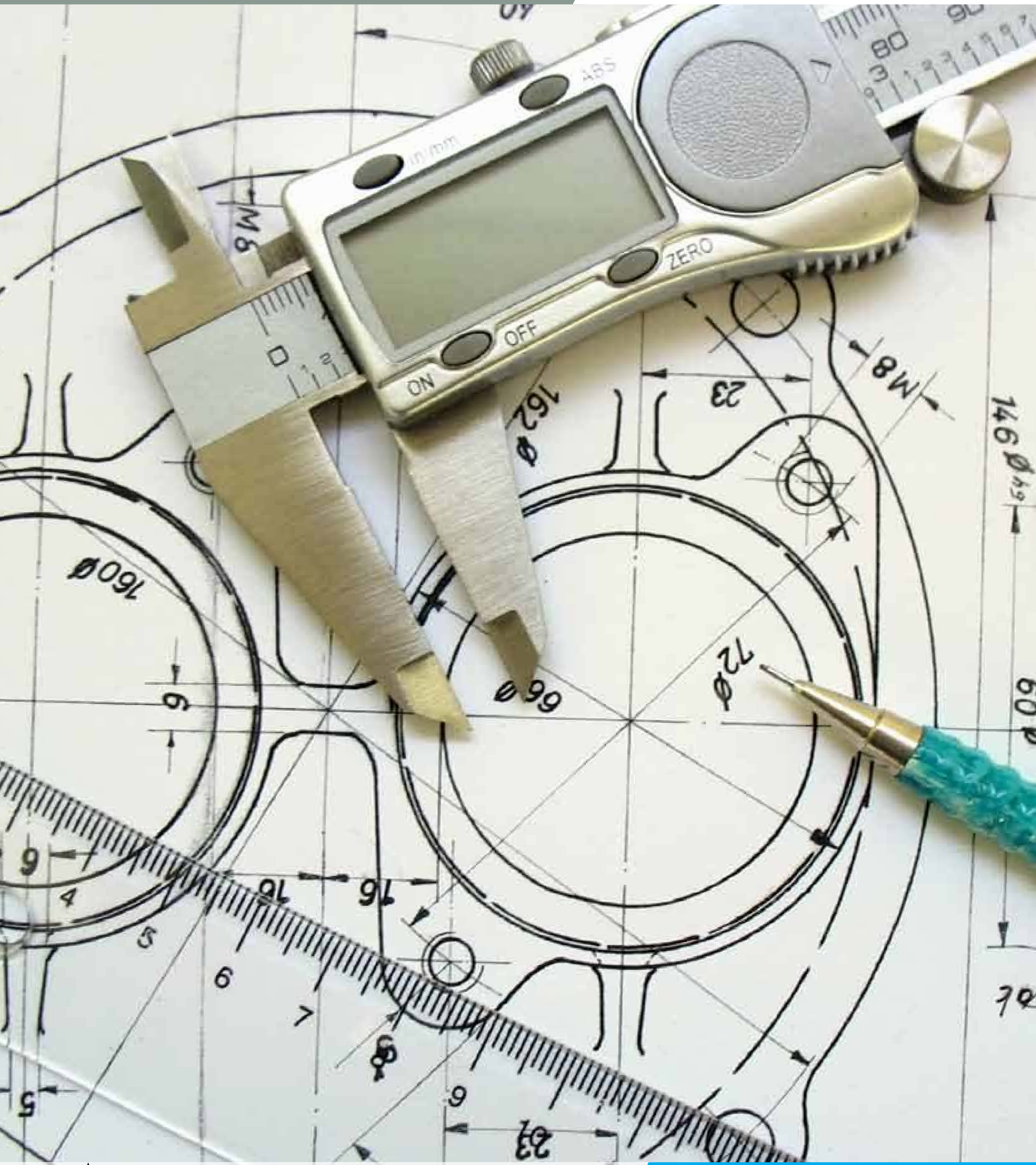
CHAIN DRIVE DESIGN GUIDE

TransDev
THE POWER IN MOTION



Reprinted excerpts from the Wippermann Catalogue 2015.

A TransDev Technical Report

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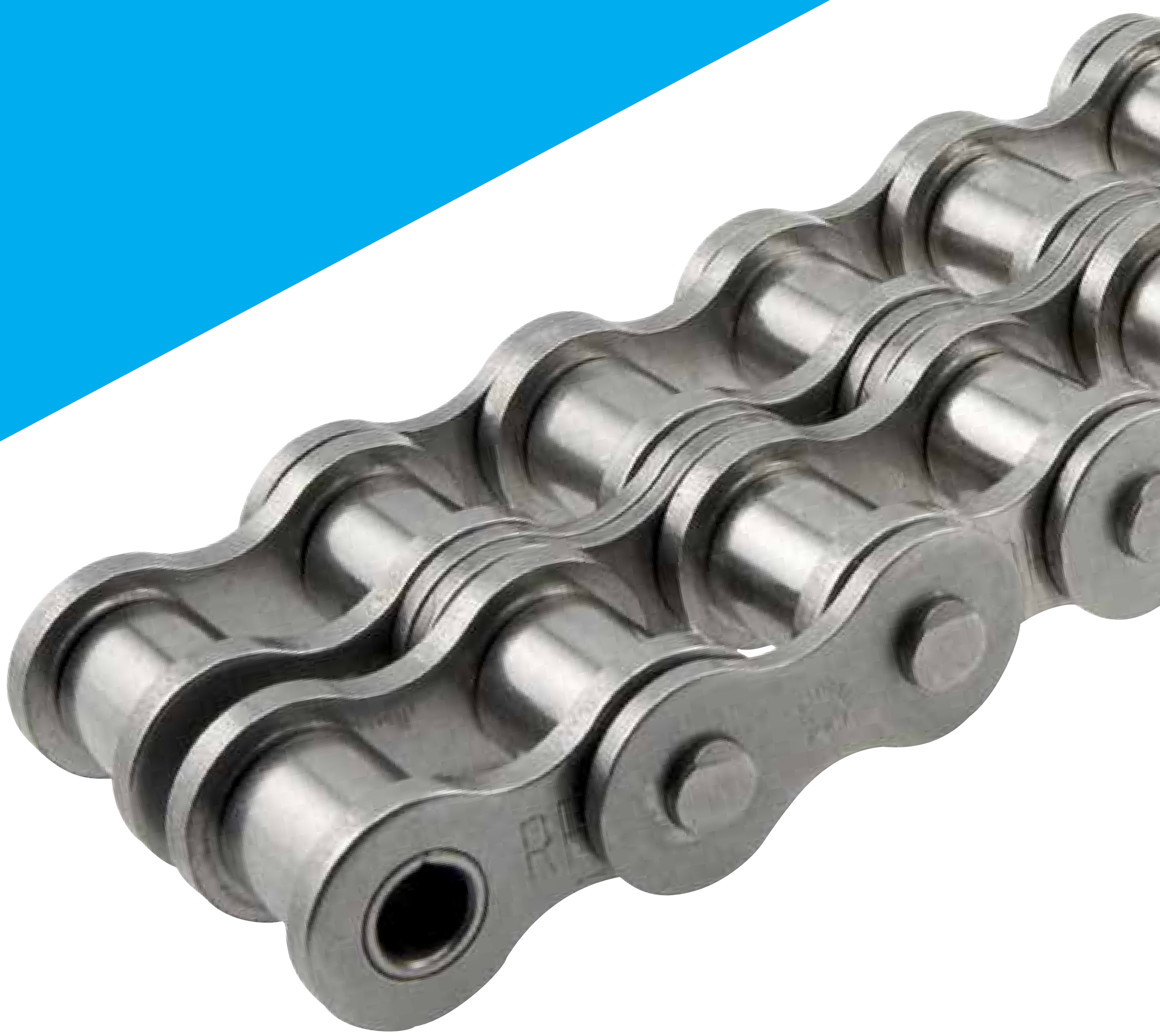
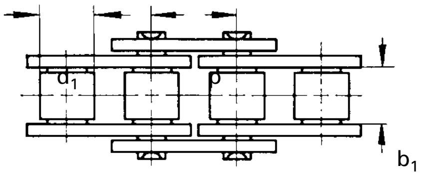
The information in this Technical Guide can also be found in the Wippermann Catalogue 2015. It forms part of a series designed to help you specify and identify the optimum Wippermann chain solution for your needs.





Steel link chains

Generally, steel link chains can only operate on one plane, and they are primarily used as drive elements for chain drives.

They are precisely determined by three main measurements:

p = **Pitch** is the distance from pin centre to pin centre.

b_1 = **Inner width** is the distance between the inner plates.

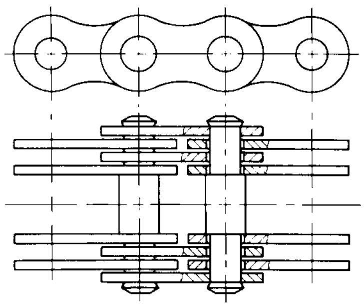
d_1 = **Roller diameter, bushing diameter or pin diameter** is the outer dimension of the cylindrical parts between the inner plates.

The characteristic feature of a steel link chain is the chain joint.

It consists of an outer and an inner link. On this joint the calculated bearing area equals the projection of the pin onto the bearing area of the inner link. It has a different size depending on the type of chain.

In the following overview the characteristic features of various types of steel link chains are briefly described.

Galle chains



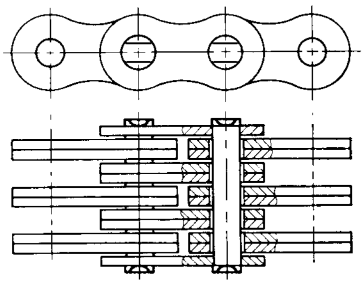
Galle chains were named after their inventor André Galle (1761-1841). A Galle chain is the simplest type of steel link chain.

The plates rotate directly on the pin lug. With this type of chain the bearing area is very small.

Therefore the chain speed should not exceed 0,3 m/s.

Consequently, Galle chains are less suitable for power transmission, and they are almost exclusively used as load chains (e.g. counterweight chains, lock chains and tack chains).

Leaf chains

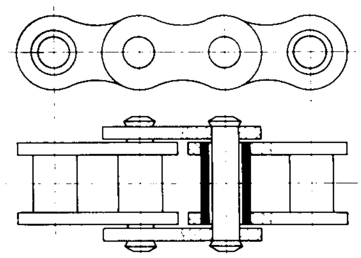


Leaf chains in normal design or reinforced design are used as load chains in cranes, hoisting gear and lifting equipment as well as for counterweights, e.g. on machine tools, and also to transmit back-and-forth movements.

The plates of leaf chains are punched from high-grade steel and are subsequently hardened and tempered to guarantee high fatigue strength. Very narrow tolerances ensure that all plates bear the same load proportions. Pins made of high alloy case-hardened steel are tempered to achieve high wear resistance. The tightly adjoining plates are designed in various combinations and rotate on the pins.

One special design is the heavy-duty type series U. On chains of this type all plates are mounted with a sliding fit and are also secured with laterally attached riveted disks. This design guarantees an even load distribution and reduces the bending load of the pins. These chains were especially developed for heavy loads and operations under harsh conditions. Due to their high fatigue strength they are particularly suitable for such application areas.

Due to their design (no tooth meshing) leaf chains cannot transmit torques. Their force direction, however, can easily be deflected by means of rollers. Even with a small working width they have a high breaking load.

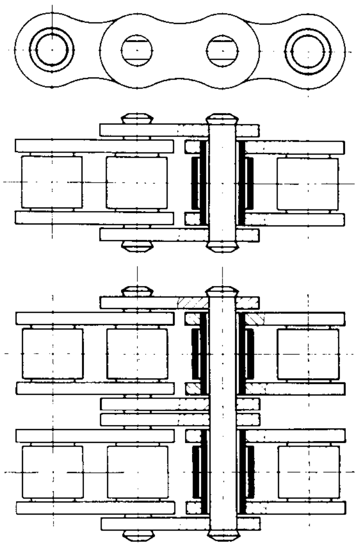


Bush chains

Bush chains are more wear-resistant than Galle chains. The inner links consist of two inner plates with two force-fitted bushings. The outer links consist of two outer plates with two force-fitted and riveted pins.

Chain speeds of up to 5 m/s are possible depending on the pitch.

Due to their robust design bush chains are mainly used as drive and conveyor chains, particularly where there are rough operating conditions, e.g. in mining or construction site equipment.



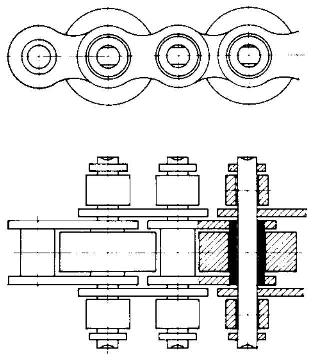
High performance roller chains

Compared to bush chains, high performance roller chains are of better quality due to the use of higher steel grades and heat treatment. Furthermore, they are produced with higher accuracy and narrower tolerances. The visible difference is the rollers, which are mounted on the bushings with running fit, and which absorb the meshing impact in the sprocket and thus reduce sprocket wear. Plates and rollers are hardened and tempered in order to achieve high fatigue strength, whereas bushings and pins, which are subject to wear, are case-hardened.

For high power transmission under restricted mounting conditions multi-strand roller chains can be used. This means that several simplex roller chains are connected by means of an end-to-end pin to form one single unit. Duplex and triplex chains are standardised.

Roller chains can be employed universally and are therefore the most common chain type. They are not only used as drive and gear chains in machine construction, but also in special designs with attachments for transport and conveyance purposes or instead of rack and pinion arrangements.

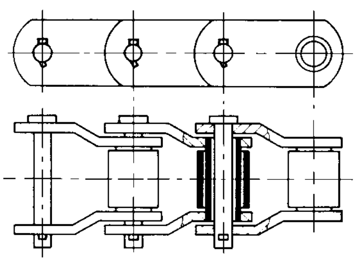
Roller chains RF made of stainless and acid-resistant steel grade 4301 have proved their value on corrosion-endangered drives and because of their anti-magnetic properties for many years. They are mainly used in the chemical, beverage and food industry.



Accumulator chains

Accumulator chains are employed, when accumulation of piece goods during transportation is required. The chain runs on lateral support rollers, whereas the conveyor roller in the middle runs freely.

The particular advantages of this type of chain lie in the simple control, the exact guiding possibilities as well as in the smooth transition from one direction to another without abrupt acceleration. During intentional or unintentional accumulation of the transported piece goods no excessive impact pressure is put on the following transport units since the power and free conveyor chain will continue to run smoothly under the goods until the end of the accumulation, when transportation will continue due to friction.



Cranked link chains (Rotary chains)

Cranked link chains (Rotary chains) are in fact roller chains, but only cranked plates are used. These plates help to give the chain a high amount of elasticity so that load impacts can easily be absorbed. It is also quite straightforward to repair cranked link chains since each individual link can be replaced.

Cranked link chains (Rotary chains) are mainly employed for applications with intermittent impacts and where the drive is exposed to rough soiling, e.g. in excavation machinery, crawlers for excavators and dozers or drilling equipment.

General information

Slack span of the return strand for horizontal drives approx. 1 % to 2 % of the shaft distance.

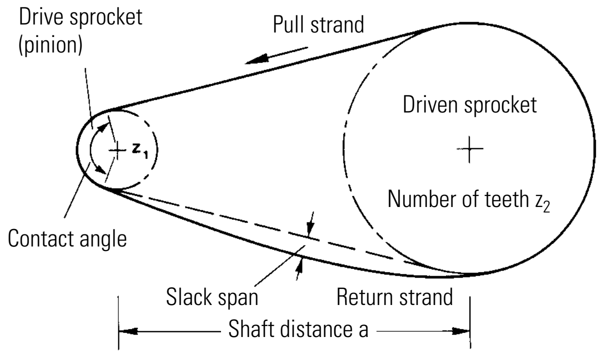
Chain contact angle on the drive sprocket 120° if possible (always the case when $a > d_02 - d_01$)

at least 90° for higher number of teeth ($z \geq 25$).

The shaft distance is normally 30 times p - 50 times p.

minimal $a_{min} > \frac{dk1 + dk2}{2}$

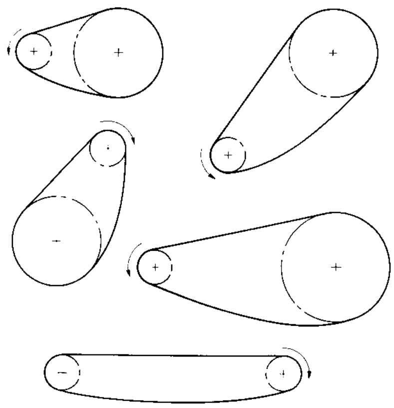
With longer shaft distances, heavy drives or vertical shafts, the chain weight of the pull strand and the return strand must be supported by means of chain support wheels, support rollers or guide strips. The number of teeth on the drive sprocket should be 19 if possible. The minimum number of teeth on a sprocket is 6 ($d_o = 2$ p), which is then only suitable for manual operation because of the polygon effect!



Chain drive configurations (assessment)

Favourable

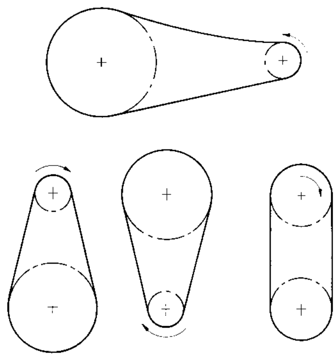
In order to guarantee trouble-free operation and a long wear life, the correct chain run for the different drive configurations has to be selected. A horizontal drive or a configuration with a drive inclined by up to 60° is common and favourable. In this case the pull strand should be at the top and the return strand at the bottom.



Less favourable

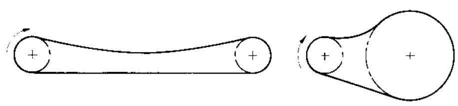
With horizontal drives and normal shaft distances the return strand may also be at the top.

Vertical drives should have the smaller sprocket at the top. The chain must be kept rather tight to stop it from getting slack and jumping off the lower sprocket. A minor deviation from the vertical position will improve the running conditions. It might be necessary to mount a jockey sprocket.



Please avoid if possible

In case of short or long shaft distances the pull strand should be at the top if possible!



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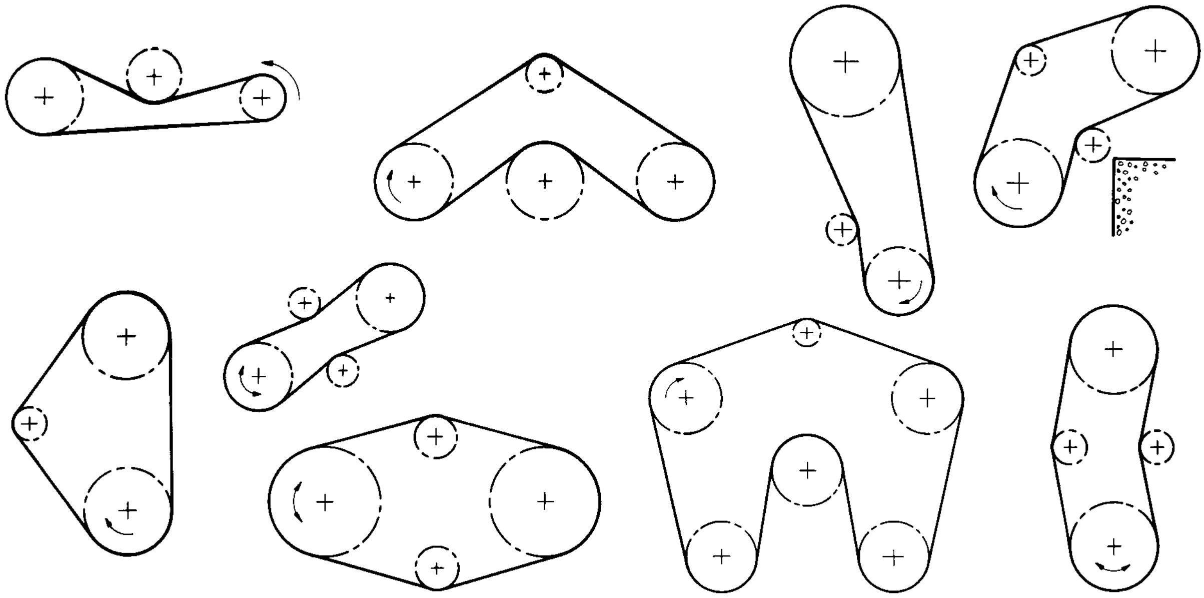
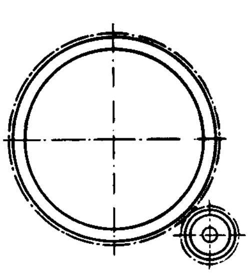
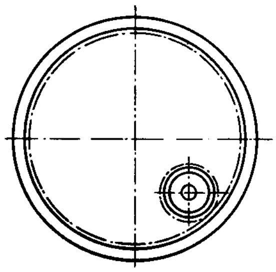
Jockey sprockets should have approximately three teeth in mesh with the return strand of the chain. On the basis of the selected number of teeth, the maximum speed (see page 119 “ratio between n and p“) must not be exceeded.



Instead of jockey sprockets, support wheels or deflexion pulleys, plastic guide rails might be advantageous in some cases to support or deflect a chain.



a) as outer sprocket

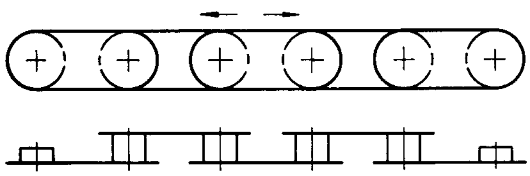


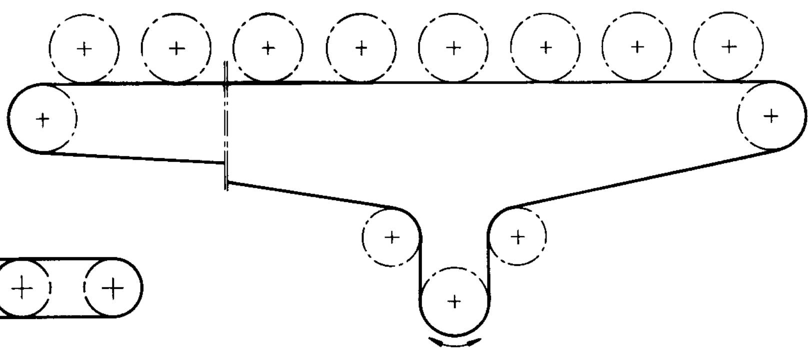
b) as inner sprocket

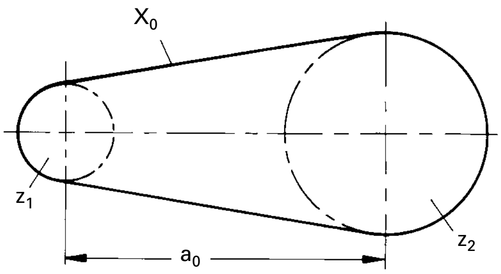
Roller chain instead of a sprocket for large wheel bodies, drums, revolving platforms etc.

Driving of roller conveyors

a) by means of alternate individual chain strands driving from roller to roller
b) by means of a circulating chain with lantern gear toothing sprockets (p.88)







With the same number of teeth $z_1 = z_2$ the chain length is:

$$X_0 = 2\frac{a_0}{p} + z$$

With different numbers of teeth z_1 and z_2 the chain length is:

$$X_0 = 2\frac{a_0}{p} + \frac{z_1 + z_2}{2} + \frac{C\,p}{a_0}$$

X = Chain length in links
X_0 = Theoretical chain length
a = Shaft distance in mm
a_0 = Theoretical shaft distance
p = Pitch in mm
z_1 = Number of teeth on small sprocket
z_2 = Number of teeth on large sprocket
C = Coefficient from table

$$C = \left(\frac{z_2 - z_1}{2\pi}\right)^2$$

The calculated number of links must always be rounded up. In case of minor differences, one pitch should be added in order to avoid assembly difficulties. If the calculation results in an uneven number of chain links, one single cranked link (0,8 of breaking load) has to be mounted. In such cases it is recommended to select the next even number of links. Then the exact shaft distance can easily be calculated according to the detailed information on page 124.

Values für "C" = $\left(\frac{z_2 - z_1}{2\pi}\right)^2$

$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C	$z_2 - z_1$	C
1	0,025	41	42,58	81	166,19	121	370,86
2	0,101	42	44,68	82	170,32	122	377,02
3	0,228	43	46,84	83	174,50	123	383,22
4	0,405	44	49,04	84	178,73	124	389,48
5	0,633	45	51,29	85	183,01	125	395,79
6	0,912	46	53,60	86	187,34	126	402,14
7	1,240	47	55,95	87	191,73	127	408,55
8	1,620	48	58,36	88	196,16	128	415,01
9	2,050	49	60,82	89	200,64	129	421,52
10	2,530	50	63,33	90	205,18	130	428,08
11	3,070	51	65,88	91	209,76	131	434,69
12	3,650	52	68,49	92	214,40	132	441,36
13	4,280	53	71,15	93	219,08	133	448,07
14	4,960	54	73,86	94	223,82	134	454,83
15	5,700	55	76,62	95	228,61	135	461,64
16	6,480	56	79,44	96	233,44	136	468,51
17	7,320	57	82,30	97	238,33	137	475,42
18	8,210	58	85,21	98	243,27	138	482,39
19	9,140	59	88,17	99	248,26	139	489,41
20	10,130	60	91,19	100	253,30	140	496,47
21	11,170	61	94,25	101	258,39	141	503,59
22	12,260	62	97,37	102	263,54	142	510,76
23	13,400	63	100,54	103	268,73	143	517,98
24	14,590	64	103,75	104	273,97	144	525,25
25	15,830	65	107,02	105	279,27	145	532,57
26	17,120	66	110,34	106	284,61	146	539,94
27	18,470	67	113,71	107	290,01	147	547,36
28	19,860	68	117,13	108	295,45	148	554,83
29	21,800	69	120,60	109	300,95	149	562,36
30	22,800	70	124,12	110	306,50	150	569,93
31	24,340	71	127,69	111	312,09	151	577,56
32	25,940	72	131,31	112	317,74	152	585,23
33	27,580	73	134,99	113	323,44	153	592,96
34	29,280	74	138,71	114	329,19	154	600,73
35	31,030	75	142,48	115	334,99	155	608,56
36	32,830	76	146,31	116	340,84	156	616,44
37	34,680	77	150,18	117	346,75	157	624,37
38	36,580	78	154,11	118	352,70	158	632,35
39	38,530	79	158,09	119	358,70	159	640,38
40	40,530	80	162,11	120	364,76	160	648,46

Example:

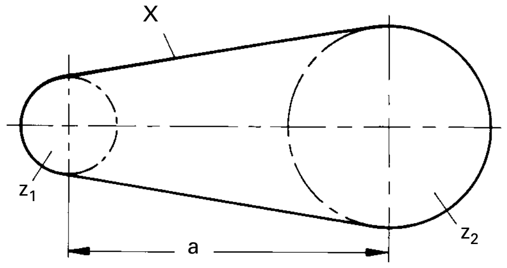
a_0 = 700 mm z_1 = 19
p = 19,05 mm z_2 = 45
C = 17,12 (für $z_2 - z_1$ =26)

$$X_0 = 2\frac{a_0}{p} + \frac{z_1 + z_2}{2} + \frac{C \cdot p}{a_0}$$

$$X_0 = \frac{2 \times 700}{19{,}05} + \frac{19 + 45}{2} + \frac{17{,}12 \times 19{,}05}{700}$$

X_0 = 73,49 + 32 + 0,466 = 105,956

X = 106 links



a = Shaft distance in mm
X = Chain length in links
p = Pitch in mm
z_1 = Number of teeth on small sprocket
z_2 = Number of teeth on large sprocket

The calculation of a chain length rarely results in an even number of links. Mostly, the result must be rounded up. In order to avoid a cranked link in the chain, an even number should be selected.

The exact shaft difference is calculated according to the following formulas:

With the same number of teeth $z_1 = z_2 = z$ the shaft distance is:

$$a = \frac{X - z}{2}\, p$$

With an uneven number of teeth z_1 and z_2 the shaft distance is:

$$a = p\,[2\,X - (z_1 + z_2)]\;B$$

The coefficient "B" is a function of $K = \frac{X - z_1}{z_2 - z_1}$ and can be taken from the following table.

Example:

X = 106 links $z_1 = 19$
p = 19,05 mm $z_2 = 45$

$$a = p\,[2\,x - (z_1 + z_2)]\,B$$

$$k = \frac{X - z_1}{z_2 - z_1} = \frac{106 - 19}{45 - 19} = \frac{87}{26} = 3{,}34615$$

The table shows a value B = 0,24825 for K = 3,2
and a value B = 0,24849 for K = 3,4

B must be calculated by means of interpolation.
The following applies:

$$\frac{\text{Difference K times table difference B}}{\text{Table difference K}}$$

$$B = 0{,}24825 + \frac{(3{,}34615 - 3{,}2) \times (0{,}24849 - 0{,}24825)}{3{,}4 - 3{,}2}$$

$$B = 0{,}24825 + \frac{0{,}14615 \times 0{,}00024}{0{,}2}$$

B = 0,24825 + 0,00017538 = 0,24843 (rounded up)

The exact shaft distance is

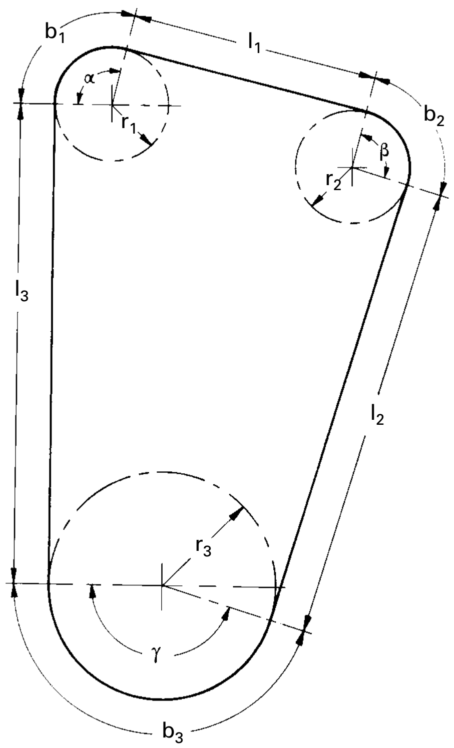
a = 19,05 (2 x 106 - 19 - 45) 0,24843

a = 700,4 mm

Coefficient "B"

K	B	K	B	K	B	K	B
13,0	0,24 991	**2,70**	0,24 735	**1,54**	0,23 758	**1,26**	0,22 520
12,0	990	**2,60**	708	**1,52**	705	**1,25**	443
11,0	988	**2,50**	678	**1,50**	648	**1,24**	361
10,0	986	**2,40**	643	**1,48**	588	**1,23**	275
9,0	983	**2,30**	602	**1,46**	524	**1,22**	185
8,0	978	**2,20**	552	**1,44**	455	**1,21**	090
7,0	970	**2,10**	493	**1,42**	381	**1,20**	0,21 990
6,0	958	**2,00**	421	**1,40**	301	**1,19**	884
5,0	937	**1,95**	380	**1,39**	259	**1,18**	771
4,8	931	**1,90**	333	**1,38**	215	**1,17**	652
4,6	925	**1,85**	281	**1,37**	170	**1,16**	526
4,4	917	**1,80**	222	**1,36**	123	**1,15**	390
4,2	907	**1,75**	156	**1,35**	073	**1,14**	245
4,0	896	**1,70**	081	**1,34**	022	**1,13**	090
3,8	883	**1,68**	048	**1,33**	0,22 968	**1,12**	0,20 923
3,6	868	**1,66**	013	**1,32**	912	**1,11**	744
3,4	849	**1,64**	0,23 977	**1,31**	854	**1,10**	549
3,2	825	**1,62**	938	**1,30**	793	**1,09**	336
3,0	795	**1,60**	897	**1,29**	729	**1,08**	104
2,9	778	**1,58**	854	**1,28**	662	**1,07**	0,19 848
2,8	758	**1,56**	807	**1,27**	593	**1,06**	564

K > 13 B = 0,25



If a chain runs on several sprockets (as shown in the drawing), graphics will usually suffice to determine the chain length since this method is sufficiently accurate and considerably simpler than mathematical calculations. To begin with, the drive is drawn schematically, if possible on a scale of 1:1 or larger. Then tangents are drawn to the pitch circles, and the central angles of the circular arc spanned by the chain are determined.

For the respective arc values please refer to the table "arc lengths".

The chain length L can then be calculated by adding up the partial lengths.

$L = l_1 + l_2 + l_3 + \ldots + b_1 + b_2 + b_3 \ldots$

$X = L/p$

The result must always be rounded up, if possible to the next even number of links. Uneven numbers should be avoided!

Arc lengths for the radius r = 1

Central angle $\varphi°$	Arc length arc φ	Central angle $\varphi°$	Arc length arc φ	Central angle $\varphi°$	Arc length arc φ	Central angle $\varphi°$	Arc length arc φ
1	0,0175	46	0,8029	91	1,5882	136	2,3736
2	0,0349	47	0,8203	92	1,6057	137	2,3911
3	0,0524	48	0,8378	93	1,6232	138	2,4086
4	0,0698	49	0,8552	94	1,6406	139	2,4260
5	0,0873	50	0,8727	95	1,6580	140	2,4435
6	0,1047	51	0,8901	96	1,6755	141	2,4609
7	0,1222	52	0,9076	97	1,6930	142	2,4784
8	0,1396	53	0,9250	98	1,7104	143	2,4958
9	0,1571	54	0,9425	99	1,7279	144	2,5133
10	0,1745	55	0,9599	100	1,7453	145	2,5307
11	0,1920	56	0,9774	101	1,7628	146	2,5482
12	0,2094	57	0,9948	102	1,7802	147	2,5656
13	0,2269	58	1,0123	103	1,7977	148	2,5831
14	0,2443	59	1,0297	104	1,8151	149	2,6005
15	0,2618	60	1,0472	105	1,8326	150	2,6180
16	0,2793	61	1,0647	106	1,8500	151	2,6354
17	0,2967	62	1,0821	107	1,8675	152	2,6529
18	0,3142	63	1,0996	108	1,8850	153	2,6704
19	0,3316	64	1,1170	109	1,9024	154	2,6878
20	0,3491	65	1,1345	110	1,9199	155	2,7053
21	0,3665	66	1,1519	111	1,9373	156	2,7227
22	0,3840	67	1,1694	112	1,9548	157	2,7402
23	0,4014	68	1,1868	113	1,9722	158	2,7576
24	0,4189	69	1,2043	114	1,9897	159	2,7751
25	0,4363	70	1,2217	115	2,0071	160	2,7925
26	0,4538	71	1,2392	116	2,0246	161	2,8100
27	0,4712	72	1,2566	117	2,0420	162	2,8274
28	0,4887	73	1,2741	118	2,0595	163	2,8449
29	0,5061	74	1,2915	119	2,0769	164	2,8623
30	0,5236	75	1,3090	120	2,0944	165	2,8798
31	0,5411	76	1,3265	121	2,1118	166	2,8972
32	0,5585	77	1,3439	122	2,1293	167	2,9147
33	0,5760	78	1,3614	123	2,1468	168	2,9322
34	0,5934	79	1,3788	124	2,1642	169	2,9496
35	0,6109	80	1,3963	125	2,1817	170	2,9671
36	0,6283	81	1,4137	126	2,1991	171	2,9845
37	0,6458	82	1,4312	127	2,2166	172	3,0020
38	0,6632	83	1,4486	128	2,2340	173	3,0194
39	0,6807	84	1,4661	129	2,2515	174	3,0369
40	0,6981	85	1,4835	130	2,2689	175	3,0543
41	0,7156	86	1,5010	131	2,2864	176	3,0718
42	0,7330	87	1,5184	132	2,3038	177	3,0892
43	0,7505	88	1,5359	133	2,3213	178	3,1067
44	0,7679	89	1,5533	134	2,3387	179	3,1241
45	0,7854	90	1,5708	135	2,3562	180	3,1416

L	= Chain length in mm
X	= Chain length in links
p	= Pitch in mm
$l_{1, 2, 3}$	= Tangent lengths in mm
$r_{1, 2, 3}$	= Pitch circle radiuses in mm
α, β, γ	= Central angles in degrees
$b_{1, 2, 3}$	= Arc lengths in mm
	= r_1 arc α, r_2 arc β, r_3 arc γ

Example:

(see above drawing)

Chain pitch p = 15,875 mm

r_1 = 43,2 mm α = 104° l_1 = 188 mm

r_2 = 43,2 mm β = 93° l_2 = 345 mm

r_3 = 86,0 mm γ = 163° l_3 = 363 mm

$b_1 = r_1$ arc α = 43,2 x 1,8151 = 78,41 mm

$b_2 = r_2$ arc β = 43,2 x 1,6232 = 70,12 mm

$b_3 = r_3$ arc γ = 86,0 x 2,8449 = 244,66 mm

$L = b_1 + b_2 + b_3 + l_1 + l_2 + l_3$

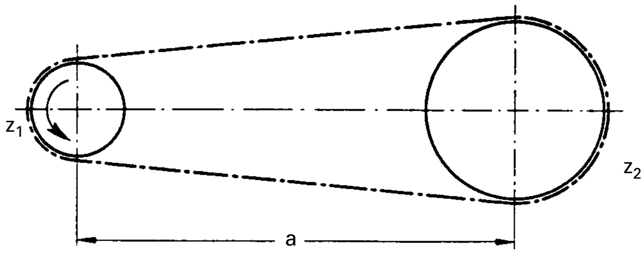
$= 78{,}41 + 70{,}12 + 244{,}66 + 188 + 345 + 363$

$= 1289{,}19$ mm

$X = \frac{L}{p} = \frac{1{,}289{,}19}{15{,}875} = 81{,}21 = \underline{\underline{82 \text{ links}}}$

1. Given are:

(Refer to the drawing in example 1, which illustrates this worked example)



Input power	$P = 0{,}16$ kW
Input speed	$n_1 = 36$ min^{-1}
Output speed	$n_2 = 10{,}75$ min^{-1}
Transmission ratio	$i = \frac{n_1}{n_2} = 3{,}35$
Mode of drive	electric gear motor
Driven machine	Conveyor (with uneven charging)
Approx. shaft centre distance	$a_0 \approx 530$ mm

2. Selection of sprockets

Selected number of teeth on drive sprocket: $z_1 = 17$
Number of teeth on driven sprocket: $z_2 = i \cdot z_1$; $z_2 = 3{,}35 \cdot 17 = 57$

3. Calculations and selection of chain

3.1 Correction of chain

Correction factor for operating conditions: $k = f_y \cdot f_i \cdot f_z$ ($f_y = 1{,}4$; $f_i = 1$; $f_z = 1{,}13$)
Correction factor for number of teeth: $k = 1{,}4 \cdot 1 \cdot 1{,}13$
Corrected power: $P_C = P \cdot k$

$P_C = 0{,}16$ kW $\cdot\, 2{,}17$

$P_C = 0{,}35$ kW

3.2 Selection of chain

For $P_C = 0{,}35$ kW and $n_1 = 36$ rpm the roller chain 10A-1 or 10B-1 is selected from the power diagrams (see pages 117-119)
The chain pitch p for a chain 10A-1 or 10B-1 is 15,875 mm (according to ISO 606).

3.3 Chain length

Calculation of number of links

$$X_0 = 2\frac{a_0}{p} + \frac{z_1 + z_2}{2} + \frac{C \cdot p}{a_0}$$

Here $C = 40{,}529$ for $z_2 - z_1 = 57 - 17 = 40$
Result:

$$X_0 = \frac{530}{15{,}875} + \frac{17 + 57}{2} + \frac{40{,}529 \cdot 15{,}875}{530}$$

$X_0 = 104{,}99$

Selected number of links X = 106 (i.e. the next higher even number).

3.4 Chain speed

$$v = \frac{n \cdot z \cdot p}{60\,000} = \frac{36 \cdot 17 \cdot 15{,}875}{60\,000} = 0{,}16 \text{ m/s}$$

4. Maximum shaft centre distance of sprockets

Maximum shaft centre distance:
$a = p\,[2\,X - (z_1 + z_2)]\,B$

Results $B = 0{,}24567$ für $\frac{X - z_1}{z_2 - z_1} = \frac{106 - 17}{57 - 17} = 2{,}23$ (interpolated)

This is the value for the shaft centre distance:
$a = 15{,}875\,[\,2 \cdot 106 - (17 + 57)\,]\,0{,}24567$
$a = 538{,}2$ mm

5. Lubrication

For v = 0,16 m/s and for a chain type 10A-1 or 10B-1 the diagram (page 109) shows the lubrication range I. Consequently, the simplest lubrication method, i.e. regular manual oil lubrication, will be sufficient in this case.

Conditions/Symptoms	Possible cause	What to do
One-sided wear on chains and sprockets	1. Shafts not parallel, sprocket and pinion not aligned	1. Realign
Wear on inner plates or on sides of sprocket teeth	1. Sprocket and pinion not aligned or shaft wobble	1. Realign sprockets
Wear on tooth heads	1. Chain elongation 2. Tooth error	1. Replace chain 2. Replace pinion and sprocket
Wear on tooth flanks, sprockets	1. Low material strength	1. Exchange for hardened sprockets
Wear on outer plates	1. Chain striking an obstruction	1. Make sure chain is not obstructed
Chain vibrates with high frequency	1. Eccentricity or sprocket wobble 2. Broken chain roller	1. Replace sprockets 2. Replace chain links or chain
Premature elongation	1. Insufficient lubrication or wrong chain size	1. Increase oil supply and check chain size
Rust-coloured discolouration of chain and pins	1. Insufficient lubrication	1. Improve lubrication
Chain jumps off sprocket	1. Excess chain slack 2. Chain riding too high on sprocket teeth due to chain wear	1. Adjust shaft centre distance or jockey sprocket 2. Replace chain
Broken chain parts	1. Drive overloaded 2. Excess chain slack and chain jumps off sprocket 3. Chain striking solid object 4. Chain speed too high 5. Imprecise toothing on the sprockets 6. Insufficient lubrication 7. Corrosion	1. Select another chain or avoid overload 2. Regular check and adjustment of shaft centre distance 3. Make sure chain is not obstructed 4. Check chain dimensioning 5. Change sprockets 6. Improve and increase lubrication 7. Avoid corrosion or use chains made of stainless material (please enquire)
Excessive noise	1. Chain striking an obstruction 2. Insufficient lubrication 3. Missing or broken rollers 4. Misalignment 5. Chain jumps off sprocket	1. Make sure chain is not obstructed 2. Improve lubrication 3. Replace chain or defective parts 4. Align shafts and sprockets 5. Re-adjust shaft centre distance

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Designation	Symbol	Unit	Basic equations
Input speed	n	min^{-1}	
Operating factor	k		$k = f_y \cdot f_i \cdot f_z$
Minimum tensile strength	F_B	N	see chain tables
Torque	M	Nm	$M = \frac{9550\,P}{n} = \frac{F \cdot d_0}{2000}$ in Nm
Correction factor for impact loads	f_y		see page 118
Correction factor for transmission ratio	f_i		see page 119
Correction factor for shaft distance	f_a		see page 119
Correction factor for number of teeth	f_z		see page 119
Bearing area	f	cm^2	see chain tables
Bearing pressure	p_r	N/cm^2	$p_r = \frac{F}{f}$ see page 117
Speed	v	m/s	$v = \frac{z \cdot p \cdot n}{60\,000}$ in m/s
Weight of chain per meter	q	kg/m	see chain tables
Power	P	kW	$P = \frac{F \cdot v}{1000} = \frac{M \cdot n}{9550}$ in kW
Diagram power	P_c	kW	$P_c = P \cdot k$ in kW
Safety factor	S		$S = \frac{F_B}{F_G}$
Impact coefficient	Y		see page 118
PCD	d_0	mm	$d_0 = \frac{p}{\sin \frac{180°}{z}}$ in mm
Pitch	p	mm	see chain tables
Transmission ratio	o		$i = \frac{n_1}{n_2} = \frac{z_2}{z_1}$
Shaft distance	a	mm	
Number of teeth	z_1, z_2		
Tensile force	F	N	$F = \frac{1000\,P}{V} = \frac{2000\,M}{d_0}$ in N
Tensile force, dynamic	F_d	N	$F_d = F \cdot f_y$ in N
Tensile force, centrifugal	F_F	N	$F_F = q \cdot v^2$ in N
Tensile force, total	F_G	N	$F_G = F_d + F_F$ in N

Questionnaire for chain drives

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What is to be conveyed or driven by the chain? (If an existing chain drive is to be replaced, please state which one!)

....

....

Chain drive

Please underline where applicable and enter the respective data if necessary!

Power requirement
(max. power to be transmitted)
power output P = ... PS/kW torque M = ... Nm tensile force F = ... N

Drive
(type and performance)
.... / .... hp/kW
(e.g. electric motor, internal combustion engine / 2, 4, 6 cylinders etc.)

Chain loading
operation period .... hours/day
□ regular □ cyclic □ impact □ alternating direction ....times per hour
□ interruption (re-start) approx. .... times per hour

Centrifugal mass for impact compensation
□ existing □ possible □ not existing □ not possible

Axial distance
a = ... mm
shaft distance is adjustable by .... mm / not adjustable
□ jockey sprocket □ clamping rail □ clamping spring □ automatic chain tensioner

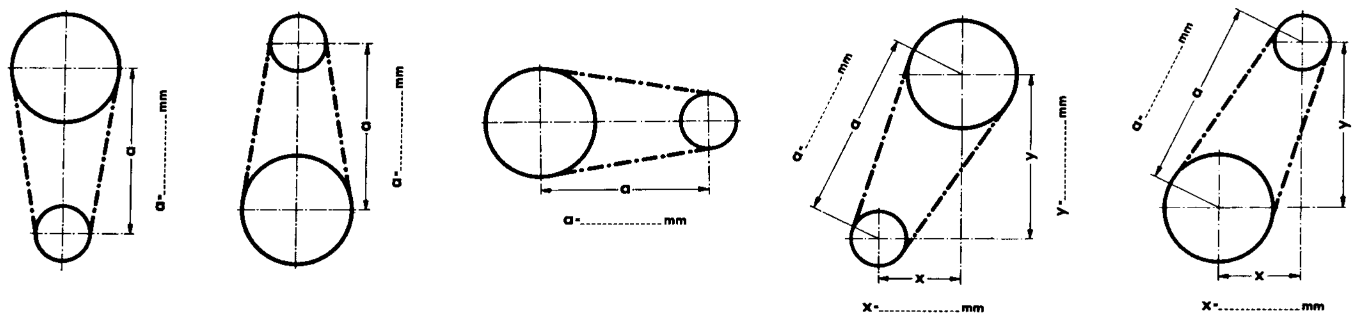
Ambient influences
□ nothing in particular □ dust □ fibres □ sand □ humidity
temperatures up to .... °C corrosion caused by ....

Chain protection box
□ dust proof □ not dust proof □ installation not possible
□ chain unprotected □ chain protected by engine / machine housing

Lubrication
□ not permitted □ manually (occasionally) □ drip feed
□ oil bath □ pressure circulation

Sprockets

	Driving sprocket	Driven sprocket
Speed	n_1 = ... rpm	n_2 = ... rpm
or		
planned transmission ratio	i = ...:..................	
Sprocket diameter (Ø) Largest possible incl. chain	max. = ... mm	max. = ... mm
Sprocket width Largest possible incl. chain	max. = ... mm	max. = ... mm
Sprocket design	....	....
Hub bore (shaft Ø)	d_1 = ... mm	d_2 = ... mm
Hub length	L_1 = ... mm	L_2 = ... mm
Hub design One-sided: standard Double-sided: symmetrical or asymmetrical	....	....
Installation on the shaft (groove sizes according to DIN)	....	....



Please enter the dimensions of the requested drive into the drawing. The driving wheel designation should be T. Please indicate the rotation direction by an arrow and in case of alternating rotation direction by a double arrow (◄—►).

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Transmission Developments Co (GB) Ltd
Dawkins Rd, Poole, Dorset, BH15 4HF
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