</td><td>Force - feed lubrication.</td></tr>
</table>

Table 3 Axis distance coefficients ρ

a = 20 p	a = 40 p	a = 80 p	a = 160 p
0,85	1,00	1,15	1,30

DIAGRAM 1

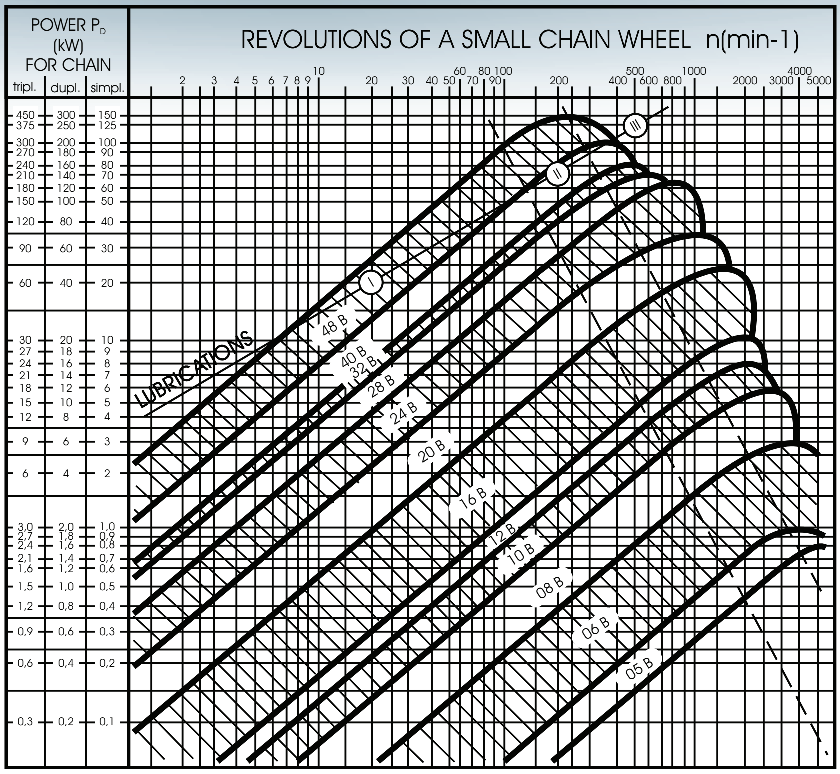
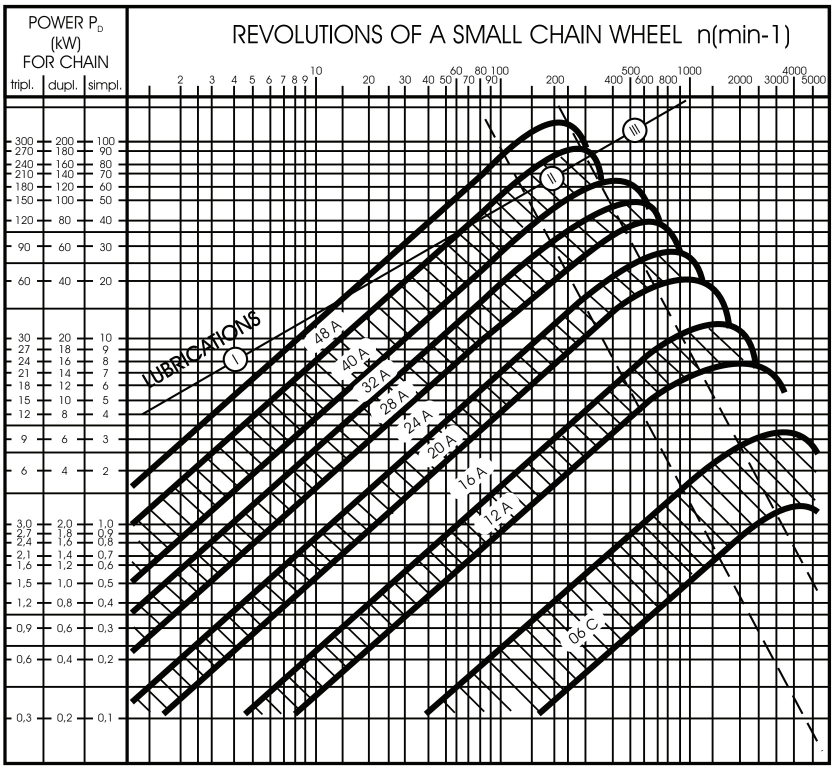


DIAGRAM 2



The values of the shock coefficient Y are selected according to the shocks, which are made by the run
machine in the transmission.

Y = 1 Shockless transmission
Y = 2 Light shocks, middle intermediate load
Y = 3 Middle shocks, abnormal intermediate load
Y = 4 Heavy shocks, middle transformed shocks

Examples of the number values Y, shows Table 4.

Calculation check of a specified chain

For inspection of a selected chain, we made the calculation of a real ratios in the chain transmission and then compare them with allowed values.
A new selection must be made, if the suggested chain does not correspond with these values.

Circumferential velocity of the chain $v = \dfrac{d.n_1}{19100}$ [ms^{-1}]

where d is the diameter of the driven wheel spacing circle

$$d = \frac{p}{\sin \dfrac{180°}{z_1}} \quad [mm]$$

n_1 = revolutions of the driving wheel [min^{-1}]

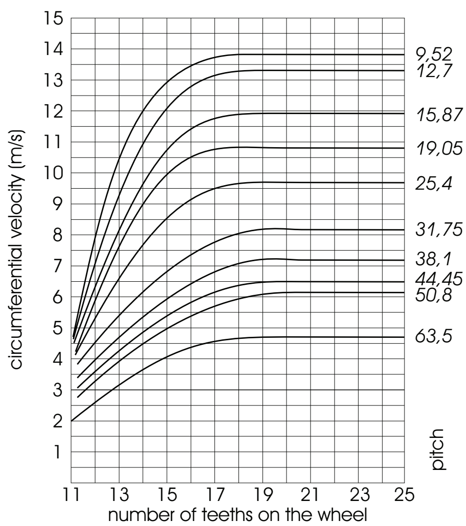
For comparison, allowed circumferential chain velocities are shown in Diagram 3.

Circumferential force from the transformed power on the chain wheel

$$F = \frac{P.\,1000}{v} \quad [N]$$

P = transformed power [kW]
v = circumferential velocity [ms^{-1}]

Diagram 3 Allowed circumferential chain velocities



Circumferential force caused by the centrifugal force
$F_{OC} = q \cdot v^2$
q = weight of 1 m chain [kg m^{-1}] (see catalogue tables)

Total traction force
$F_t = F_O + F_{OC}$ [N]

Pressure calculation in the chain joint

$$p_p = \frac{F_t}{S} \quad [MPa]$$

S = chain joint surface
$S = d_2 \cdot b_2$ [mm^2] d_2 = pin diameter [mm] b_2 = outer width of the inner chain link (bush length) [mm], - see catalogue tables.

Direction pressure in the chain joint [p_1] is shown in Table 5 and it is necessary for the allowed pressure determination.

Allowed pressure in the chain joint
$p_d = p_1 \cdot \lambda$ [MPa]
λ = friction coefficient (see Table 6)

Table 6 Friction coefficient λ

Warning:
$p_p < P_d$

Shock coefficient Y	Chains according to CSN	Friction coefficient λ																			
		$a = 20\ p$					$a = 40\ p$					$a = 80\ p$					$a = 160\ p$				
		i					i					i					i				
		1	2	3	5	7	1	2	3	5	7	1	2	3	5	7	1	2	3	5	7
1	02 3311, 02 3321	0,69	0,80	0,87	0,98	1,04	0,83	0.93	1,00	1,09	1,15	1,00	1,12	1,19	1,27	1,32	1,24	1,38	1,45	1,53	1,57
	02 3315	0,55	0,64	0,70	0,78	0,82	0,66	0,74	0,80	0,87	0,92	0,80	0,90	0,95	1,02	1,06	0,99	1,10	1,16	1,22	1,26
2	02 3311, 02 3321	0,50	0,58	0,64	0,72	0,76	0,60	0,68	0,73	0,79	0,84	0,73	0,82	0,87	0,93	0,97	0,91	1,01	1,06	1,12	1,15
	02 3315	0,40	0,46	0,51	0,58	0,61	0,48	0,55	0,58	0,63	0,67	0,58	0,66	0,70	0,75	0,78	0,73	0,81	0,85	0,90	0,92
3	02 3311, 02 3321	0,44	0,50	0,55	0,62	0,66	0,52	0,59	0,63	0,69	0,73	0,63	0,71	0,75	0,80	0,83	0,78	0,87	0,92	0,96	0,99
	02 3315	0,35	0,40	0,44	0,49	0,52	0,42	0,47	0,50	0,55	0,57	0,50	0,56	0,60	0,64	0,66	0,62	0,69	0,73	0,77	0,79
4	02 3311, 02 3321	0,40	0,46	0,51	0,57	0,61	0,48	0,54	0,58	0,63	0,67	0,58	0,65	0,69	0,74	0,77	0,72	0,80	0,84	0,89	0,91
	02 3315	0,32	0,37	0,40	0,45	0,48	0,38	0,43	0,45	0,50	0,53	0,46	0,52	0,55	0,59	0,61	0,58	0,64	0,67	0,71	0,73

Table 4 Number values Y

Shown values are the middle values at the axis distance $a = 40\ p$. At the unfavourable conditions, it is necessary to count with additions.		Driving machines										
		Electrical engine	Combustion engines					Water turbines		Steam turbines	Piston steam machines	Driving transmissions (group drive)
			Low speed		High speed							
			1 cylinder	2 cylinders	Up to 2 cylinders	4 cylinders	6 cylinders and more	Fast	Slow			
Lathes, drills		1,4										
Milling machines		1,5										
Planing machines		2,3										
Shaping machines		2										
Drawing machines		1,8										
Press machines	hydraulic	1,8			2,8	2,5	2,2					
	eccentric	2,5										
	lever	2										
Machines for wood grinding		1,8	4,5	4	3,7	3	2,5	2,5	3,5		3,5	1,8
Weaving machines		2										2
Sawing machines	revolving	1,5										
	drilled	2										
Spinning machines		1,5										1,5
Piston compressors	single stage	2,5		5	4,5	4	3,5					
	duplex	2		4,5	4	3,5	3					
Centrifugal compressors	single stage	1,6	4	3,2	3	2,5	2					
	duplex	1,3	3	2,7	2,5	2	1,6					
Superchargers		1,5		3	2,7	2,5	2					
Ventilators		2,5		3,7							3,5	2,5
Piston pumps	single cylinder	2	5	4	3,5	3	2,6	2,5	3,5			
	double cylinder	1,8	4	3,5	3	2,7	2,3	2,2	2,7			
Centrifugal pumps		1,5	3	2,8	2,5	2,2					2,5	
Rolling trains	transmissed	2,5										
	direct	3										
Crushing cylinders		2										2
Ball mills		1,8										1,8
Tube mills		2										2
Hammer mills		2,5		5	4,5	4	3,5					2,5
Calenders	transmissed	2,5										
	direct	3										
Cellulose grinders		1,8						3,2	3		3,5	1,8
Shaking screens		2		4	3,5	3,2	2,8				4	2
Peening rammers		2	5	4	3,5	3,2						
Rotating mixers		1,7	4	3,2	3	2,5	2					
Diggers		3			5	4,5	4				5	
Soil milling machines				5	4,5	4						
Mixers		1,6										1,6
Bulk material transporters		1,5	3	2,8	2,5	2,2	2				2,8	1,5
Piece material transporters		2	4	3,5	3	2,7	2			1		
Lifting machines		2,5	5	4	3,5	3	2,6			1,5		
Fork lift trucks		3			4,5	3,5				1,5		
Mining winches		2,5										
Generators	big equipment	1		2				1,2	1,5		1,8	1
	small equipment	1,5		2,8				1,7	2,5		2	1,5
Driven transmissions		1,5				2,3	2	2	2,5		2,5	1,5

Table 5 Direction pressure in the chain joint p_1

Chain speed v m s^{-1}	Pressure in the chain joint p_1 in [N cm^{-2}] at the small wheel number of teeth													
	11	12	13	14	15	16	17	18	19	20	21	22	23	24	≥ 25
0,1	3129	3129	3129	3139	3139	3149	3178	3198	3198	3208	3247	3247	3247	3247	3286
0,2	2796	2923	3002	3012	3021	3021	3041	3041	3041	3071	3100	3119	3149	3169	3189
0,4	2590	2708	2757	2825	2845	2865	2894	2914	2933	2943	2963	2972	2992	3021	3051
0,6	2413	2511	2609	2678	2708	2737	2776	2786	2815	2835	2845	2865	2904	2943	2972
0,8	2246	2384	2453	2531	2570	2619	2659	2678	2708	2727	2757	2776	2796	2835	2855
1,0	2129	2266	2335	2413	2472	2541	2560	2590	2639	2668	2678	2708	2746	2766	2796
1,5	1864	2001	2119	2207	2276	2335	2403	2433	2462	2492	2521	2551	2580	2600	2619
2,0	1668	1805	1933	2029	2109	2178	2217	2276	2325	2364	2394	2423	2453	2482	2511
2,5	1511	1658	1795	1893	1982	2050	2090	2148	2188	2227	2266	2305	2345	2347	2413
3,0	1364	1521	1648	1756	1854	1942	2001	2050	2090	2129	2168	2207	2237	2276	2305
4,0	1138	1305	1442	1560	1667	1746	1815	1873	1913	1962	2001	2040	2070	2109	2132
5,0	932	1109	1275	1393	1491	1589	1668	1736	1785	1834	1877	1903	1942	1972	2011
6,0		952	1108	1256	1364	1472	1550	1619	1658	1697	1746	1785	1725	1864	1893
7,0			961	1099	1236	1354	1432	1501	1560	1599	1648	1687	1727	1766	1805
8,0				981	1118	1226	1334	1403	1472	1521	1560	1609	1648	1687	1717
10,0					912	1050	1148	1236	1305	1364	1403	1442	1491	1530	1560
12,0						883	991	1099	1167	1236	1285	1334	1373	1403	1442
15,0							785	912	999	1059	1118	1167	1216	1256	1295
18,0								736	814	893	952	1010	1069	1118	1158
21,0									667	755	814	883	942	991	1030
24,0									500	588	667	730	804	863	912

Direction values according to Table 5 are valid for about 10.000 working hours at Y = 1, μ = 1, ϱ = 1, chain run over 2 wheels and the transmission ratio

$$i = \frac{n_1}{n_2} = \frac{z_2}{z_1} = 3.$$

Safety coefficient against breaking at statical load

$$k_s = \frac{F_B}{F_t} \geq 7$$

F_B = chain strenght in breaking [N] (see catalogue tables)

Safety coefficient against the breaking at dynamical load

$$Kd = \frac{F_B}{F_t Y} \geq 5$$

Y = shocks coefficient (see Table 4)

If the suggested chain does not perform any of the mentioned values, it's necesary to choose the chain with bigger pitch, or with higher strenght.

Chain number of links calculation from chosen axis distance

$$x = 2 \cdot \frac{a}{p} + \frac{z_1 + z_2}{2} + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \cdot \frac{p}{a}$$

According to the calculated chain number of links we choose the nearest even number of links. Exceptionally we choose the odd number of links, because it's necesary to use the reducing link, which decreases the chain strenght.
We recalculate the axis distance for the chosen number of links.

Axis distance calculation

$$a = \frac{p}{8}\left[2x - z_1 - z_2 + \sqrt{\left(2x - z_1 - z_2\right)^2 - F\left(z_2 - z_1\right)^2}\right]$$

we find the coefficient *F* in Table 7

Table 7 Coefficient *F*

$\frac{X - z_1}{z_2 - z_1}$	*F*	$\frac{X - z_1}{z_2 - z_1}$	*F*	$\frac{X - z_1}{z_2 - z_1}$	*F*	$\frac{X - z_1}{z_2 - z_1}$	*F*
12,00	0,8106	2,90	0,8116	1,37	0,8215	1,19	0,8310
11,00	0,8106	2,80	0,8118	1,36	0,8219	1,18	0,8318
10,00	0,8107	2,70	0,8119	1,35	0,8222	1,17	0,8326
9,00	0,8107	2,60	0,8121	1,34	0,8226	1,16	0,8336
8,00	0,8107	2,50	0,8123	1,33	0,8230	1,15	0,8346
7,00	0,8108	2,40	0,8125	1,32	0,8234	1,14	0,8358
6,00	0,8108	2,30	0,8127	1,31	0,8238	1,13	0,8372
5,00	0,8109	2,20	0,8130	1,30	0,8243	1,12	0,8387
4,80	0,8109	2,10	0,8134	1,29	0,8248	1,11	0,8405
4,60	0,8109	2,00	0,8138	1,28	0,8253	1,10	0,8425
4,40	0,8110	1,90	0,8143	1,27	0,8258	1,09	0,8448
4,20	0,8110	1,80	0,8150	1,26	0,8264	1,08	0,8474
4,00	0,8110	1,70	0,8158	1,25	0,8270	1,07	0,8503
3,80	0,8111	1,60	0,8170	1,24	0,8276	1,06	0,8537
3,60	0,8112	1,50	0,8185	1,23	0,8282	1,058	0,8544
3,40	0,8113	1,40	0,8207	1,22	0,8289	1,056	0,8551
3,20	0,8114	1,39	0,8209	1,21	0,8295	1,054	0,8559
3,00	0,8115	1,38	0,8212	1,20	0,8302	1,052	0,8567

Chain wheel dimension calculation for roller and bush chains

We substract the chain wheel dimensions according to the relations shown in Tables 8 and 9.

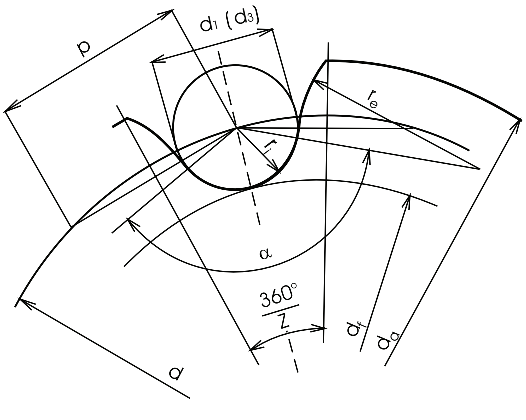


Table 8

PARAMETERS	Zn	FORMULA FOR CALCULATION
Pitch circle diameter	d	$d = \dfrac{p}{\sin \dfrac{180°}{z}}$
Root circle diameter	d_f	$d_f = d - 2r_i$
For the gap with smallest width 1. bottom tooth gap radius 2. tooth face radius 3. openning angle	 $r_{i\,min}$ $r_{e\,min}$ α_{max}	 $r_{i\,min} = 0{,}505 \cdot d_1\ (d_3)$ $r_{e\,min} = 0{,}12 \cdot d_1\ (d_3) \cdot (z+2)$ $\alpha_{max} = 140° - \dfrac{90°}{z}$
For the gap with biggest width 1.bottom tooth gap radius 2.tooth face radius 3.openning angle	 $r_{i\,max}$ $r_{e\,max}$ α_{min}	 $r_{i\,max} = 0{,}505 \cdot d_1\ (d_3) + 0{,}069 \cdot \sqrt[3]{d_1\ (d_3)}$ $r_{e\,max} = 0{,}008 \cdot d_1\ (d_3) \cdot (z^2+180)$ $\alpha_{min} = 120° - \dfrac{90°}{z}$
roller diameter	d_1	
bush diameter	d_3	
addendum circle diameter	d_a	$d_{a\,min} = d + 0{,}5 \cdot d_1(d_3)$ $d_{a\,max} = d + 1{,}25\,p - d_1(d_3)$

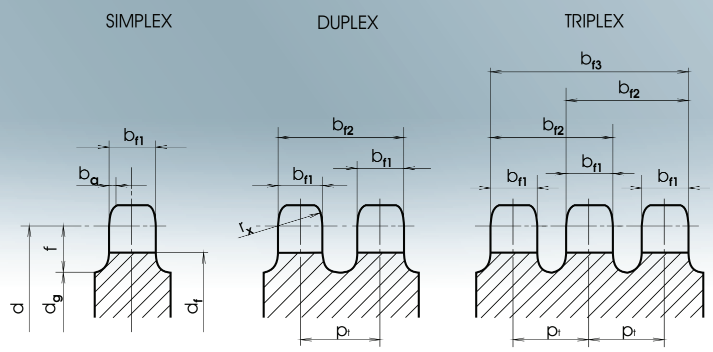
Dimensions d_3 are valid only for bush chains.

Basic dimension calculation of the rims cross cut at simple, duplex, triplex chain wheels, must be made according to Table 9.

Table 9

PARAMETER TITLE		Zn	FORMULA FOR CALCULATION	
biggest rim diameter		d_g	$d_g = d - 2f$	
pitch circle and rim radius difference		f	$f = 0{,}7p$ for standard chains $f = 0{,}4p$ for long pitch chains	
distance between the rows at multiple row chain		p_t	according to datas in catalogue tables	
tooth round radius		r_x	$r_x = 1{,}5\ d_1$	
tooth round value		b_a	$b_a = (0{,}1 \div 0{,}15)\ d_1$	
chain inner width		b_1	according to datas in catalogue tables	
chain wheel tooth width			chain pitch	
			$p \leqq 12{,}7$	$p > 12{,}7$
	simplex	b_{f1}	$0{,}93\ b_1$	$0{,}95\ b_1$
	duplex		$0{,}91\ b_1$	$0{,}93\ b_1$
	triplex		$0{,}88\ b_1$	$0{,}93\ b_1$
chain wheel rim width	duplex	b_{f2}	$0{,}91\ b_1 + p_t$	$0{,}93\ b_1 + p_t$
	triplex	b_{f3}	$0{,}91\ b_1 + p_t$	$0{,}93\ b_1 + p_t$

CHAIN WHEELS :



Tables 8 and 9 construction supplement :

1. Evolvent shape (radius r_e) of the tooth head is allowed. You must keep the standard values, which are in Tables 8 and 9.

2. Recommended maximum tooth surface roughness is Ra = 3,2 - 6,3
   Modification A in extent to Ra = 3,2.
   Modification B in extent to Ra = 6,3.

3. Allowed maximum radial run-out of the root circle is 0,0007 d_f + 0,076 mm, but the top is 0,76 mm. Maximum side run-out of the root circle can be 0,0009 d_f + 0,076 mm, but the top is 1,14 mm.
   These methods are valid for the general transmission usage, for which special requirements are not suggested. In special cases it is necessary to choose smaller deviations, respect to the chain transmission run exactness, for example car timing gears.

4. A special standard is valid for the mentioned calculation formulas and suggestions, which are not valid for multiple gear freewheels of bicycles.

5. Limited deviations of the chain wheel tooth width b_{f1} at single row, b_{f2} at duplex and h14 at triplex are chosen, dimension b_{f3} is informative.

Standard CSN 260491 indicates dimensions of the chain wheels for long pitch chains.
CZ Retezy, Ltd. resumes its long time experience in the field of chain transmission into a complex calculation and suggestion of the chain transmission with computer help. Hereby, we offer our customers service and help with the chain transmission solution.
We recommend our customers to contact Design office of CZ Retezy, Ltd., which can help with the suggestion or suggest the optimal solution for the required transmission. Please, remember that a well suggested transmission insures long durability, no failure and low maintenance costs of the chain transmissions.

Chain wheel materials

Materials used for the chain wheels vary according to the transmission type and to the number of teeth of the chain wheel.
For the pinions, it is better to use cemented steels for example 12020, 14220, because the chain wheel heat treated teeths are very hard and resist against the wear. These materials are mostly used, because at small number of teeth of the chain wheel, the tooth touches the chain frequently, and is worn-out. Steel without treatment, for example 11600, is possible to use for wheels with bigger number of teeth, because the tooth is not so frequently in touch with the chain and transformed force is resolved to more teeth. Big chain wheels (for chains with pitch 19,05 mm and more) are produced from cast iron with hardness about HB = 220, or from steels for castings. Chain wheels produced from plastic, zinc alloy etc. transforming low forces can be used for different transmissions (type writers, printing machines, toys etc.).

<u>Rules for a well-functioning chain transmission</u>

If you follow the below mentioned rules, you will be satisfied with the long durability of the chain transmission.
Any deviation from these rules could cause a sudden damage of the chain and its lower durability.

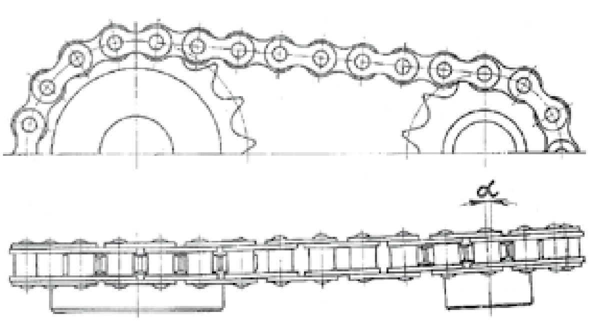
Rules:

- To avoid the transmission oscillation, dimension the shafts and bearings sufficiently.

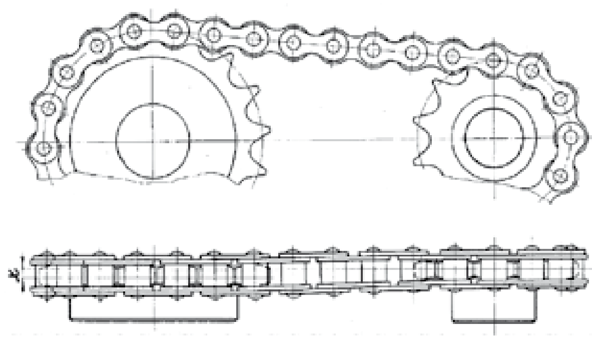
- Chain wheels must be mounted in the line and shaft axis must be parallel. Mounting mistakes are shown in Layouts 2 and 3. At a wrong mounted transmission, the chain is stressed not only by the traction force, but also by the bending force, which decreases the chain durability. It can cause the break of the chain. The chain plates with the side force effect, touch the chain wheel sides and cause excessive wear.

- It is suggested to locate the chain wheel as near as possible to the bearings, in order to decrease the pressure in them and also to decrease the chain wheels oscillation because of production incorrectness.

- At a small chain wheel keep the minimum number of teeth $z_1 = 17$, because the big angle at wheels with a low number of teeth over which the chain joint must turn have a big influence on excessive wear.



Layout 2 Chain wheels axis parallelism defect



Layout 3 Chain wheels offset

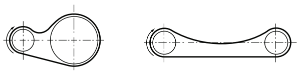
- At a big chain wheel it is recommended not to exceed the number of teeth $z_2 = 120$ at standard transmissions.

- Examples in Layouts 4, 5, 6 show how it is possible to arrange the chain transmission. For improvement of the chain kinematics, it is better to locate the tensile branch up. If a stretching roller is used in the chain transmission, it is necessary to use such a stretching roller that has odd number of teeth, or a stretching roller which is smooth. It is better to choose a firm stretching roller with adjusting, then with a spring, because that causes additional force into the chain.

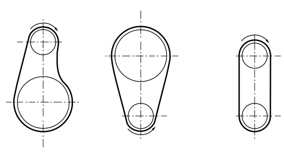
- Use a small chain wheel with odd number of teeth (if the construction allows) for reaching a chain steady wear.

- 100 multiple of the chain pitch is the maximum allowed axis distance for standard transmissions.

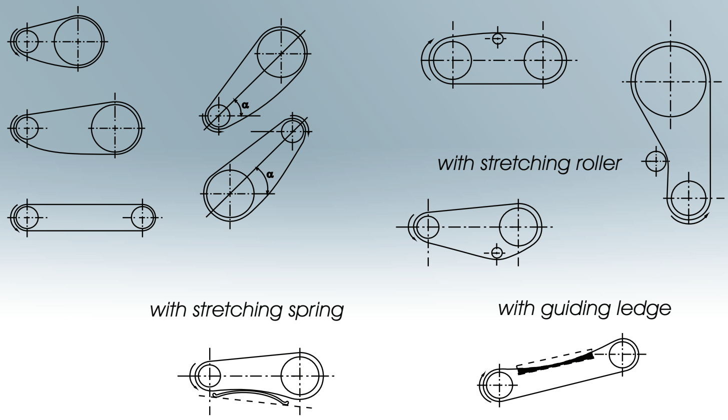
- For levelling the initial prolongation of the chain and for stress decreasing of the worn chain, it is necessary to constructionaly ensure the axis adjusting of one of the shafts. If it is not possible to follow these conditions, it is necessary to inbuild a stretching roller into the transmission.



Layout 4 Incorrectly solved transmissions



Layout 5 Preferably solved transmissions

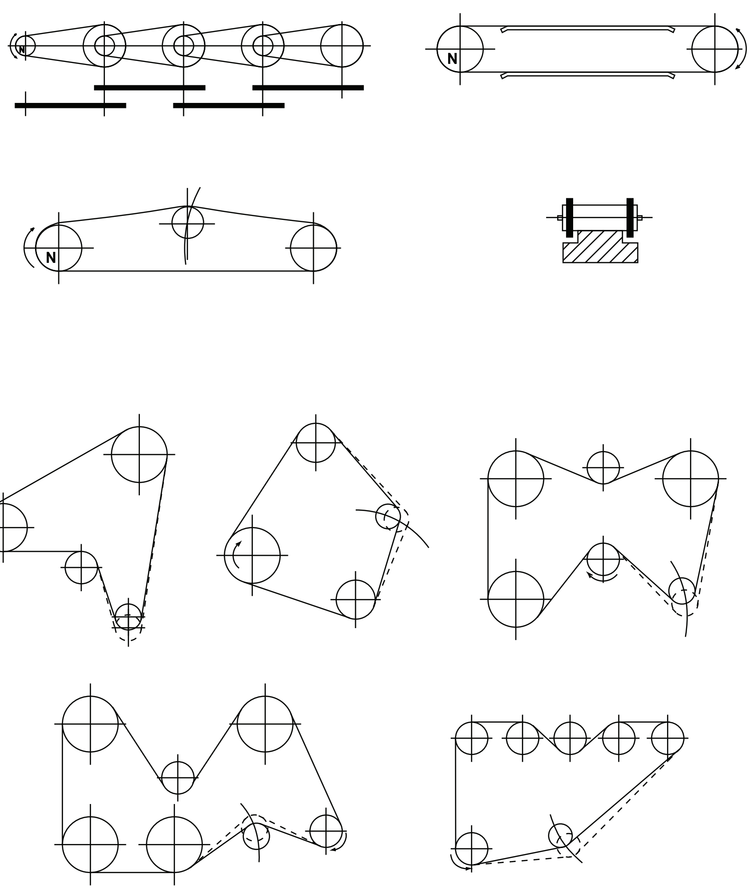


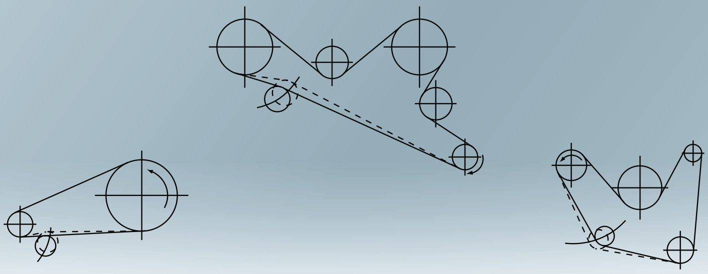
Layout 6 Correctly solved transmissions

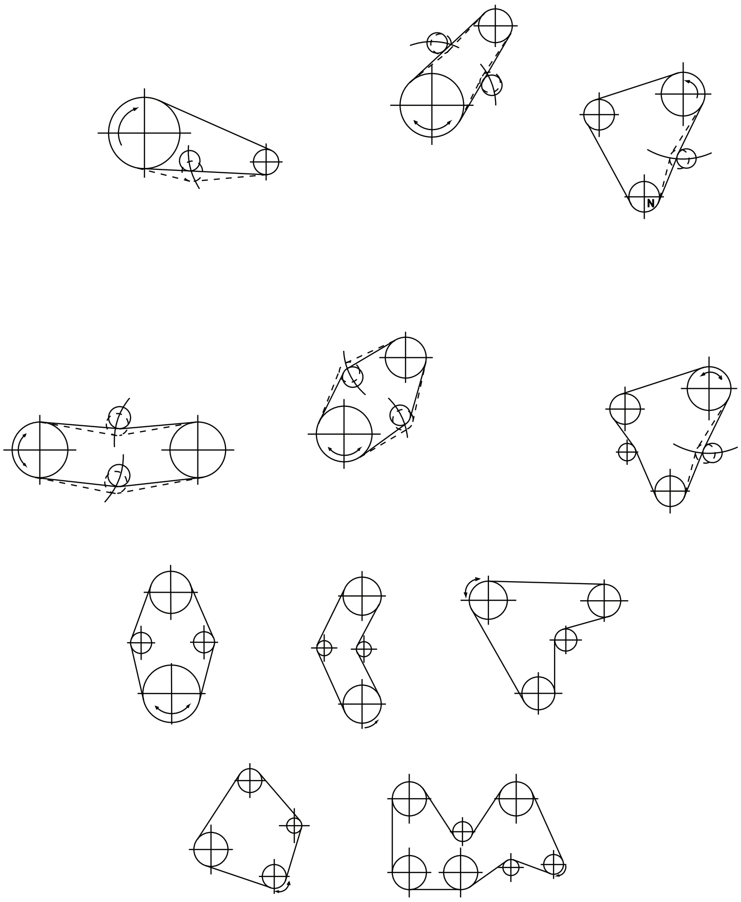
Chain transmission solutions

Chain transmissions, as mentioned before, have many advantages against other transmissions (belt, gearwheels and others), that is why it also allows to solve complicated transmission systems with a lot of chain wheels. But like at every transmission it is necessary to follow some specific instructions for a good function and durability. The instruction, how to preferably solve the chain transmissions including the stretching elements and guiding lathes shows Layout 1.

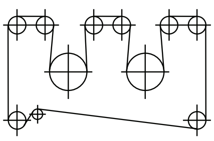
Layout 1 Chain transmission arrangement examples



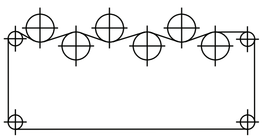




Drive of multiple shafts with stretching wheels



Drive of a roller track



Drive of a light rolling mill

Quality with Tradition

CHAIN DURABILITY

Because the chain is a mechanical part assembled from many parts, it has its own technical durability.
Chain durability is determined by the allowed operation elongation. The value of elongation is given by the standard and it is determined so that the chain transmission could ensure the quality force transfer, sufficient for safety operating.
Chain elongation ΔL is expressed by the length difference of the worn out chain L and basic chain lenght L_z

$\Delta L = L - L_z$

Chain basic length Lz is calculated : $Lz = x \cdot p$
x = number of links
p = chain pitch

Value of the allowed elongation ΔL_{max} is not the same at all chains :

a) roller and bush chains for general use ΔL_{max} = 2 % Lz according to CSN, ΔL_{max} = 3% Lz according to DIN

b) high speed chains are mostly used in car industry (timing, balancing…) ΔL_{max}= 1% Lz is recommended

c) leaf chains (measured in the part, which is in contact with the return pulley) ΔL_{max} =3% Lz

d) sport chains (motorcycles, bicycles) have their allowed specifical elongation according to customer´s usage.

Allowed chain elongation mentioned in per cent, relate to the over-all chain length.

The allowed maximum production tolerance of new chains from the basic dimension:
+ 0,15 % from over-all chain length - roller chain (according to CSN and DIN)
+ 0,10 % from over-all chain length - bush, high speed (only according to CSN).

How to measure the chain elongation

1. Dissmount the chain from the transmission and clean it. It is important to avoid the dirt and rest of the lubricant between the pin and the bush, these can distort the measuring.
   Put the cleaned chain on a flat plate, stretch it, in order to take up the clearances between the parts, then measure the length with a measuring scale (outer holes pitch after connecting link). It is easier to measure 50 or 100 pitches. Subtract the basic length Lz from the measured length. The resulting value $\Delta L = L - Lz$ the chain elongation.

For quick determination of the extension use the following table ΔL_{max} = approximately 2 %.

Chain pitch		ΔL_{max} on 50 links	ΔL_{max} on 100 links
in inch	in mm	[mm]	[mm]
	8,0	8,0	16,0
3/8"	9,525	9,5	19,0
1/2"	12,7	12,7	25,4
5/8"	15,875	16,0	32,0
3/4"	19,05	19,0	38,0
1"	25,4	25,5	51,0

2. Less exact method for the elongation evaluating is measuring directly on the transmission.
   You must measure the chain the stretched section. The procedure of the calculation is the same like in point 1. For better exactness we measure the length L on arbitrary number of links (as many as possible).