Selection of a driving roller chain

The following rules must be observed during the design of the chain transmission and the selection of the chain:

The chain, shaft and bearings must be properly dimensioned. The sprocket axes must be exactly parallel and chattering of the chain must be prevented.

An effective lubricating system must be designed, preferably the one that does not require maintenance by operators.
(oil dip, oil spray)

Sprockets must be fixed as close as possible to the shaft bearings. This lowers the pressure exerted to bearings and also lowers the chatter of chain sprockets resulting from inaccuracy in fabrication.

The small sprockets should have at least 17 teeth. Sprockets with less teeth bend the chain more and thus diminish its endurance and increase the operating noise.

The big sprockets should not have more than 120 teeth.

If the chain transmission utilises a tightening sprocket, this sprocket must have an odd number of teeth. Sprockets with 13 teeth were found to perform best.

The dimensions of the sprockets must be exact, this pertains especially to the shape of teeth - minimum tolerances in fabrication.

The driving section of the chain should be preferably the top one to improve the kinematics of the chain transmission.

The pitch should be selected based on the chainvelocity. Double- or multiple-strand chain of low pitch is suitable for high-velocity transmissions.

To acquire uniform wear of the chain, whenever the design permits, use an odd number of teeth on the small sprockets.

1. LOAD TYPE

During the selection of a chain, the character of the load and the daily operating time must be assessed. Many devices expose the chain to shocks. Examples: Diesel engines, rolling mills, piston pumps, excavators, hoists. In these cases, the force calculated from the transmitted power must be scaled up by the shock coefficient (1.1 to 1.4) bases on the type of operation and you should use the scaled up force in all subsequent calculations. Continuous operation also leads to higher load and thus the permitted load must be lowered. Again, this effect is included in calculation by a coefficient (ca. 1.2) that is used to multiply the transmitted power in all calculations. Both coefficients mentioned above are summed up and shown in tables in the paragraphs describing the calculation.

2. TRANSMITTING RATIO

The transmitting ratio should not exceed 8 to 9 (low-pitch chains) or 6 to 7 (high-pitch chains). In special cases, higher ratio chain transmissions can be designed. These transmissions, however, exhibit short operating life because of frequent contact of the chain with the small sprockets . Chain transmissions of ratio i = ca. 3, where the number of teeth of the driving sprockets z_1 is selected so that the number of teeth of the driven sprocket z_2 is 60 to 70, possess the best characteristics. If the big sprocket has more than 60 to 70 teeth, the operating life of the chain is lowered, because even the new (a little elongated) chain cannot engage well with such a big number of teeth (the number of engaged teeth is proportionate to the angle of engagement) - this leads to rough operation of chain transmission. Such chain must be changed more often; it can, however, be used on chain transmissions with a lower number of teeth. Speed-increasing chain transmissions should be avoided. If used, the transmitting ratio must be low and the small (in this case: driven) sprockets must have at least 25 teeth (preferably more, especially with high-speed transmissions).

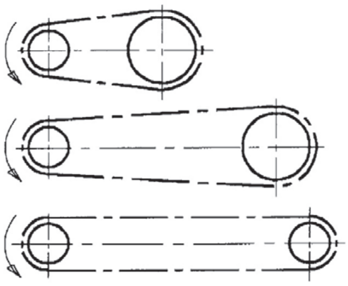
3. AXIAL DISTANCE, CHAIN LENGTH AND TRANSMISSION DESIGN

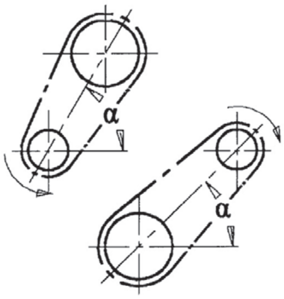
For a given chainpitch and number of teeth of both sprockets, the axial distance must be selected so that the chain length equals to a multiply of the pitch. To avoid the need for a coupling link, the chain should consist of an even number of links. So it is best to define the number of chainlinks, to round that number and then to calculate the axial distance.

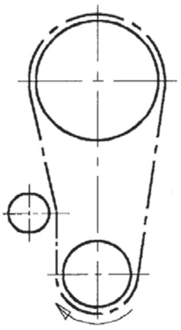
Within normal operating conditions, the axial distance should equal to 40 times the chainpitch. In special cases, the axial distance could be lowered, even to a degree where the sprockets almost touch each other. This lowers the operating life of the chain, as the chain engages the sprockets more often. On the other hand, axial distance can be greater, but in that case the excess weight of the chain stresses the chain sprockets bearings. The additional sprockets should be used to support the extended sections of chain longer than 1.5 meter. Short chains, a small number of teeth and high chainvelocity lead to warming of the chain. Such chain must be lubricated by appropriate lube to prevent its degradation. Maximum permissible axial distance is 100 times the chainpitch. In such cases, the axes must be adjustable to compensate for the stretch of the chain, as well as for the stretch of worn chain. If the axes are fixed, a tightening sprockets must be employed in the idle section. Its range must be at least 2 times the chainpitch. The involvement of tightening sprockets also allows to avoid the Y-shaped coupling link, which would otherwise lower the permissible chain load by 20%. The shafts must be dimensioned to prevent chatter.

Heavy-duty chain transmissions cannot employ "fly" attachment of sprockets (the axis must be fixed on both sides of the sprockets). The axes must be parallel and must not cause centrifugal force. The best design of chain transmission is characterised by 60 degrees angle between the tie-line of sprockets axes and the horizontal level.

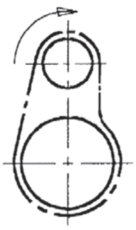
Well-designed transmissions

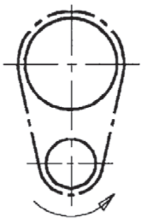


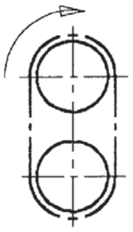




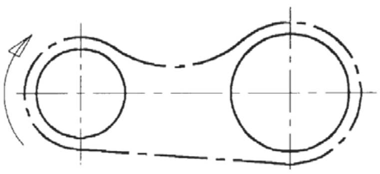
Less-desirable design of transmissions

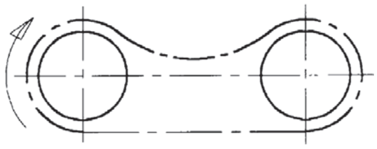






Wrong design of transmissions



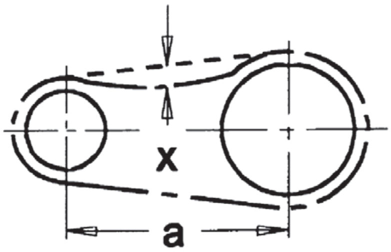


4. CORRECT ASSEMBLY OF CHAIN

Roller chains are usually being connected by connecting links. Whenever possible, use chains with an odd number of links so that both ends of the chain consist of an inner link. Thus the chain can be connected with a straight connecting link. Its spring must be assembled in the direction of the chain movement (see Fig. 2). The chain with an even number of links uses an offset link. The upper section of chain must always be the driving one, whereas the lower section must be the idle one. The chain must not be tightened too much, some sag is desired. Chains that are tightened too much chatter (can even fall off the sprockets) and warm during operation. The right sag is shown in Fig.1. Never mount a new chain onto worn chain sprockets! When mounting a new chain onto new sprockets, check the shape of teeth to find out whether the chainlinks slide off the teeth well. The run-in (several days) is recommended after the new chain is mounted. The length of the chain settles after the run-in. Maximum permissible stretch of the chain is 2% (1% for high-speed chains).

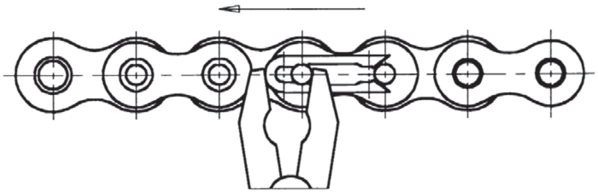
Right chain sag x (mm)

Fig. 1



Axes Distance a (mm)	**100**	**250**	**500**	**750**	**1 000**	**1 250**	**1 500**	**1 750**	**2 000**	**2 250**	**2 500**
Sag x	3	11	23	36	49	61	74	87	105	112	125

Fig. 2
Connecting link spring mounted in direction of chain movement



New chain
A new chain must be checked for its length (quality of fabrication).
The following rules apply to the measurement:

- measure 50 links of dry (preservation removed) chain on a level surface
- strain the chain with specified tension. The value is calculated by multiplying the square of the pitch by a coefficient of 0.8 for single-strand chains, 1.5 for double-strand chains and 2.2 for triple-strand chains.

The pitch values are put in millimetres and the result is in Newton.

The permissible deviation of nominal length is +0.15% of table value of pitch.

Chain in operation
The stretch of a chain in operation is measured and compared to a defined limit (+2%, to prevent excess wear of sprockets) in the following way:

- lay the chain onto a level surface and measure its length with a steel ruler
- subtract the appropriate multiple of chainpitches
- the difference must not exceed 2% of the total length of an equal number of links of a new chain.

The measurement is easier if you measure 50 or 100 links. This way, the stretch can equal up to one pitch (for 50 measured links), of two pitches (for 100 measured links).

5. MAINTENANCE OF ROLLER CHAINS

To reach long operating life, the chain must be regularly and appropriately maintained. This applies especially for a chain transmissions exposed to factors of surrounding environment, e.g. transmissions which are not sealed within a protective casing against water, dust etc. The operating life is positively influenced by perfect lubrication and regular maintenance based on the number of operating hours of the device and the chain transmission itself. The lubrication can be accomplished in several ways, based on the design of the transmission and the following rules apply:

a) chainvelocity up to 3 m/s - occasional manual lubrication by oilcan, brush or drip oiler (4 to 12 drips per minute)
b) chainvelocity up to 7 m/s - drip oiler (20 drips per minute) or oil dip
c) chainvelocity up to 12 m/s - lubrication under pressure, oil jets focused onto chain section.
d) chainvelocity over 12 m/s - lubrication by oil spray

The best way of lubrication is oil dip, which is the most reliable. This method requires the following conditions to be met:

- the chain should not be completely dipped, as this leads to warming and drag
- a tenuous oil should be used (viscosity 4 to 5 deg. E) to cover all chain components and sprockets with an oil film. This prevents wear of both the sprockets and the chain itself.

To prevent oil splashing and/or intrusion of objects into the transmission, every chain transmission should be fitted with a suitable casing.

The wear occurs mainly on the sliding surfaces of the chain link, e.g. pins, bushings, rollers and on the surfaces of sprockets. Individual components of the chainlinks are produced of high-quality materials. The components are heat-treated to create surface layer resistant to wear. The surfaces of the chain and the sprockets are to be kept clean and lubricated. As the working surfaces of the chain are hidden within the links, they can be cleaned only by washing the chain in suitable solution.

We recommend the following way of chain maintenance (except for oil-dipped chains):

- wash the chain in kerosene, benzine or trichloroethylene and rub the deposits off the sides of the chainlinks with a scrubber
- soak the cleaned chain for 24 hours in kerosene to dissolve the deposits within the bushings and rollers
- wash the chain in benzine or trichloroethylene while moving the chain around to flush the deposits out of bushings and rollers
- remove the chain from the wash and inspect it visually. Should you hear noise when you move the chain (result of friction between the deposits and inner surfaces of bushings and rollers), continue with the soaking until the chain is absolutely clean
- after the cleaning is complete, inspect the chain once again and replace any damaged parts
- prepare a lubricating bath of dissolved solid lubricant. The temperature should be 80 deg. C. The dissolved lubricant sticks to inner surfaces of the chain. This way, the lubricant does not splash after the chain warms during operation.
- the chain must be moved within the lubricating bath to expel air and allow the lubricant to penetrate the links. The lubrication is completed when air bubbles stop to emerge from the bath.
- remove the chain from the bath and let it cool down
- mount the chain to cleaned sprockets. Use the same chain length as before.
- never mount a new chain onto worn sprockets as this leads to early wear of the chain!

6. SELECTION OF A CHAIN ACCORDING TO DESIRED OPERATING LIFE

The maximum load of chain transmissions is determined to prevent excess wear of chainlinks and related stretch of the chain. The stretch (the following value subject to common wear and the number of teeth) should not exceed 2% of nominal chain length (for common use), based on the assumption that the chain can be tightened as needed.

The following formulas for selection of a chain (according to desired operating life) are valid when no dynamic strain (chatter of driving or idle section, chatter of whole transmission) affects the chain.

6.1. Determination of the transmitted power

a) The chain selection must be based upon the operating conditions. The starting point for calculation is transmitted the power “N” and the chainvelocity “v”.

Thus: $$P = \frac{N \, . \, 1000}{v} \quad (N)$$

N = transmitted power in kW
v = chain velocity in m/s

b) When the traction force “P” and the chainvelocity “v” are known, the transmitted power “N” can be calculated as follows:

$$N = \frac{P \, . \, v}{1000} \quad (kW)$$

P = traction force in N

6.2. Determination of the shock coefficient "Y"

Use table A_1 or A_2 to determine the shock coefficient Y for the selected kind of transmission.

TABLE A_I

Shock coefficient	*Kind of transmission*
1	No shocks
2	Light shocks, medium occasional load
3	Medium shocks, extreme occasional load
4	Heavy shocks, medium steady load

TABLE A_2 - Shock coefficients for various types of transmissions

Driven machinery	*Driving machinery – Shock coefficient “Y”*										
	Electromotor	*Combustion engines*					*Turbines*			Steam engines	Transmis-sions
		Low speed		High speed			Water		Steam		
		1 cyl.	2 cyl.	2 cyl.	4 cyl.	6 cyl.	High	Low			
Lathes, drills	1,4									3,5	
Millers	1,5										
Planers	2,3										
Shapers	2,0										
Machinery	1,8										
Hydraulic presses	1,8			2,8	2,5	2,2					
Tappet presses	2,5										
Cranked lever presses	2,0										
Wood-fabrication machinery	1,8	4,5	4,0	3,7	3,0	2,5	2,5	3,5			1,8
Looms	2,0										2,0
Knitting machines, revolving	1,5									3,5	
Knitting machines, reverse	2,0										
Piston compressors, 1 stage	2,5		5,0	4,5	4,0	3,5					1,5
Piston compressors, 2 stage	2,0		4,5	4,0	3,5	3,0					
Radial compressor, 1 stage	1,6	4,0	3,2	3,0	2,5	2,0					
Radial compressor, 1 stage	1,3	3,0	2,7	2,5	2,0	1,6					
Superchargers	1,5		3,0	2,7	2,5	2,0					
Fans	2,5		3,7								2,5

Driven machinery	*Driving machinery – Shock coefficient "Y"*										
	Electromotor	*Combustion engines*					*Turbines*			Steam engines	Line shaftings
		Low speed		High speed			Water		Steam		
		1 cyl.	2 cyl.	2 cyl.	4 cyl.	6 cyl.	High	Low			
Piston pumps, 1 cylinder	2,0	5,0	4,0	3,5	3,0	2,6	2,5	3,5			2,5
Piston pumps, 2 cylinder	1,8	4,0	3,5	3,0	2,7	2,3	2,2	2,7			
Rolling mills w/ trans.	2,5										
Rolling mills w/o trans.	3,0										
Mill rolls	2,0										2,0
Ball grinders	1,8										1,8
Tubular grinders	2,0										2,0
Hammer grinders	2,5		5,0	4,5	4,0	3,5					2,5
Driving engines w/ trans.	2,5										
Driving engines w/o trans.	3,0										
Pulp grinders	1,8						2,2	3,0		3,5	1,8
Bolters	2,0		4,0	3,5	3,2	2,8				4,0	2,0
Tampers	2,0	5,0	4,0	3,5	3,2						
Mixing drums	1,7	4,0	3,2	3,0	2,5	2,0					
Excavators	3,0			5,0	4,5	4,0				5,0	
Rotavators			5,0	4,5	4,0					5,0	
Mixers	1,6										1,6
Conveyors (loose matters)	1,5	3,0	2,8	2,5	2,2	2,0				2,8	1,5
Conveyors (pieces)	2,0	4,0	3,5	3,0	2,7	2,0					
Hoists	2,5	5,0	4,0	3,5	3,0	2,6					
Forklifts	3,0			4,5	3,5						
Tackles	2,5										
Generators – light duty	1,5		2,0				1,2	1,5	1,0	1,8	1,0
Generators – heavy duty	1,0		2,8				1,7	2,5	1,5	2,0	1,5
Driven transmissions	1,5				2,3	2,0	2,0	2,5	1,5	2,5	1,5

6.3. Determination of transmission ratio "i"

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

i = transmission ratio

z_1 = number of teeth of the driving sprockets

z_2 = number of teeth of the driven sprockets

n_1 = revs of the driving sprockets

n_2 = revs of the driven sprockets

Transmission rations

Number of teeth of the driving sprockets z_1	*Number of teeth of the driven sprockets z_2*																		
	11	**13**	**15**	**17**	**19**	**21**	**23**	**26**	**28**	**31**	**35**	**39**	**43**	**48**	**53**	**59**	**66**	**73**	**81**
7	1,57	1,86	2,14	2,43	2,71	3,00	3,29	3,71	4,00	4,43	5,00	5,57	6,14	6,86	7,57	8,43	9,43	10,43	11,57
9	1,22	1,44	1,67	1,89	2,11	2,33	2,56	2,89	3,11	3,44	3,89	4,33	4,78	5,33	5,89	6,56	7,33	8,11	9,00
11	1,00	1,18	1,36	1,55	1,73	1,91	2,09	2,36	2,55	2,82	3,18	3,55	3,91	4,36	4,82	5,36	6,00	6,64	7,36
12	0,92	1,08	1,25	1,42	1,58	1,75	1,92	2,17	2,34	2,58	2,92	3,25	3,58	4,00	4,42	4,92	5,50	6,09	6,75
13	0,85	1,00	1,15	1,31	1,46	1,62	1,77	2,00	2,16	2,39	2,70	3,00	3,31	3,69	4,08	4,54	5,08	5,62	6,24
14	0,79	0,93	1,07	1,22	1,36	1,50	1,64	1,86	2,00	2,22	2,50	2,79	3,07	3,43	3,79	4,22	4,72	5,22	5,79
15	0,73	0,87	1,00	1,13	1,27	1,40	1,53	1,73	1,87	2,07	2,33	2,60	2,87	3,20	3,53	3,93	4,40	4,86	5,40
17	0,65	0,77	0,88	1,00	1,12	1,24	1,35	1,53	1,65	1,82	2,06	2,30	2,53	2,82	3,12	3,47	3,88	4,29	4,77
19	0,58	0,69	0,79	0,89	1,00	1,11	1,21	1,37	1,48	1,63	1,84	2,05	2,26	2,53	2,79	3,11	3,47	3,84	4,26
21	0,52	0,62	0,71	0,81	0,90	1,00	1,10	1,24	1,33	1,48	1,67	1,86	2,05	2,29	2,52	2,81	3,14	3,38	3,86
23	0,48	0,57	0,65	0,74	0,83	0,91	1,00	1,13	1,22	1,35	1,52	1,70	1,87	2,09	2,31	2,57	2,87	3,17	3,52
25	0,44	0,52	0,60	0,68	0,76	0,84	0,92	1,04	1,12	1,24	1,40	1,56	1,72	1,92	2,12	2,36	2,64	2,92	3,24
26	0,42	0,50	0,58	0,65	0,73	0,81	0,88	1,00	1,08	1,19	1,35	1,50	1,65	1,85	2,04	2,27	2,54	2,81	3,12
28	0,40	0,47	0,54	0,61	0,68	0,75	0,82	0,93	1,00	1,11	1,25	1,39	1,54	1,71	1,89	2,11	2,36	2,61	2,89
31	0,36	0,42	0,48	0,55	0,61	0,68	0,74	0,84	0,90	1,00	1,13	1,26	1,39	1,55	1,71	1,90	2,13	2,36	2,61
35	0,32	0,37	0,43	0,49	0,54	0,60	0,66	0,74	0,80	0,89	1,00	1,11	1,23	1,37	1,51	1,69	1,89	2,09	2,31
39	0,28	0,33	0,38	0,44	0,49	0,54	0,59	0,67	0,72	0,80	0,90	1,00	1,10	1,23	1,36	1,51	1,69	1,87	2,08
40	0,28	0,33	0,38	0,43	0,48	0,53	0,58	0,65	0,70	0,78	0,88	0,98	1,08	1,20	1,33	1,48	1,65	1,83	2,03

6.4. Determination of the power coefficient “k”

Table B lists the values of power coefficient “k” based on the already known shock coefficient “Y”, the number of teeth of the sprocket “z_1”, and the transmission ratio “i”.

TABLE B – power coefficient “k”

Transmission ratio i	Shock coef. Y = 1 Number of teeth z_1 of the sprocket - see A				Shock coef. Y = 2 Number of teeth z_1 of the sprocket - see A.				Shock coef. Y = 3 Number of teeth z_1 of the sprocket - see A.				Shock coef. Y = 4 Number of teeth z_1 of the sprocket - see A.			
$z_2 : z_1$	**13**	**17**	**21**	**>=25**	**13**	**17**	**21**	**>=25**	**13**	**17**	**21**	**>=25**	**13**	**17**	**21**	**>=25**
1 : 1	(0,39)	0,73	0,92	1,11	(0,28)	0,54	0,67	0,81	(0,24)	0,42	0,58	0,70	(0,22)	(0,34)	0,53	0,64
2 : 1	0,50	0,83	1,05	1,26	(0,36)	0,60	0,76	0,92	(0,27)	0,52	0,66	0,80	(0,25)	0,43	0,61	0,73
3 : 1	0,59	0,88	1,12	1,36	0,43	0,65	0,82	0,99	(0,33)	0,56	0,71	0,86	(0,27)	0,51	0,65	0,79
5 : 1	0,64	0,96	1,22	1,49	0,47	0,70	0,89	1,09	0,40	0,60	0,77	0,94	(0,33)	0,57	0,71	0,86
7 : 1	0,67	1,02	1,30	1,59	0,49	0,75	0,95	1,16	0,42	0,64	0,82	1,00	(0,35)	0,59	0,75	0,92
Values given for distance of axes a = 40 x p; a = 80 x p increases “N” to 115%, a = 20 x p decreases “N” to 85% of original value (N = transmitted power)																

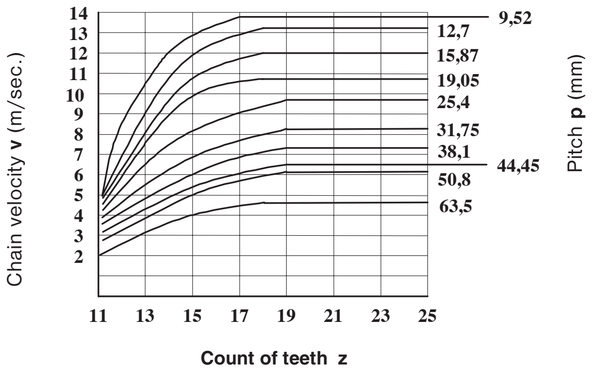
6.5. Determination of the chainvelocity ”v”

$$v = \frac{z_1 \,.\, p \,.\, n_1}{60000} = \frac{d_t \,.\, \pi \,.\, n_1}{60000} \quad (m/s)$$

$$d_t = \frac{p}{\sin \dfrac{180°}{z_1}} \quad (mm)$$

v = chain speed (m/sec.)
z_1 = number of teeth of the driving sprocket
n_1 = revs of the driving sprocket
d_t = pitch circle of the driving sprocket (mm)
p = pitch (mm)

Table C - Permissible chain velocities



6.6. Determination of the lubrication coefficient “l_2”

Table D – lubrication coefficient l_2

		Lubrication		Lubrication coefficient l_2			
				Lubrication			No lubrication
Power range see Table C	Chain Velocity (m/sec.)	Suitable	Permissible	Suitable Permissible	Sufficient		
					Without	With	
					Contamination		
I	up to 4	Oil drips, 4 to 14 per minute	Grease Manual lubrication	1	0,6	0,3	0,15
II	up to 7	Oil dip	Oil drips ca. 20 per minute	1	0,3	0,15	Impermissible
III	up to 12	Oil pressure	Oil dip W/ splash ring	1	Impermissible		
	Over 12	Oil spray	Oil pressure	1			

Oils posses good adhesion within the range of operating temperatures 20 to 40 degrees C. The chain should not be completely dipped.

6.7. Determination of the chain type coefficient „φ“

φ = 1,5 - for chains according to DIN8181, ISO 1275
φ = 1,0 - for chains according to DIN8187, ISO R 606
φ = 0,8 - for other chains

6.8. Determination of the axes distance coefficient „σ“

The following table shows the values of axes distance coefficient “σ” based on axes distance “a”.

a	*20 x p*	*40 x p*	*80 x p*	*160 x p*
σ	0,85	1,00	1,15	1,30

6.9. Determination of the diagrammatic power “N_d” - selection of suitable chain type

$$Nd = \frac{N}{k \, . \, l_2 \, . \, \varphi \, . \, \sigma} \quad (kW)$$

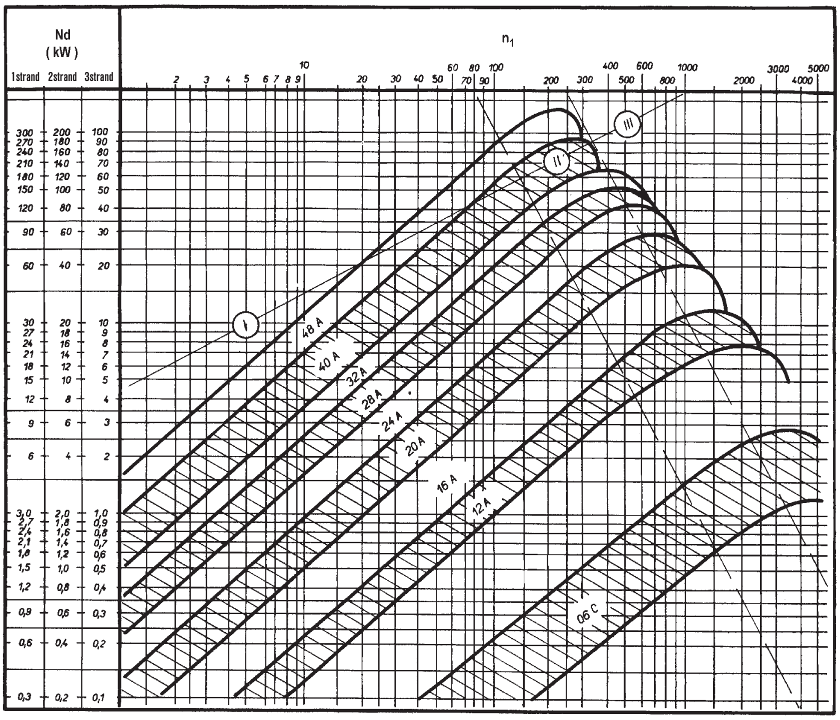
Nd = *diagrammatic power (kW)*
N = *transmitted power (kW)*
k = *power coefficient*
l_2 = *lubrication coefficient*
φ = *chain type coefficient*
σ = *axes distance coefficient*

Use chart E to determine chainpitch p and lubrication method (power ranges introduced in table D are marked in chart E as traverse bands I, II and III) from given revs and N_d.

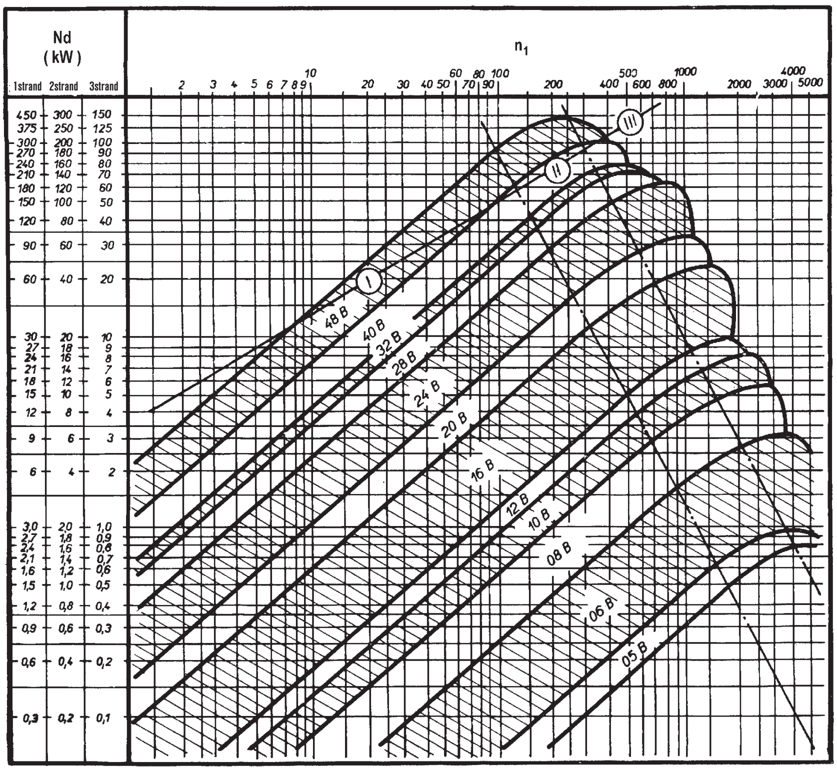
Table E - power and rev chart

Power "N_d" (kW), revs of sprocket n_1 (rpm)

Power-rev chart for roller chains DIN 8188 (American)



Power-rev chart for roller chains DIN 8187 (European)



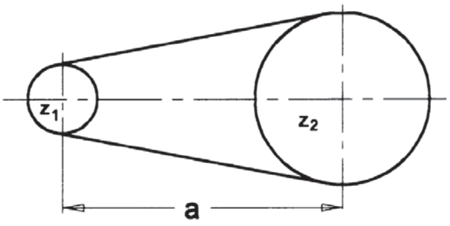
6.10. Determination of the number of chainlinks and the chain length "X"

Given the chainpitch and the number of teeth of the sprockets, the axes distance must be selected so that total chain length is an integer multiple of pitch. To avoid the need of an offset link, the chain should consist of even number of links. Thus, it is easier to determine the number of chainlinks first, then round this number and derive the distance of sprockets axes from it.

The formula for the number of chainlinks follows:

$$X = 2 \cdot \frac{a}{p} + \frac{z_1 + z_2}{2} + \frac{C \cdot p}{a}$$

X = number of chainlinks
a = axes distance (mm)
p = chainpitch (mm)
z_1 = number of teeth of the small sprocket
z_2 = number of teeth of the big sprocket
$C = \left(\frac{z_2 - z_1}{2\pi} \right)^2$ *see Table F coefficient "C"*



Note: offset link lowers the permissible chain load by as much as 30%

Example of calculation:

Given values: $a = 1500$ mm; $p = 31,75$ mm; $z_1 = 23$; $z_2 = 76$

Put into equation:

$$C = \left(\frac{76 - 23}{2\pi} \right)^2 = 71,19 \quad \text{see Table F , coefficient „ C "}$$

Put into equation:

$$X = 2 \cdot \frac{1500}{31,75} + \frac{23+76}{2} + \frac{71,19.31,75}{1500} = 145,495 = 146 \text{ chainlinks}$$

We select the chain of 146 links and adapt the axes distance.

Table F - coefficient „C" for the determination of the chain length

$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C
1	0,03	21	11,18	41	42,60	61	94,31	81	166,29	101	258,54	121	370,86
2	0,10	22	12,27	42	44,71	62	97,42	82	170,42	102	263,69	122	377,02
3	0,23	23	13,41	43	46,86	63	100,59	83	174,60	103	268,88	123	383,22
4	0,41	24	14,60	44	49,07	64	103,81	84	178,83	104	274,13	124	389,48
5	0,63	25	15,84	45	51,32	65	107,08	85	183,12	105	279,42	125	395,89
6	0,91	26	17,13	46	53,63	66	110,40	86	187,45	106	284,77	126	402,14
7	1,24	27	18,48	47	55,91	67	113,77	87	191,83	107	290,17	127	408,55
8	1,62	28	19,87	48	58,39	68	117,19	88	196,27	108	295,62	128	415,01
9	2,05	29	21,31	49	60,85	69	120,67	89	200,75	109	301,12	129	421,52
10	2,53	30	22,81	50	63,36	70	124,19	90	205,29	110	306,67	130	428,08
11	3,07	31	24,36	51	65,92	71	127,76	91	209,88	111	312,27	131	434,69
12	3,65	32	25,95	52	68,53	72	131,39	92	214,52	112	317,92	132	441,36
13	4,28	33	27,60	53	71,19	73	135,06	93	219,21	113	323,63	133	448,07
14	4,97	34	29,28	54	73,91	74	138,79	94	223,95	114	329,38	134	454,83
15	5,70	35	31,05	55	76,67	75	142,56	95	228,74	115	335,18	135	461,64
16	6,19	36	32,85	56	79,48	76	146,39	96	233,58	116	341,04	136	468,51
17	7,32	37	34,70	57	82,34	77	150,27	97	238,47	117	346,94	137	475,42
18	8,21	38	36,60	58	85,26	78	154,20	98	243,41	118	352,90	138	482,39
19	9,15	39	38,55	59	88,22	79	158,18	99	248,40	119	358,90	139	489,41
20	10,14	40	40,55	60	91,24	80	162,21	100	253,45	120	364,96	140	496,47

6.11. *Determination of the distance of the sprocket axes "a"*

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

X = number of chainlinks
a = axes distance (mm)
p = chainpitch (mm)
z_1 = number of teeth of the small sprocket
z_2 = number of teeth of the big sprocket
F = coefficient given in table G

The maximum permissible (in special cases) axes distance is 6 meters.
Common axes distance is:

a = from 30 to 60 . p

Because of inevitable stretch of the chain in operation, the sprocket axes must be adjustable or chain tensioners must be employed.

TABLE G - 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	1,190	0,8310
11,00	0,8106	2,80	0,8118	1,36	0,8219	1,180	0,8318
10,00	0,8107	2,70	0,8119	1,35	0,8222	1,170	0,8326
9,00	0,8107	2,60	0,8121	1,34	0,8226	1,160	0,8336
8,00	0,8107	2,50	0,8123	1,33	0,8230	1,150	0,8346
7,00	0,8108	2,40	0,8125	1,32	0,8234	1,140	0,8358
6,00	0,8108	2,30	0,8127	1,31	0,8238	1,130	0,8372
5,00	0,8109	2,20	0,8130	1,30	0,8243	1,120	0,8387
4,80	0,8109	2,10	0,8134	1,29	0,8248	1,110	0,8405
4,60	0,8109	2,00	0,8138	1,28	0,8253	1,100	0,8425
4,40	0,8110	1,90	0,8143	1,27	0,8258	1,090	0,8448
4,20	0,8110	1,80	0,8150	1,26	0,8264	1,080	0,8474
4,00	0,8110	1,70	0,8158	1,25	0,8270	1,070	0,8503
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

6.12. *Determination of the diameter of the pitch circles of the sprockets „d_{t1}, d_{t2}“*

$$d_{t1} = \frac{p}{\sin \frac{180°}{z_1}} \quad \text{(mm)}$$

$$d_{t2} = \frac{p}{\sin \frac{180°}{z_2}} \quad \text{(mm)}$$

d_{t1} = pitch circle diameter of the small sprocket (mm)
d_{t2} = pitch circle diameter of the big sprocket (mm)
p = chainpitch (mm)
z_1 = number of teeth of the small sprocket
z_2 = number of teeth of the big sprocket

6.13. Determination of the traction force "P" on the sprockets

$$P = \frac{1000 . N}{v} \quad (N)$$

P = traction force (N)
N = transmitted power (kW)
v = chainvelocity (m/sec.)

6.14. Determination of the centrifugal force "G" on the sprockets

The centrifugal force exerted to the chainlinks depends on the acceleration of their mass.
Its value is calculated according to the following formula:

$$G = Q . v^2 \ (N)$$

G = centrifugal force (N)
Q = weight of 1 meter of chain (kg/m)
v = chainvelocity (m/sec.)

The effect of the centrifugal force is negligible for chain velocities less than 4 m/s.

6.15. Determination of the total chain load “P_c”

In order to determine the total load of the chain, it is necessary to calculate the traction force "P" and the centrifugal force "G":

$$Pc = P + G \quad (N)$$

Pc = total load exerted on chain (N)
P = traction force (N)
G = centrifugal force (N)

The total load calculated this way serves as a base for the selection of the chain type and size.

6.16. Determination of the static safety coefficient „γ_{stat}“

To design a safe chain transmission where all its elements suffice their load, the total load is multiplied by a safety coefficient:

$$\gamma_{stat} = \frac{F_B}{Pc} >= 7$$

γ_{stat} = *static safety coefficient*
F_B = *breaking load (N) – of chain determined from the catalogue of chains*
Pc = total load exerted on chain (N)

The following table lists the recommended safety coefficient for roller chains:

Chainvelocity	*Pitch p < 25,4 mm*	*Pitch p > 25,4 mm*
up to 4 m/sec.	20 – 30	10 – 15
up to 10 m/sec.	30 – 40	15 – 25
over 10 m/sec.	> 40	–

6.17. Determination of the dynamic safety coefficient „γ_{dyn}“

$$\gamma_{dyn} = \frac{F_B}{Pc\,.\,Y} >= 5$$

γ_{dyn} = *dynamic safety coefficient*
F_B = *breaking load (N) - of chain determined from the catalogue of chains*
Pc = *total load exerted on the chain (N)*
Y = *shock coefficient (see Table A_1 or A_2)*

6.18. Determination of the specific pressure on the bearing area „p_i“

TABLE H - specific pressure on bearing area “p_i”

Chain velocity m/sec.	*Specific pressure on the bearing area p_i (MPa) based on the number of teeth of the small sprocket*							
	11	*13*	*15*	*17*	*19*	*21*	*23*	*25*
0,1	31,29	31,29	31,29	31,78	31,98	32,47	32,47	32,86
0,2	27,96	30,02	30,21	30,41	30,41	31,00	31,49	31,89
0,4	25,9	27,57	28,45	28,94	29,33	29,63	29,92	30,51
0,6	24,13	26,09	27,08	27,76	28,15	28,45	29,04	29,72
0,8	22,46	24,53	25,70	26,59	27,08	27,57	27,96	28,55
1,0	21,29	23,35	24,72	25,60	26,39	26,78	27,46	27,96
1,5	18,64	21,19	22,76	24,03	24,62	25,21	25,80	26,19
2,0	16,68	19,33	21,09	22,17	23,35	23,94	24,53	25,11
2,5	15,11	17,95	19,82	20,90	21,88	22,66	23,45	24,13
3,0	(13,64)	16,48	18,54	20,01	20,90	21,68	22,37	23,05
4,0	(11,38)	14,42	16,67	18,15	19,13	20,01	20,70	21,32
5,0	(9,32)	(12,75)	14,91	16,68	17,85	18,77	19,42	20,11
6,0		(11,08)	13,64	15,50	16,58	17,46	18,25	18,93
7,0		(9,61)	(12,35)	14,32	15,60	16,48	17,27	18,05
8,0			(11,18)	(13,34)	14,72	15,60	16,48	17,17
10,0			(9,12)	(11,48)	(13,05)	14,03	14,91	15,60
12,0				(9,91)	(11,67)	(12,85)	13,73	14,42
15,0				(7,85)	(9,99)	(11,18)	(12,16)	12,95

Operating conditions enclosed in brackets are not recommended.

6.19. Determination of the friction coefficient „I_1“

Table I - friction coefficient “I_1“

Shock coefficient Y	*Friction coefficient I_1*														
	a = 20 . p					*a = 40 . p*						*a = 80 . p*			
	$z_2 : z_1$					*$z_2 : z_1$*						*$z_2 : z_1$*			
	1:1	*2:1*	*3:1*	*5:1*	*7:1*	*1:1*	*2:1*	*3:1*	*5:1*	*7:1*	*1:1*	*2:1*	*3:1*	*5:1*	*7:1*
1	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
2	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
3	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
4	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

6.20. Determination of the permissible pressure „$p_{zul.}$“ on the bearing area

$$p_{zul.} = p_i \,.\, I_1 \,.\, I_2 \quad \text{(MPa)}$$

$p_{zul.}$ = *permissible pressure on the bearing area*
p_i = *specific pressure on the bearing area within ideal conditions see Table H (MPa)*
I_1 = *friction coefficient see Table I*
I_2 = *lubrication coefficient see Table D*

6.21. Determination of the specific pressure „p_v“

The formula for determination of the “p_v” value follows:

$$p_v = \frac{Pc}{f} \quad \text{(MPa)} \quad \text{where } f = d_1 \times b_2 \text{ (mm}^2\text{)}$$

p_v = *specific pressure on the bearing area (MPa)*
Pc = *total chain load (N)*
f = *bearing area (mm²)*
d_1 = *chain pin diameter (mm)*
b_2 = *outer width of inner chain element (mm)*

$$p_v < p_{zul.}$$

6.22. Calculation Example

Given values:

Transmitted power	N = 3,5 kW
Revs of driving sprockets	n_1 = 2760/min.
Revs of driven sprockets	n_2 = 920/min.
Number of teeth of the small sprocket	z_1 = 21
Number of teeth of the big sprocket	z_2 = 63
Operating environment	gearbox, oil bath
Driving machinery	electromotor
Driven machinery	piston compressor, 1 stage
Axes distance	a = 500 mm (with chain tensioner)

a) determination of the transmitted power “N“
given, **N = 3,5 kW**

b) determination of the shock coefficient “Y”
- see Table A_1 or A_2
- reading **Y = 2**

c) determination of the transmission ratio “i”

$$i = \frac{n_1}{n_2} = \frac{2760}{920} = 3$$

d) determination of the power coefficient “k”
- see Table B for Y = 2 ; z_1 = 21 ; i = 3
- reading **k = 0,82**

e) determination of the lubrication coefficient “l_2”
- see Table D, the chain runs in oil bath
- $l_2 = 1$

f) determination of the chain type coefficient “φ”

- we choose B modification of chain according to DIN 8187
- $\varphi = 1$

g) determination of the axes distance coefficient “σ”

- given the estimated axes distance a = 500 mm and estimated chainpitch p = 12,7 mm (a = 40 x p)
- $\sigma = 1$

h) determination of the chainvelocity “v”

$$v = \frac{d_{t1} . \pi . n_1}{60000} = \frac{85,12 . \pi . 2760}{60000} = 12,3 \text{ m/s}$$

$$d_t = \frac{p}{\sin \frac{180°}{z_1}} = \frac{12,7}{\sin \frac{180}{21}} = 85,12 \text{ mm}$$

i) determination of the diagrammatic power „Nd“

$$Nd = \frac{N}{k . l_2 . \varphi . \sigma} = \frac{3,5}{0,82 . 1 . 1 . 1} = 4,27 \text{ kW}$$

- see Table E, given values n_1 = 2760/min. and Nd = 4,27 kW
- **reading: chain type 08B = 12,7 x 7,75 single strand**

j) determination of the number of chainlinks „X“

$$X = 2 . \frac{a}{p} + \frac{z_1 + z_2}{2} + \frac{C . p}{a} = 2 \frac{500}{12,7} + \frac{21+63}{2} + \frac{44,71.12,7}{500} = 121,98 \ . \ . \ . \ 122 \text{ chainlinks}$$

- see Table F to determine the value of “C”

k) determination of the exact axes distance „a“

- unnecessary as the transmission involves the chain tensioner

l) determination of the pitch circle diameter of the sprockets “d_{t1}, d_{t2}“

$$d_{t1} = 85,12 \text{ mm}$$

$$d_{t2} = \frac{p}{\sin \frac{180°}{z_2}} = \frac{12,7}{\sin \frac{180}{63}} = 254,78 \text{ mm}$$

m) determination of the traction force “P” on driving sprocket

$$P = \frac{1000 . N}{v} = \frac{1000 . 3,5}{12,3} = 290 \text{ N}$$

n) determination of the centrifugal force “G“ on sprocket

$G = Q.v^2 = 0,7.12,3^2 = 105,903$ N . . . 106 N

Q = 0,7 kg/m according to table in catalogue DIN 8187

o) determination of the total chain load „Pc“

$Pc = P + G = 290 + 106 = \mathbf{396\ N}$

p) determination of the static safety coefficient „γ_{stat}”

$$\gamma_{stat} = \frac{F_B}{Pc} >= 7 \qquad \gamma_{stat} = \frac{18000}{396} = 45{,}46 >= 7$$

F_B = 18000 N according to table in catalogue DIN 8187

q) determination of the dynamic safety coefficient „γ_{dyn}”

$$\gamma_{dyn} = \frac{F_B}{Y\,.\,Pc} >= 5 \qquad \gamma_{dyn} = \frac{18000}{2\,.\,396} = 22{,}73 >= 5$$

- see Table A_2 to determine the value of Y = 2 according to types of driving and driven machinery

r) determination of the specific pressure on bearing area
- see Table H - reading: $p_i = 12{,}85$ **MPa**

s) determination of the friction coefficient “l_1”
- see Table I - reading: $l_1 = 0{,}73$

t) determination of the permissible pressure “$p_{zul.}$“ on bearing area

$p_{zul.} = p_i\,.\,l_1\,.\,l_2 = 12{,}85\,.\,0{,}73\,.\,1 = 9{,}38$ **MPa**

- see Table D for determination of the value of $l_2 = 1$ according to the operating environment

u) determination of the specific pressure “p_v“

$$p_v = \frac{Pc}{f} = \frac{396}{50} = 7{,}92\ \text{MPa}$$

- f = 50 mm^2 according to table in catalogue in DIN 8187

$$p_v < p_{zul.}$$

$$7{,}92 < 9{,}38$$

Single strand chain according to DIN 8187 is sufficient.