INTERNATIONAL STANDARD

ISO 10823

Second edition
2004-02-01

Guidelines for the selection of roller chain drives

Méthode de sélection des transmissions par chaîne à rouleaux

Reference number
ISO 10823:2004(E)

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Foreword

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ISO 10823 was prepared by Technical Committee ISO/TC 100, *Chains and chain wheels for power transmission and conveyors*.

This second edition cancels and replaces the first edition (ISO 10823:1996), which has been technically revised.

Guidelines for the selection of roller chain drives

1 Scope

This International Standard gives guidelines for the selection of chain drives, composed of a roller chain and sprockets conformant with ISO 606, for industrial applications.

The selection procedures and the chain ratings it describes provide for roller chain drives operating under specified conditions, as defined in 9.1, 9.2 and in Clause 10, with a life expectancy of approximately 15 000 h.

Owing to the wide variations in loading characteristics, environmental conditions and achieved maintenance, it is desirable that the supplier of the chains and sprockets be consulted to ensure that the performance of the product meets the requirements specified both by the user and by this International Standard.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 606, *Short-pitch transmission precision roller chains and chain wheels*

3 Symbols

The symbols and units used in this International Standard are given in Table 1.

4 Basic equations

4.1 Input power

The power to be transmitted is the input P, in kilowatts, to the drive sprocket. If input torque is the known requirement, then P can be derived from the following equation:

$$P = \frac{M \times n_1}{9\,550} \tag{1}$$

4.2 Corrected power

To allow for the characteristics of the drive system and the type of load to be transmitted, the input power, P, is multiplied by factors to obtain the corrected power, P_c.

$$P_c = P \times f_1 \times f_2 \tag{2}$$

Table 1 — Symbols, designations and units

Symbol	Designation	Unit
a	Maximum centre distance	mm
a_0	Approximate centre distance	mm
f_1	Application factor to allow for the operating conditions (see Table 2)	—
f_2	Factor for number of teeth on small sprocket [see Figure 4 and Equation (5)]	—
f_3	Factor for calculation of the number of links with different number of teeth (see Table 5)	—
f_4	Factor for the calculation of the centre distance with different numbers of teeth (see Table 6)	—
i	Speed ratio	—
M	Input torque	N·m
n_1	Input sprocket speed	min^{-1}
n_2	Output sprocket speed	min^{-1}
n_s	Small sprocket speed	min^{-1}
p	Chain pitch	mm
P	Input power	kW
P_c	Corrected power	kW
v	Chain speed	m·s^{-1}
X	Number of pitches in chain	—
X_0	Calculated number of pitches in chain	—
z_1	Number of input sprocket teeth	—
z_2	Number of output sprocket teeth	—
z_s	Number of small sprocket teeth	—

5 Drive design specifications

The following design features should be specified before the chain and sprockets are selected:

a) power to be transmitted;

b) type of driver and driven machinery;

c) speeds and sizes of the driver and driven shafts;

d) centre distance and layout of the shafts;

e) environmental conditions.

NOTE Shaft sizes, unusually long or short centre distances, and/or a complex layout could influence the drive selection.

6 Sprocket selection

Determine the number of teeth on the sprockets using the following procedure:

a) select the desired number of teeth for the input sprocket;

b) determine the speed ratio, i, using the equation:

$$i = \frac{n_1}{n_2} \tag{3}$$

c) determine the number of teeth on the output sprocket, z_2, using the equation:

$$z_2 = i \times z_1 \tag{4}$$

It is good practice to use sprockets with not less than 17 teeth and not more than 114 teeth.

If the chain drive operates at high speed or if it is subjected to impulse loads, the small sprocket should have at least 25 teeth and the teeth should be hardened.

7 Chain calculations and selection

7.1 Normal operating conditions and drive capacities for chains

The typical capacity rating charts shown in Figures 1, 2 and 3 apply to chain drives operating under the following conditions:

a) a chain drive with two sprockets on parallel horizontal shafts;

b) a small sprocket with 19 teeth;

c) a simplex chain without cranked link;

d) a chain length of 120 pitches (different chain lengths will affect chain life);

e) a speed ratio of from 1:3 to 3:1;

f) an expected life of 15 000 h;

g) an operating temperature between –5 °C and +70 °C;

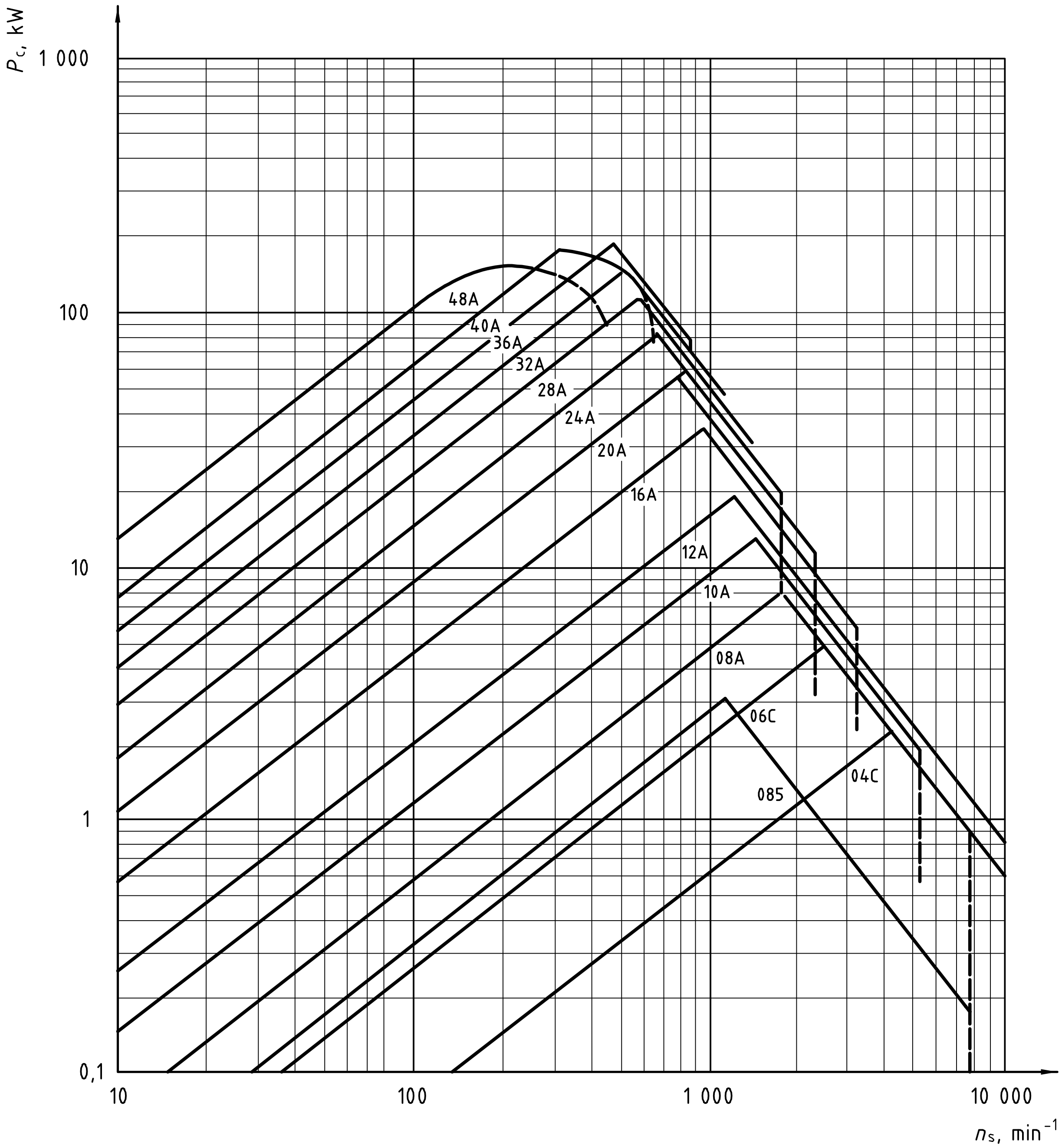
h) sprockets correctly aligned and chain maintained in correct adjustment (see Clause 10);

i) uniform operation without overload, shocks or frequent starts;

j) clean and adequate lubrication throughout the chain's life (see Clause 9).

Figures 1, 2 and 3 can be used to select the size of chain suitable for a chain drive as a function of the corrected power, P_c, and the small sprocket rotational speed, n_s.

The capacity rating charts given in Figures 1, 2 and 3 are representative of those published by chain manufacturers. Individual manufacturers can rate their chains differently. It is therefore recommended that the appropriate manufacturer's rating chart be consulted.



Key

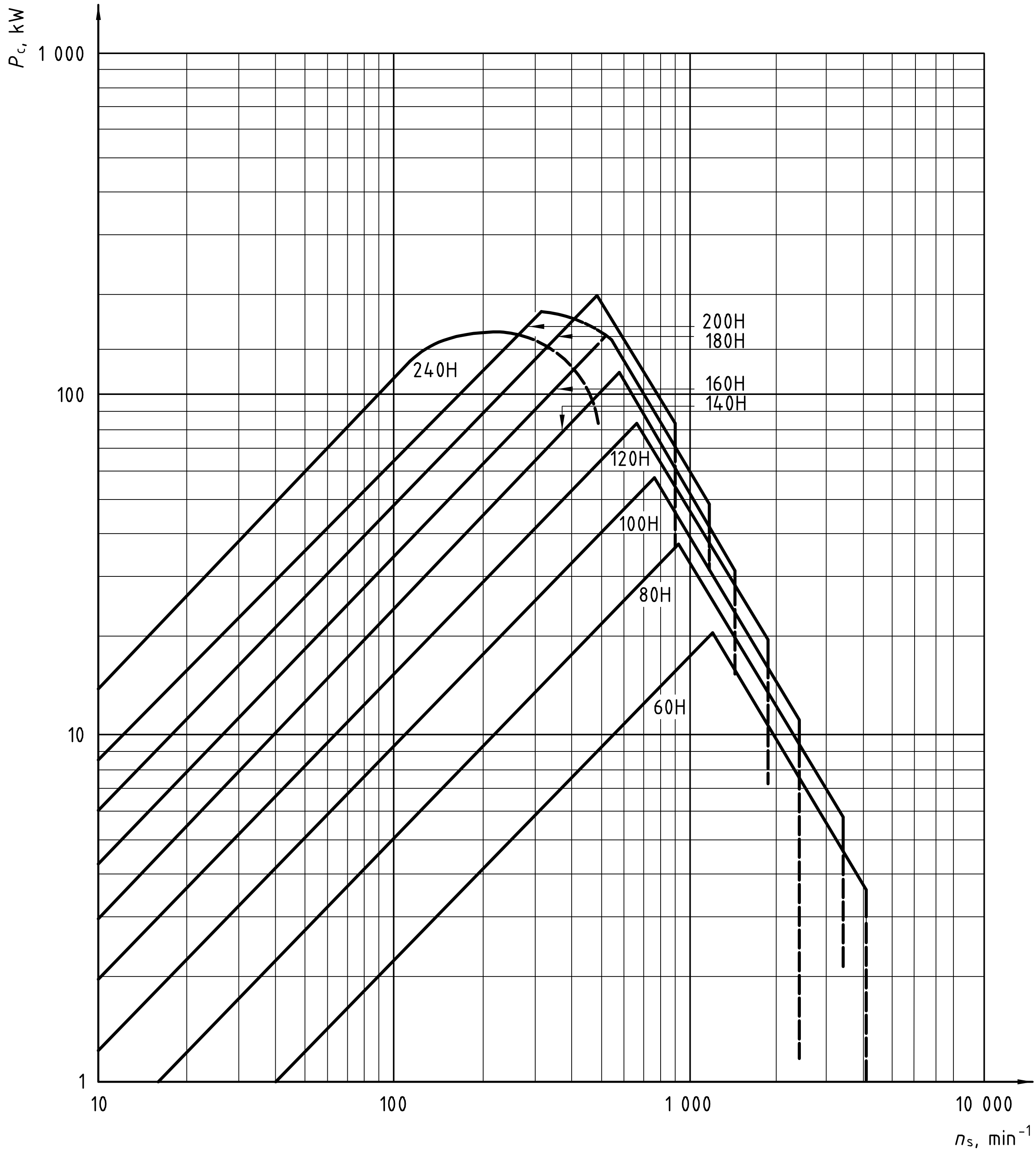
P_c corrected power

n_s small sprocket speed

NOTE 1 The power rating of duplex chain can be calculated by multiplying the value of P_c for simplex chain by 1,7.

NOTE 2 The power rating of triplex chain can be calculated by multiplying the value of P_c for simplex chain by 2,5.

Figure 1 — Typical capacity chart for selection of Type A simplex chains based on a 19-tooth sprocket conforming with ISO 606



Key

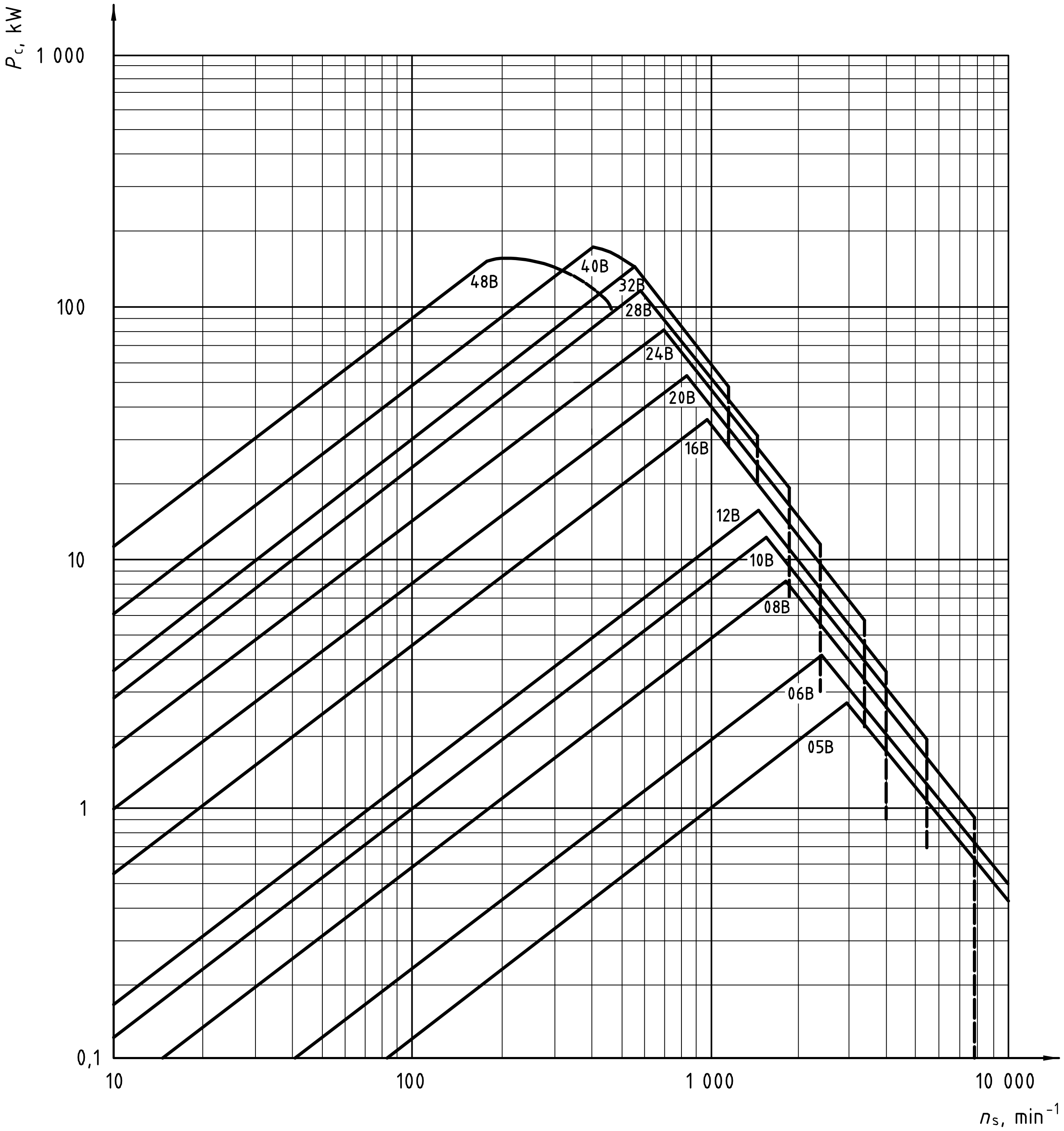
P_c corrected power

n_s small sprocket speed

NOTE 1 The power rating of duplex chain can be calculated by multiplying the value of P_c for simplex chain by 1,7.

NOTE 2 The power rating of triplex chain can be calculated by multiplying the value of P_c for simplex chain by 2,5.

Figure 2 — Typical capacity chart for selection of Type A heavy-series simplex chains based on a 19-tooth sprocket conforming with ISO 606



Key

P_c corrected power

n_s small sprocket speed

NOTE 1 The power rating of duplex chain can be calculated by multiplying the value of P_c for simplex chain by 1,7.

NOTE 2 The power rating of triplex chain can be calculated by multiplying the value of P_c for simplex chain by 2,5.

Figure 3 — Typical capacity chart for selection of Type B simplex chains based on a 19-tooth sprocket conforming with ISO 606

7.2 Correction for other operating conditions for chains

7.2.1 Power correction

If the characteristics of the chain drive and its operating conditions are different from those described in 7.1, the transmitted power shall be corrected by using Equation (2).

The derivation of factors f_1 and f_2 are given in 7.2.2 and 7.2.3.

7.2.2 Application factor f_1

Factor f_1 takes into account dynamic overloads dependant on the chain drive operating conditions and resulting, in particular, from the nature of the driver and driven elements. The value of factor f_1 can be selected directly or by analogy using Table 2 in conjunction with the definitions given in Tables 3 and 4.

Table 2 — Application factor f_1

Driven machine characteristics (see Table 4)	**Driver machine characteristics** (see Table 3)		
	Smooth running	**Slight shocks**	**Moderate shocks**
Smooth running	1,0	1,1	1,3
Moderate shocks	1,4	1,5	1,7
Heavy shocks	1,8	1,9	2,1

Table 3 — Definitions of characteristics of driver machines

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

Table 4 — Definitions of characteristics of driven machines

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

7.2.3 Factor f_2

Factor f_2 takes account of the number of teeth on the small sprocket only for the portion of the power ratings limited by plate fatigue. Its value shall be determined using Equation (5). Values of f_2 for 11 to 45 teeth are shown in Figure 4.

$$f_2 = \left(\frac{19}{z_s} \right)^{1,08} \quad (5)$$

Use the equations in B.3 and B.4 to account for the number of teeth on the small sprocket for the portions of the power ratings limited by roller and bushing impact fatigue and pin–bushing galling.

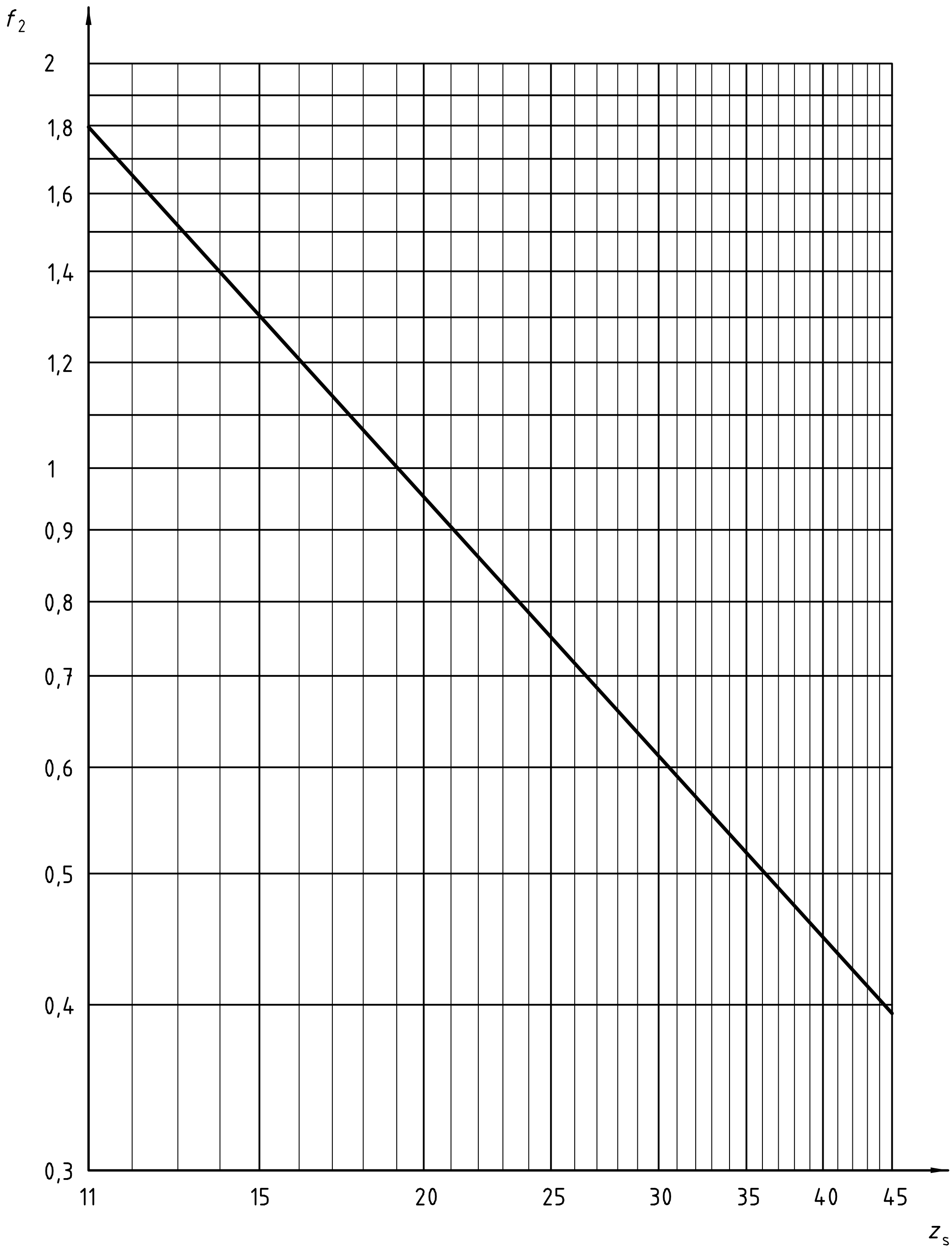


Figure 4 — Factor f_2 allowing for the number of teeth on the small sprocket z_s

7.3 Chain selection

From the chain capacity charts (see Figures 1, 2 and 3), select the smallest pitch of simplex chain that will transmit the required power at the required speed of the small sprocket.

Where the speed exceeds the limit of the smallest pitch simplex chain, or a more compact drive is necessary, a multiplex chain of smaller pitch should be considered. Select multiplex chains from the capacity charts (see Figures 1, 2 and 3) using the factors provided in Note 1 and Note 2 with each chart.

7.4 Chain length

For a drive with two sprockets, having a known chain pitch p and approximate centre distance a_0, calculate the number of chain pitches, X_0, using Equations (6) and (7).

The calculated number of pitches, X_0 should be rounded up to a whole even number, X, to avoid the use of cranked links.

For sprockets with the same number of teeth ($z = z_1 = z_2$):

$$X_0 = 2\frac{a_0}{p} + z \qquad (6)$$

For sprockets with different number of teeth:

$$X_0 = 2\frac{a_0}{p} + \frac{z_1 + z_2}{2} + \frac{f_3 \times p}{a_0} \qquad (7)$$

where factor $f_3 = \left(\frac{|z_2 - z_1|}{2\pi} \right)^2$

Calculated values for f_3 are given in Table 5.

7.5 Chain speed

Calculate the chain speed using the following equation:

$$v = \frac{n_1 \times z_1 \times p}{60\ 000} \qquad (8)$$

Table 5 — Calculated values of factor f_3

$\lvert z_2 - z_1 \rvert$	f_3	$\lvert z_2 - z_1 \rvert$	f_3	$\lvert z_2 - z_1 \rvert$	f_3	$\lvert z_2 - z_1 \rvert$	f_3	$\lvert z_2 - z_1 \rvert$	f_3
1	0,025 3	21	11,171	41	42,580	61	94,254	81	166,191
2	0,101 3	22	12,260	42	44,683	62	97,370	82	170,320
3	0,228 0	23	13,400	43	46,836	63	100,536	83	174,500
4	0,405 3	24	14,590	44	49,040	64	103,753	84	178,730
5	0,633 3	25	15,831	45	51,294	65	107,021	85	183,011
6	0,912	26	17,123	46	53,599	66	110,339	86	187,342
7	1,241	27	18,466	47	55,955	67	113,708	87	191,724
8	1,621	28	19,859	48	58,361	68	117,128	88	196,157
9	2,052	29	21,303	49	60,818	69	120,598	89	200,640
10	2,533	30	22,797	50	63,326	70	124,119	90	205,174
11	3,065	31	24,342	51	65,884	71	127,690	91	209,759
12	3,648	32	25,938	52	68,493	72	131,313	92	214,395
13	4,281	33	27,585	53	71,153	73	134,986	93	219,081
14	4,965	34	29,282	54	73,863	74	138,709	94	223,187
15	5,699	35	31,030	55	76,624	75	142,483	95	228,605
16	6,485	36	32,828	56	79,436	76	146,308	96	233,443
17	7,320	37	34,677	57	82,298	77	150,184	97	238,333
18	8,207	38	36,577	58	85,211	78	154,110	98	243,271
19	9,144	39	38,527	59	88,175	79	158,087	99	248,261
20	10,132	40	40,529	60	91,189	80	162,115	100	253,302

8 Maximum sprocket centre distance

For the number of chain pitches, X, derived in 7.4, determine the maximum distance between centres of the sprockets, a, using Equations (9) or (10).

For two sprockets with the same number of teeth ($z = z_1 = z_2$):

$$a = p\left(\frac{X - z}{2}\right) \tag{9}$$

For two sprockets with different numbers of teeth:

$$a = f_4 p\left[2X - \left(z_1 + z_2\right)\right] \tag{10}$$

Values for factor f_4 are given in Table 6.

Table 6 — Calculated values of factor f_4

$\left\lvert \frac{X - z_s}{z_2 - z_1} \right\rvert$	f_4	$\left\lvert \frac{X - z_s}{z_2 - z_1} \right\rvert$	f_4	$\left\lvert \frac{X - z_s}{z_2 - z_1} \right\rvert$	f_4	$\left\lvert \frac{X - z_s}{z_2 - z_1} \right\rvert$	f_4
13	0,249 91	2,7	0,247 35	1,54	0,237 58	1,26	0,225 20
12	0,249 90	2,6	0,247 08	1,52	0,237 05	1,25	0,224 43
11	0,249 88	2,5	0,246 78	1,50	0,236 48	1,24	0,223 61
10	0,249 86	2,4	0,246 43	1,48	0,235 88	1,23	0,222 75
9	0,249 83	2,3	0,246 02	1,46	0,235 24	1,22	0,221 85
8	0,249 78	2,2	0,245 52	1,44	0,234 55	1,21	0,220 90
7	0,249 70	2,1	0,244 93	1,42	0,233 81	1,20	0,219 90
6	0,249 58	2,0	0,244 21	1,40	0,233 01	1,19	0,218 84
5	0,249 37	1,95	0,243 80	1,39	0,232 59	1,18	0,217 71
4,8	0,249 31	1,90	0,243 33	1,38	0,232 15	1,17	0,216 52
4,6	0,249 25	1,85	0,242 81	1,37	0,231 70	1,16	0,215 26
4,4	0,249 17	1,80	0,242 22	1,36	0,231 23	1,15	0,213 90
4,2	0,249 07	1,75	0,241 56	1,35	0,230 73	1,14	0,212 45
4,0	0,248 96	1,70	0,240 81	1,34	0,230 22	1,13	0,210 90
3,8	0,248 83	1,68	0,240 48	1,33	0,229 68	1,12	0,209 23
3,6	0,248 68	1,66	0,240 13	1,32	0,229 12	1,11	0,207 44
3,4	0,248 49	1,64	0,239 77	1,31	0,228 54	1,10	0,205 49
3,2	0,248 25	1,62	0,239 38	1,30	0,227 93	1,09	0,203 36
3,0	0,247 95	1,60	0,238 97	1,29	0,227 29	1,08	0,201 04
2,9	0,247 78	1,58	0,238 54	1,28	0,226 62	1,07	0,198 48
2,8	0,247 58	1,56	0,238 07	1,27	0,225 93	1,06	0,195 64

9 Lubrication

9.1 Methods of lubrication

The method of lubrication that should be used to ensure satisfactory control of wear in the chain drive is determined by the speed and capacity rating of the chain.

The lubrication ranges, which define the minimum required methods of lubrication to be used, are derived from the chart given in Figure 5. The definitions of the lubrication ranges are as follows.

Range 1: oil supply by means of oil can or brush, applied manually at frequent intervals.

Range 2: drip feed lubrication.

Range 3: oil bath or disc lubrication.

Range 4: forced-feed lubrication with filter and, if necessary, an oil cooler.

NOTE An oil cooler could be necessary if the drive operates in a confined space at high power and speeds.

9.2 Oil viscosity

The viscosity classes of oils that should be used for chain drive lubrication at different operating ambient temperatures are shown in Table 7.

Ensure that the lubricating oil is free from contaminants, particularly abrasive particles.

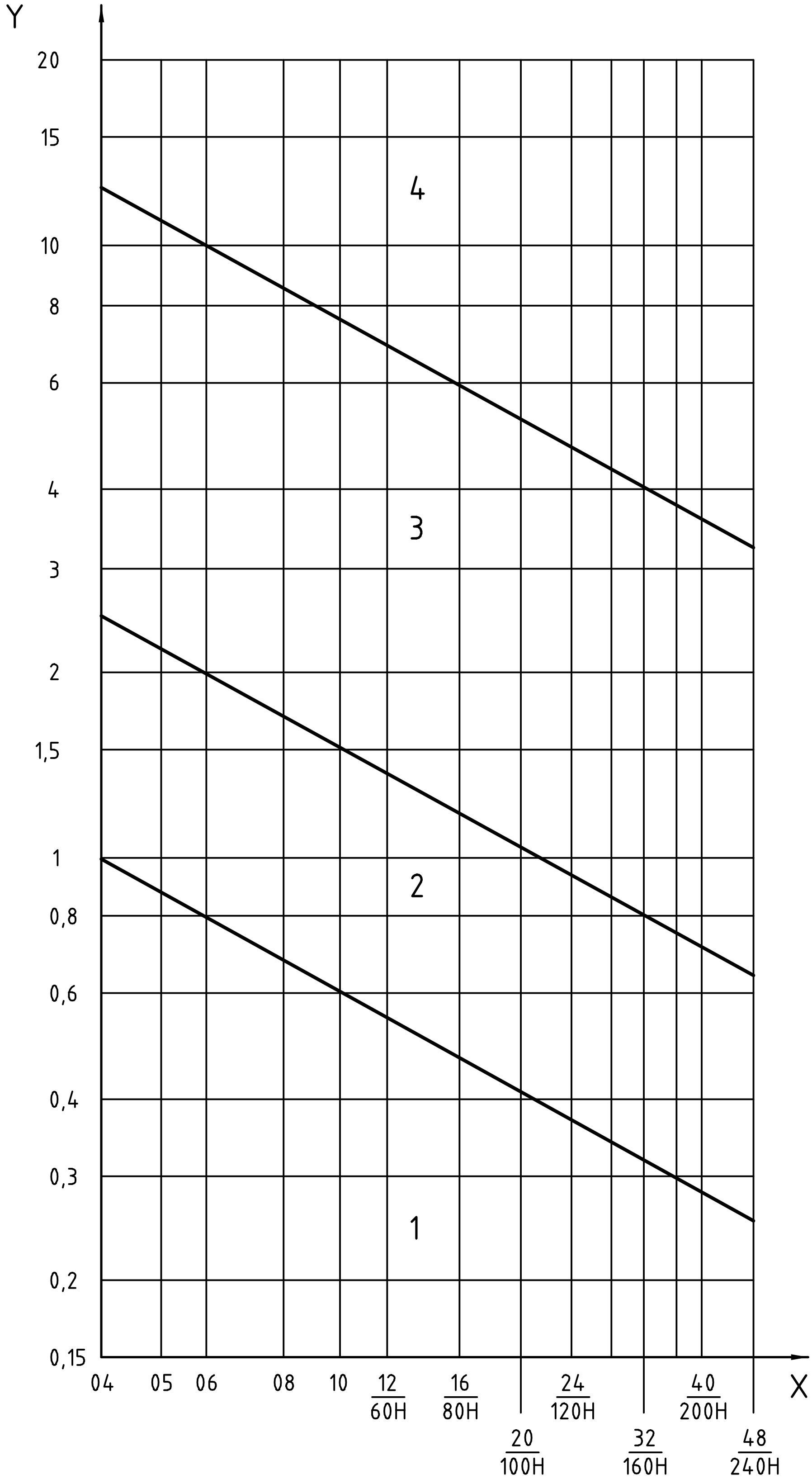
Table 7 — Chain drive lubrication oil viscosity class

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

10 Good practice in drive design

10.1 Sprocket centre distance

The preferred centre distance should measure between 30 times and 50 times the chain pitch. There should be a minimum contact arc of 120° on the small sprocket.



X $\frac{\text{A-series}}{\text{A-heavy}}$ or B-series chain numbers

Y chain speed v, $m \cdot s^{-1}$

For 1, 2, 3 and 4, see 9.1.

Figure 5 — Lubrication ranges selection chart

10.2 Chain adjustment

The recommended method of chain adjustment is by correction of the centre distance.

Rotate the sprockets away from each other to make one span taut. Then measure the total mid-span movement, A–C, in the slack span (see Figure 6).

When the centres are inclined less than 45° from the horizontal, the amount of movement, A–C, should be from 2 % (± 1 %) to 6 % (± 3 %) of the centre distance.

When the centres are inclined more than 45° from the horizontal, the amount of movement, A–C, should be from 1 % (± 0,5 %) to 3 % (± 1,5 %) of the centre distance.

10.3 Idlers

Alternatively, adjustment of the chain can be achieved by means of idlers, idler sprockets or other suitable means, especially in the case of a chain drive that has an inclination of more than 60° to the horizontal.

Care shall be taken to ensure that additional forces are not applied to the chain.

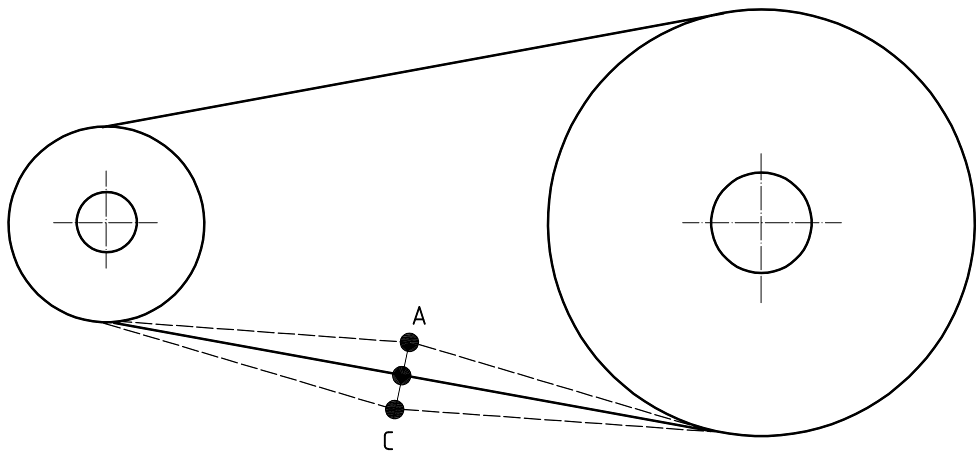
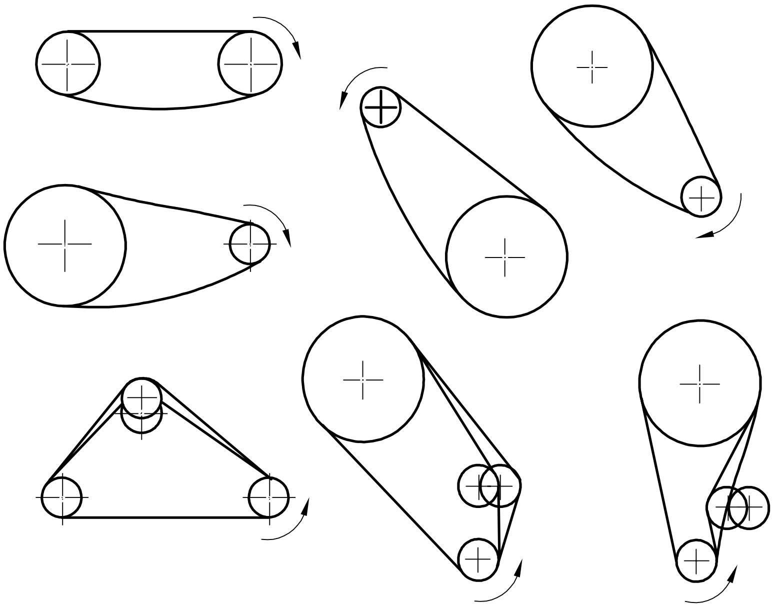


Figure 6 — Chain slack adjustment

10.4 Drive layout

Drive arrangements that normally provide good function and life are shown in Figure 7.



NOTE For drive arrangements not shown, consult a chain manufacturer.

Figure 7 — Commonly used drive arrangements

Annex A
(informative)

Example of chain drive selection

A.1 Given parameters

The layout of the chain drive to which the example refers is shown diagrammatically in Figure A.1. The parameters are as follows.

Power transmitted:	$P = 1{,}40$ kW
Input speed:	$n_1 = 100$ min^{-1}
Output speed:	$n_2 = 34$ min^{-1}
Speed ratio:	$i = n_1/n_2 = 2{,}94$
Driving machine:	geared electric motor
Driven machine:	conveyor, non-uniformly loaded
Approximate centre distance:	$a_0 = 850$ mm

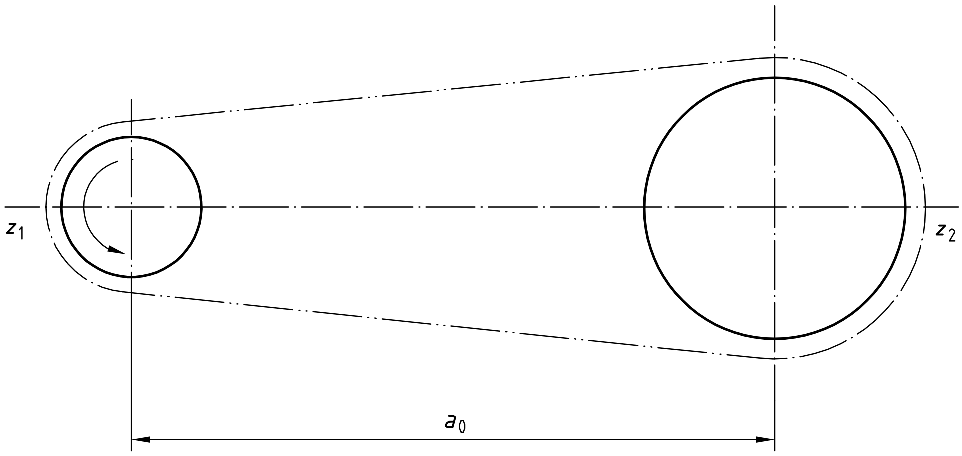


Figure A.1 — Chain drive layout

A.2 Sprocket selection

Selected number of teeth for driving sprocket

$z_1 = 17$

Number of teeth on driven sprocket

$z_2 = i \times z_1 = 2{,}94 \times 17 = 50$ [from Equation (4)]

A.3 Chain calculations and selection

A.3.1 Power correction

Application factor: f_1 = 1,4 (derived from Table 2)

Sprocket factor: f_2 = 1,13 [derived from Equation (5) and Figure 4]

Corrected power: $P_c = P \times f_1 \times f_2$ [from Equation (2)]

$= 1,4 \times 1,4 \times 1,13$

$= 2,21$ kW

A.3.2 Chain selection

Applying P_c = 2,21 kW and n_1 = 100 min^{-1} to the chain drive capacity charts in Figures 1, 2 and 3, select roller chain 16A – 1, 60H – 1, or 16B – 1.

Chain pitch, p, is 25,4 mm for 16A and 16B chains, and 19,05 mm for 60H chain (in conformance with ISO 606).

A.3.3 Chain length

Calculated number of links

$$X_0 = 2\frac{a_0}{p} + \frac{z_1 + z_2}{2} + \frac{f_3 \times p}{a_0} \text{ [from Equation (7)]}$$

where f_3 = 27,585

when $|z_2 - z_1| = |50 - 17| = 33$ (from Table 5)

therefore,

— for 16A and 16B chains:

$$X_0 = \frac{2 \times 850}{25,4} + \frac{17 + 50}{2} + \frac{27,585 \times 25,4}{850} = 101,25 \text{ pitches}$$

and

the selected number of links: X = 102 pitches (i.e. next higher even number).

— for 60H chain:

$$X_0 = \frac{2 \times 850}{19,05} + \frac{17 + 50}{2} + \frac{27,585 \times 19,05}{850} = 123,36 \text{ pitches}$$

and

the selected number of links: X = 124 pitches (i.e. next higher even number).

A.3.4 Chain speed

$$v = \frac{n_1 \times z_1 \times p}{60\ 000} \text{ [from Equation (8)]}$$

$$v = \frac{100 \times 17 \times 25{,}4}{60\ 000} = 0{,}72 \text{ m·s}^{-1} \text{ for 16A and 16B chains, and}$$

$$v = \frac{100 \times 17 \times 19{,}05}{60\ 000} = 0{,}54 \text{ m·s}^{-1} \text{ for 60H chain}$$

A.4 Maximum sprocket centre distance

Maximum centre distance:

$$a = f_4\, p \left[2X - \left(z_1 + z_2\right)\right] \text{ [from Equation (10)]}$$

where f_4 = 0,247 00 for 16A and 16B chains, when

$$\frac{X - z_s}{\left|z_2 - z_1\right|} = \frac{102 - 17}{\left|50 - 17\right|} = 2{,}576 \text{ (interpolated from Table 6)}$$

and f_4 = 0,248 30 for 60H chain, when

$$\frac{X - z_s}{\left|z_2 - z_1\right|} = \frac{124 - 17}{\left|50 - 17\right|} = 3{,}242 \text{ (interpolated from Table 6)}$$

therefore,

a = (0,247 00 × 25,4) × [(2 × 102) – (17 + 50)] = 859,5 mm for 16A and 16B chains, and

a = (0,248 30 × 19,05) × [(2 × 124) – (17 + 50)] = 856,15 mm for 60H chain

A.5 Lubrication

Applying v = 0,72 m·s^{-1} with chain 16A – 1 or 16B – 1 to the lubrication range chart given in Figure 5, select lubrication Range 2, which requires the minimum method of oil supply to be by means of drip feed lubrication (see 9.1).

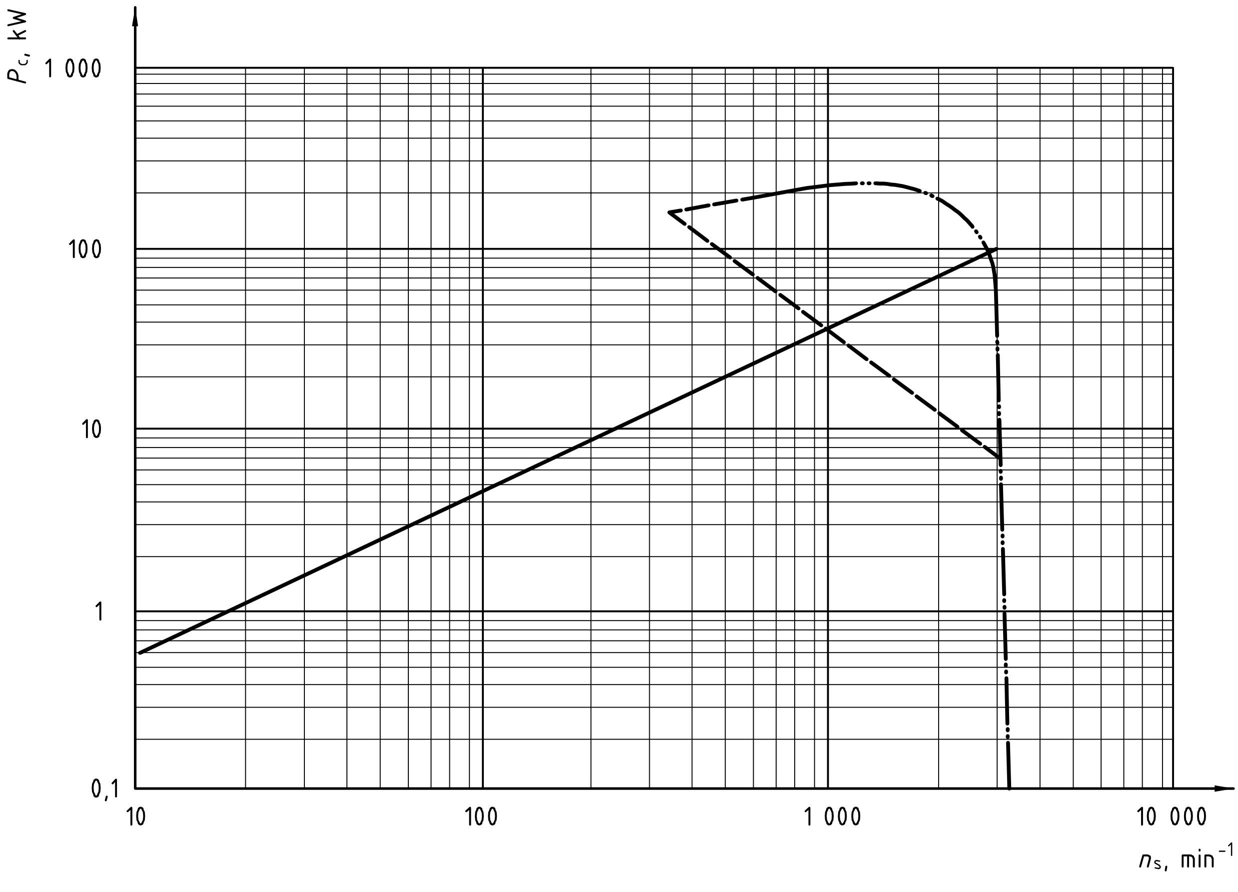
Applying v = 0,54 m·s^{-1} with chain 60H – 1 to the lubrication range chart given in Figure 5, select lubrication Range 2. This requires the minimum method of oil supply to be by means of drip feed lubrication (see 9.1).

Annex B
(informative)

Power rating equations

B.1 Power rating graph

The power ratings for 16A and 16B chains are shown in Figure B.1. The power limited by plate fatigue is represented by the solid line extending from approximately 0,6 kW at 10 min^{-1} to 100 kW at 3 000 min^{-1}. The power limited by roller-bushing impact fatigue is represented by the dashed line extending from approximately 160 kW at 350 min^{-1} to 6,5 kW at 3 000 min^{-1}. The power limited by pin–bush galling is represented by the dot-dashed line extending from approximately 150 kW at 350 min^{-1} to slightly more than 200 kW at 1 500 min^{-1} to 0,1 kW at 3 300 min^{-1}. The power rating for the chain, at a specified speed, is the least of these three at the specified speed.



Key

P_c corrected power

n_s small sprocket speed

Figure B.1 — Roller chain power rating elements for a 19-tooth sprocket

B.2 Equations for power ratings limited by plate fatigue

For A-series chains:

$$P_c = \frac{z_s^{1,08} \times n_s^{0,9} \times 99 A_i\, p^{(1,0-0,000\,8p)}}{6 \times 10^7} \text{ kW}$$

where A_i is the sectional area of two inner plates

$$A_i = 0{,}118\, p^2 \text{ mm}^2$$

For 085 chain:

$$P_c = \frac{z_s^{1,08} \times n_s^{0,9} \times 86{,}2 A_i\, p^{(1,0-0,000\,8p)}}{6 \times 10^7} \text{ kW}$$

where A_i is the sectional area of two inner plates

$$A_i = 0{,}0745 p^2 \text{ mm}^2$$

For A-series heavy chains:

$$P_c = \frac{z_s^{1,08} \times n_s^{0,9} \left(t_H / t_S\right)^{0,5} 99 A_i\, p^{(1,0-0,000\,8p)}}{6 \times 10^7} \text{ kW}$$

where

A_i is the sectional area of two standard inner plates

$$A_i = 0{,}118 p^2 \text{ mm}^2$$

t_H is the thickness of heavy series inner plate, in millimetres;

t_S is the thickness of standard series inner plate, in millimetres.

For B-series chains:

$$P_c = \frac{z_s^{1,08} \times n_s^{0,9} \times 99 A_i\, p^{(1,0-0,000\,9p)}}{6 \times 10^7} \text{ kW}$$

where

A_i is the sectional area of two standard inner plates

$$A_i = 2\, t_i\, (0{,}99 h_2 - d_b) \text{ mm}^2$$

d_b is the estimated bushing diameter, in millimetres

$$d_b = d_2 \left(\frac{d_1}{d_2}\right)^{0,475}$$

t_i is the estimated thickness of inner plates, in millimetres

$$t_i = \frac{b_2 - b_1}{2,11}$$

and

b_1 is the minimum width between inner plates, in millimetres;

b_2 is the maximum width over inner link, in millimetres;

d_1 is the maximum roller diameter, in millimetres;

d_2 is the maximum pin diameter, in millimetres;

h_2 is the maximum inner plate depth, in millimetres;

p is the chain pitch, in millimetres.

B.3 Equations for power limited by roller and bush impact fatigue

For A-series, A-series heavy and B-series chains, except 04C, 06C and 085:

$$P_c = \frac{953,5 z_s^{1,5} \times p^{0,8}}{n_s^{1,5}} \text{ kW}$$

For 04C and 06C chains:

$$P_c = \frac{1\,626,6 z_s^{1,5} \times p^{0,8}}{n_s^{1,5}} \text{ kW}$$

For 085 chain:

$$P_c = \frac{190,7 z_s^{1,5} \times p^{0,8}}{n_s^{1,5}} \text{ kW}$$

B.4 Equation for power limited by pin–bush galling

For A-series, A-series heavy and B-series chain:

$$P_c = \frac{z_s n_s p}{3\,780 K_{PS}} \left[4,413 - 2,073 \left(\frac{p}{25,4} \right) - 0,027\,4 z_s - \ln \left(\frac{n_s}{1\,000 K_{PS}} \right) \times \left\{ 1,59 \times \lg \left(\frac{p}{25,4} \right) + 1,873 \right\} \right] \text{ kW}$$

where K_{PS} is the speed modification factor, according to Table B.1.

Table B.1 — Speed modification factor

Chain pitch mm	K_{PS}
$\leqslant$ 19,05	1,0
25,40 to 31,75	1,25
38,10	1,30
44,45	1,35
50,80 to 57,15	1,40
63,50	1,45
76,20	1,50

B.5 Equations for lubrication speed limits

Maximum speed for Type 1 lubrication:

$$v = 2{,}8p^{-0{,}56} \text{ m}\cdot\text{s}^{-1}$$

Maximum speed for Type 2 lubrication:

$$v = 7{,}0p^{-0{,}56} \text{ m}\cdot\text{s}^{-1}$$

Maximum speed for Type 3 lubrication:

$$v = 35p^{-0{,}56} \text{ m}\cdot\text{s}^{-1}$$

ICS 21.220.30

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