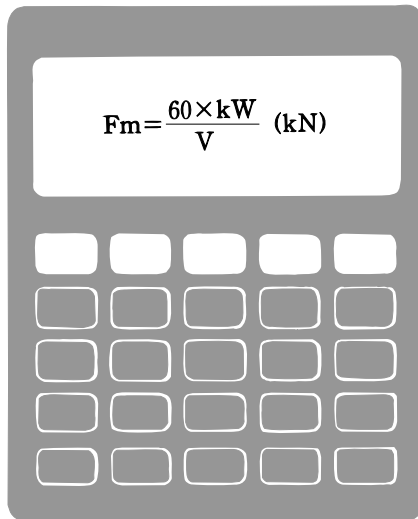
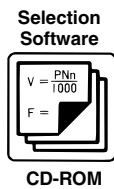


A Drive Chain Selection Program offering "General Selection" and "Slow-Speed Selection" of 6 chain types (BS/DIN, ANSI 80th, LAMBDA, SUPER SERIES, DP and WP) is available on request.



ROLLER CHAIN SELECTION

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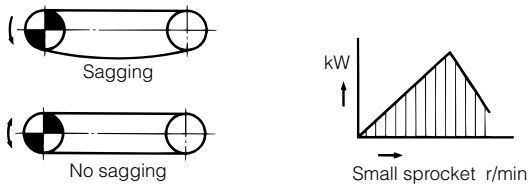
⚠ When there are regulations by law or guidelines governing the selection of a chain, please follow both of these as well as the selection methods mentioned in this catalog, and then select the chain with the most leeway.

Roller Chain Selection

1. Selection Guide

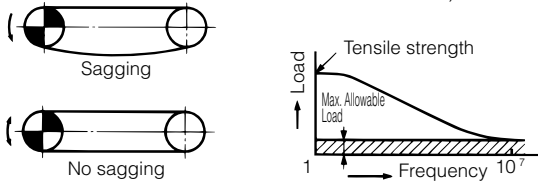
Application — Essential points for selection — **Selection method**

Ordinary transmission — Selection based on kW ratings table — **General selection**



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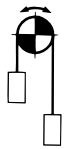
Ordinary transmission — Selection based on Max. Allowable Load (economical selection, chain speed $v = 50$ m/min) — **Slow speed selection**



Starting frequency - more than 5 times/day (8hrs)
Page 81

Slow speed selection (special)
Starting frequency - more than 6 times/day (8hrs)
Page 83

Lifting Application — Selection based on Max. Allowable Load (chain speed $V = 50$ m/min) Please use F-CLs or exclusive CLs for end-bolts — **Lifting roller chain selection**



Page 84

Chain type	Connecting parts that can be used			
	M-CL	F-CL	2-pitch OL	1-pitch OL
RS	○	○	○	□
SUPER	—	○	—	—
RSD-A	○	—	—	□
RSD-A-NP	○	—	—	□
RSDX-A	○	—	—	—
RS-KT	△	○	—	□
RS-SN	○	○	○	□
RS	○	○	○	△
SUPER	—	○	—	—
RS-HT	—	○	—	—
SUPER-H	—	○	—	—
Ultra-Super	—	○	—	—
NP	○	○	—	△
WP	○	○	—	△
DP	○	○	—	△
SS, AS, LS	○	—	—	○
PC	○	—	—	—
PC-SY	○	—	—	—
NS	○	—	—	○
TI	○	—	—	○
RS-KT	△	○	—	△
RS-SN	○	○	○	△

CL : abbrev. of Connecting Link
OL : abbrev. of Offset Link

○ : Available □ : Allow for percentage decline in kW ratings (Refer to each kW ratings table)
△ : Allow for percentage decline in strength (Refer Pgs. 81 - 83) — : Unavailable X : Not applicable Dotted line : Made-to-order

Data required for roller chain selection

- 1) Driven machine
- 2) Load classification
- 3) Source of power
- 4) kW to be transmitted
- 5) Diameter and RPM of driving shaft
- 6) Diameter and RPM of driven shaft
- 7) Center distance between shafts

Necessary power (motor) characteristics for the special method of chain selection

- 1) Moment of inertia
- 2) Rated torque
- 3) Starting torque
- 4) Stalling torque

2. SERVICE FACTORS

Tsubaki offers simplex, duplex and triplex chains in RF06B to RS40B of BS/DIN European standard. In ANSI American standard, up to 6 strands are available as standard items from RS40 to RS240 and up to triplex for RS25 and RS35. In multiple strand chain drives, the load is unequal across the width of the chain, so the transmission capability of multiple strand chain is calculated using multiple strand factors shown in the table below.

Table 1 : Multi-strand factor

No. of strands	Multi-strand factor
2 strands	1.7
3 strands	2.5
4 strands	3.3
5 strands	3.9
6 strands	4.6

Service factor K_s

The chain's transmission capacity is affected if there is frequent load fluctuation. The appropriate service factor K_s must be applied based on the source of power and type of machine as shown in the table below.

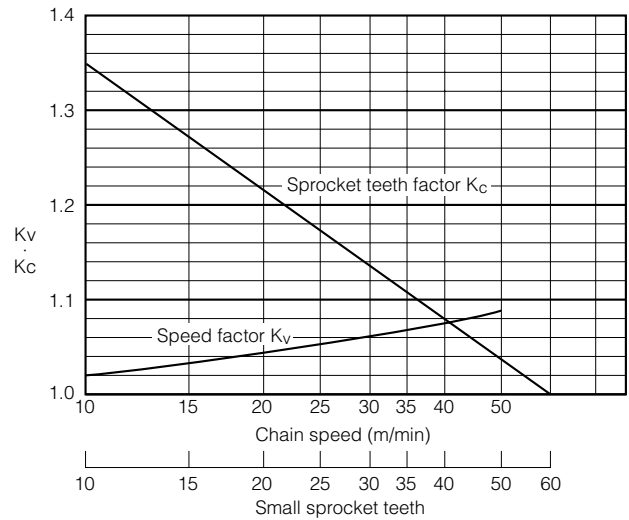
Please note that the service factor is never smaller than 1.0.

Table 2 : Service factor K_s

Type of Impact	Machines	Source of Power		
		Electric Motor or Turbine	Internal Combustion Engine With hydraulic drive	Without hydraulic drive
Smooth	Belt Conveyors with small load fluctuation, chain conveyors, centrifugal blowers, ordinary textile machines, ordinary machines with small load fluctuation.	1.0	1.0	1.2
Some impact	Centrifugal compressors, marine engines, conveyors with some load fluctuation, automatic furnaces, dryers, pulverizers, general machine tools, compressors, general work machines, general paper mills.	1.3	1.2	1.4
Large impact	Press, construction or mining machines, vibration machines, oil well rigs rubber mixers, rolls, roll gangs, general machines with reverse or large impact loads.	1.5	1.4	1.7

Speed factor K_v and sprocket teeth factor K_c

Table 3 : Speed factor, K_v and sprocket teeth factor K_c



Shock factor K

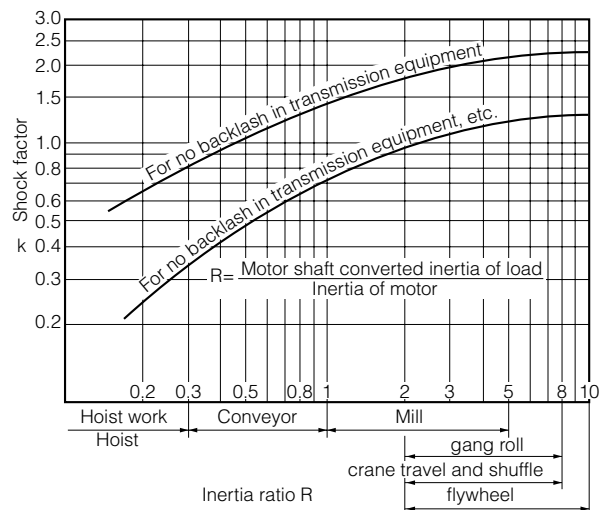
This coefficient is determined by the rate of inertia between the prime mover and the driven machinery (rate of I, GD^2) as well as the amount of backlash in the transmission equipment.

When rate of inertia $R > 10, R = 10$

When rate of inertia $R < 0.2, R = 0.2$

When I or GD^2 for either the prime mover or driven machinery is unknown, use the value of R on table 4.

Table 4 : Shock factor K



Imbalance load factor K_u

When carrying out shuttle traction and lifting with two chains, or four chains for shuttle drive and lifting, the chain tension is not uniform. This must be accounted for by multiplying the following imbalance load coefficient K_u to adjust the left-and-right load imbalance.

Example : For four lifting strands, the imbalance load factor for one strand $K_u = 0.6 \times 0.6 = 0.36$

Table 5 : Imbalance load factor K_u

2 lifting strands	0.6
4 lifting strands	0.36

3. Roller Chain Provisional Selection Tables

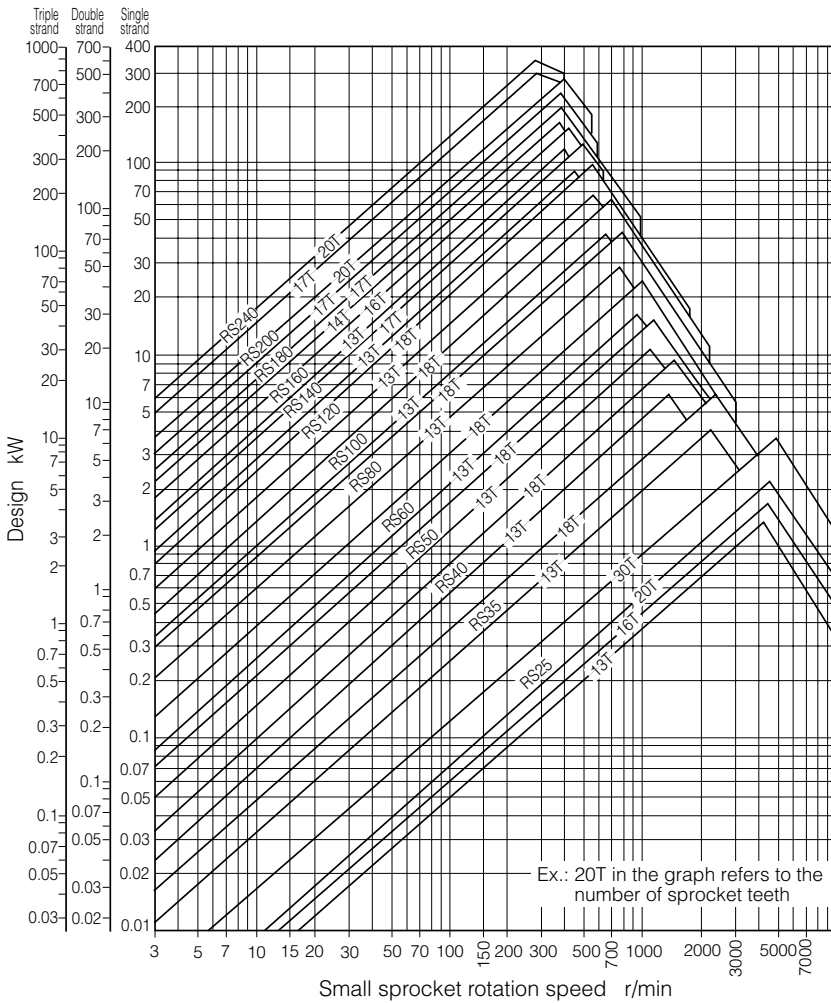


Table 6: Provisional selection chart for RS Roller Chain (Lambda Roller Chain)

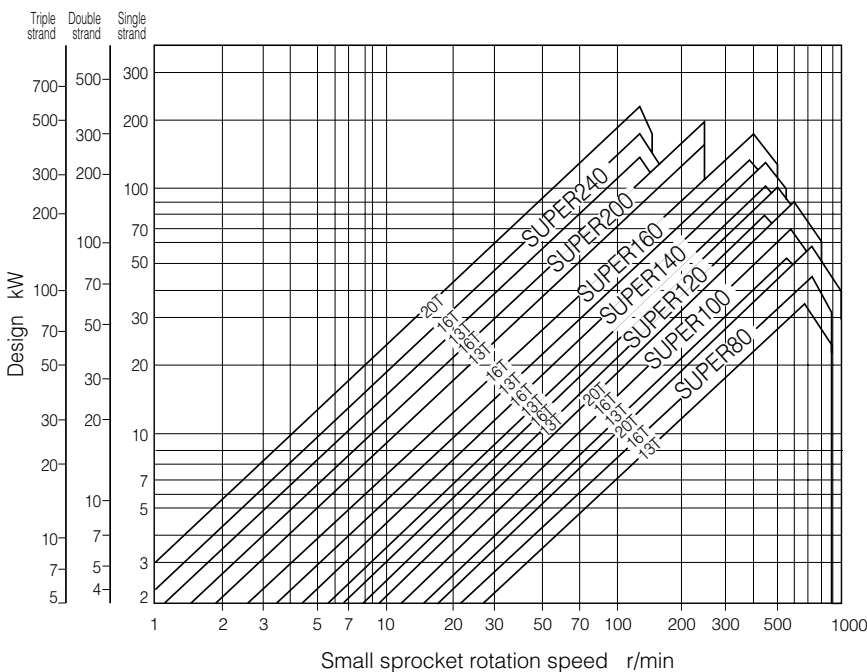


Table 7: Provisional selection chart for SUPER Roller Chain

How to use this table (Table 6)

1. Example: Single strand chain, design kW = 5 kW

(1) Assume that the speed of the small sprocket is 100 r/min. Judging from the intersecting point of design kW value of 5 kW (vertical axis) and the speed value of 100 r/min (horizontal axis), RS80 and a sprocket with between 13 and 18 teeth would be appropriate. Therefore, based on the position of the intersection, we can see that a 14T sprocket can be used.

(2) Assume that the speed of the small sprocket is 300 r/min. Following the same procedure shown in the above example, RS60 and a sprocket with less than 13 teeth or RS50 and a sprocket with more than 18 teeth would be appropriate. This table is used for tentative selections only. The kW ratings tables should be used to confirm the chain sizes.

(3) When the chain speed is less than 50 m/min., it is more economical to select your RS Roller Chain by slow speed selection.

(4) Please allow for a 20% drop in the kW rating values shown in the design kW ratings chart (Table 6) when 1-pitch offset links are used.

(5) A 4-pitch offset link is available for SUPER Roller Chain and the kW ratings are the same as in Table 7.

4. SELECTION FORMULAE

4-1 Symbols and units used in formulae (Table 8)

Symbol	Definition	SI unit	Gravitational unit
C	Center distance in pitches	—	—
C'	Center distance between shafts	m	m
d	Pitch circle diameter of the small sprocket	mm	mm
D	Outer diameter of the drum	mm	mm
F _b	Chain tension when the prime mover is decelerating (stalling)	kN	kgf
F' _b	Design chain tension when the prime mover is decelerating (stalling)	kN	kgf
F _c	Chain tension of shuttle drive	kN	kgf
F' _c	Design chain tension of shuttle drive	kN	kgf
F _ℓ	Chain tension from torque on load side (actual load)	kN	kgf
F' _ℓ	Design chain tension from torque on load side (actual load)	kN	kgf
F _m	Chain tension from prime mover rated output	kN	kgf
F' _m	Design chain tension from prime mover rated output	kN	kgf
F _{ms}	Chain tension from starting torque of prime mover	kN	kgf
F' _{ms}	Design chain tension from starting torque of prime mover	kN	kgf
F _{mb}	Chain tension from stalling torque of prime mover	kN	kgf
F' _{mb}	Design chain tension from stalling torque of prime mover	kN	kgf
F _s	Chain tension when prime mover accelerates (starting)	kN	kgf
F' _s	Design chain tension when prime mover accelerates (starting)	kN	kgf
F _w	Chain tension from load (actual load)	kN	kgf
F' _w	Design chain tension from load (actual load)	kN	kgf
f ₁	Coefficient of friction between roller and rail (with lubrication 0.14, without lubrication 0.21)	—	—
<u>G</u>	Standard acceleration from gravity $\underline{G} = 9.80665 \text{ m/s}^2$	—	—
i	Speed ratio (example) if ratio is 1/30 then i = 30	—	—
I _ℓ {GD ² ℓ}	Converted moment of inertia of the loaded prime mover output shaft	kg·m ²	kgf·m ²
I _m {GD ² m}	Moment of inertia of the prime mover output shaft	kg·m ²	kgf·m ²
K	Shock factor	—	—
K _c	Sprocket teeth factor	Refer Table 3	—
K _s	Service factor	Refer Table 2	—
K _t	Temperature coefficient	Refer Table 10	—
K _u	Imbalance load factor	Refer Table 5	—
K _v	Speed factor	Refer Table 3	—
L	Chain length (number of links)	—	—
m	Unit mass of chain	kg/m	kgf/m
M {W}	Mass of load (weight)	kg	kgf
μ	Coefficient of friction between the rail and the axle = 0.1 (shuttle drive) Coefficient of friction between the rotating body and the support rollers = 0.3 (pin gear)	—	—
n	RPM of the small sprocket	r/min	rpm
n ₁	RPM of driver shaft	r/min	rpm
n ₂	RPM of driven shaft	r/min	rpm
N	No. of teeth for large sprocket	—	—
N'	No. of teeth for small sprocket	—	—
P	Chain pitch	mm	mm
R	Inertia ratio	—	—
S	Attachment height for RS attachment chain (distance from the drum surface to the chain pitch center)	mm	mm
t _b	The time for deceleration of the prime mover (when stalling)	s	s
t _s	The time for acceleration of the prime mover (when starting)	s	s
T _b	Stalling torque of the prime mover	%(kN·m)	%(kgf·m)
T _s	Starting torque of the prime mover	%(kN·m)	%(kgf·m)
T _ℓ	Load torque	kN·m	kgf·m
T _m	Working torque	kN·m	kgf·m
T _n	Rated torque of the prime mover	kN·m	kgf·m
V	Chain speed	m/min	m/min

SELECTION

4-2 Formulae (Table 9)

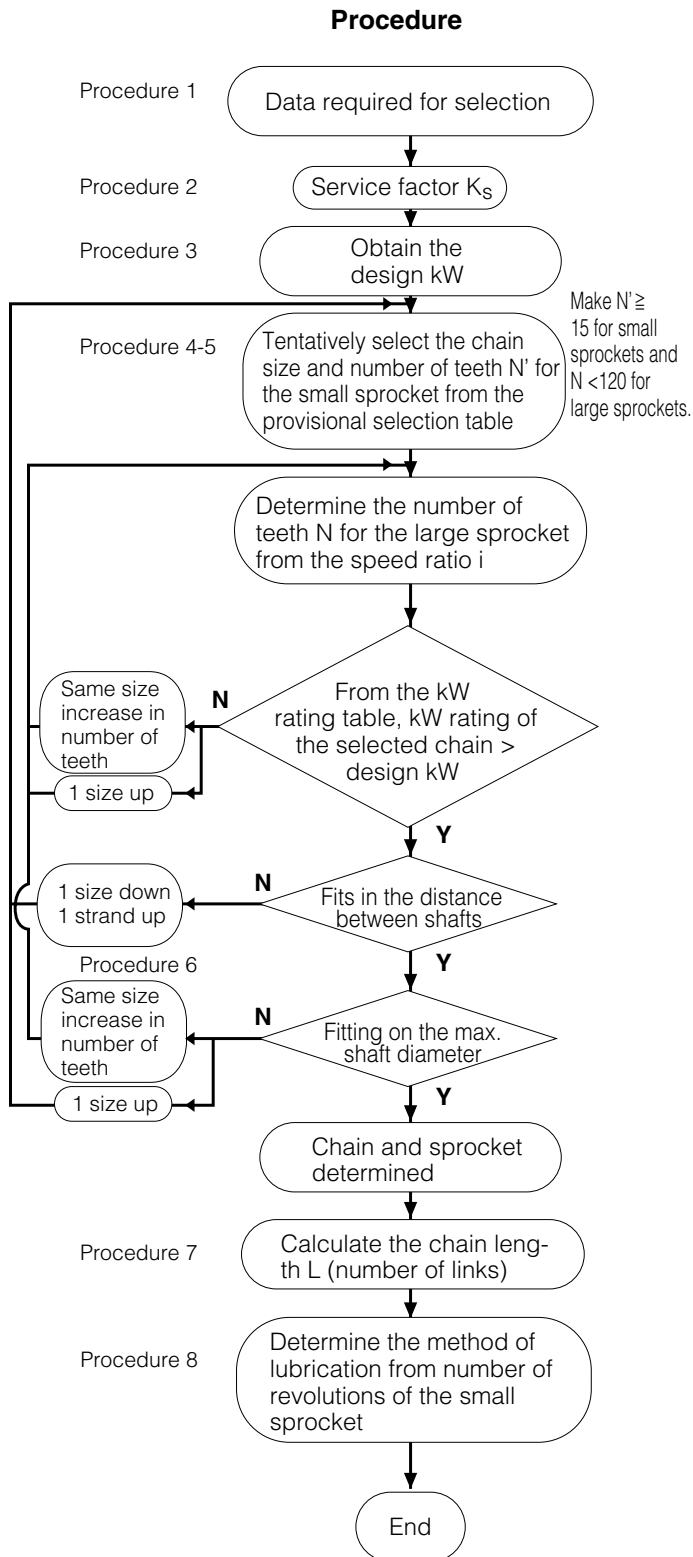
- 1) Perform all selections by taking the transmission efficiency including the chain as $\eta = 1$
 2) Use the calculated value in items 11 and 12 from this table for the tension and transmission kW used in the selection.

Item	SI unit	Gravitational unit
1. Chain length (number of links): L, ordinary transmission	For ordinary transmission between two shafts (1) Where the number of teeth and distance between shafts has been decided for both sprockets. $L = \frac{N+N'}{2} + 2C + \frac{\left(\frac{N-N'}{6.28}\right)^2}{C}$ (2) Where the number of links of chain and number of teeth has been decided. $C = \frac{1}{8} \left\{ 2L - N - N' + \sqrt{(2L - N - N')^2 - \frac{8}{9.86} (N - N')^2} \right\}$ Even if the fractional part of the value found for L (below that of the decimal point) is small, round it up to the nearest integer and add a link. An offset link must be used when an odd number of links exist; however, if possible, change the number of teeth on the sprocket or the distance between shafts so that an even number of links may be used.	
2. Chain speed: V	$V = \frac{P \times N' \times n}{1000}$ (m/min)	
3. Chain tension from prime mover rated output = F_m	$F_m = \frac{60 \times kW}{V}$ (kN)	$F_m = \frac{6120 \times kW}{V}$ (kgf)
4. Inertia where the motor shaft converts the moment of inertia of the load I(GD ²): I_ℓ (GD ² ℓ)	$I_\ell = M \times \left(\frac{V}{2\pi n_1}\right)^2$ (kg·m ²)	$GD^2_\ell = W \times \left(\frac{V}{\pi n_1}\right)^2$ (kgf·m ²)
5. Prime mover rated torque: T_n	$T_n = 9.55 \times \frac{kW}{n_1}$ (kN·m)	$T_n = 974 \times \frac{kW}{n_1}$ (kgf·m)
6. Load torque: T_ℓ	Lifting $T_\ell = \frac{M \times d}{2 \times 1000 \times i} \times \frac{G}{1000}$ (kN·m) Shuttle traction $T_\ell = F'_c \times \frac{1}{2 \times 1000 \times i}$ (kN·m)	$T_\ell = \frac{W \times d}{2 \times 1000 \times i}$ (kgf·m)
7. Working torque: T_m	$T_m = \frac{T_s(\%) + T_b(\%)}{2 \times 100} \times T_n$ (kN·m) OR $T_m = \frac{T_s(\text{kN·m}) + T_b(\text{kN·m})}{2}$ (kN·m)	$T_m = \frac{T_s(\%) + T_b(\%)}{2 \times 100} \times T_n$ (kgf·m) OR $T_m = \frac{T_s(\text{kgf·m}) + T_b(\text{kgf·m})}{2}$ (kgf·m)
8. Chain tension from starting torque: F_{ms} Chain tension from stalling torque: F_{mb}	$F_{ms} = \frac{T_s(\%) \times i}{\{d / (2 \times 1000)\} \times 100} \times T_n \times 1$ (kN) OR $F_{ms} = \frac{T_s(\text{kN·m}) \times i}{d / (2 \times 1000)} \times 1$ (kN) $F_{mb} = \frac{T_b(\%) \times i}{\{d / (2 \times 1000)\} \times 100} \times T_n \times 1.2^*$ (kN) OR $F_{mb} = \frac{T_b(\text{kN·m}) \times i}{d / (2 \times 1000)} \times 1.2^*$ (kN) * constant	Same as left (kgf·m)

Item	SI unit	Gravitational unit
9. Chain tension when the prime mover accelerates: F_s Chain tension when the prime mover decelerates: F_b	$F_s = \frac{M \times V}{t_s \times 60 \times 1000} + F_w^* \text{ (kN)}$ $F_b = \frac{M \times V}{t_b \times 60 \times 1000} + F_w^* \text{ (kN)}$	$F_s = \frac{W \times V}{t_s \times 60 \times G} + F_w^* \text{ (kgf)}$ $F_b = \frac{W \times V}{t_b \times 60 \times G} + F_w^* \text{ (kgf)}$
* For shuttle traction F_w becomes F_c		
10. Shuttle traction chain tension: F_c	$F_c = (M \times \mu + 2.1 \times m \times C' \times f_1) \times \frac{G}{1000} \text{ (kN)}$	$F_c = W \times \mu + 2.1 \times m \times C' \times f_1 \text{ (kgf)}$
11. Design kW (for general selection)	Design kW = Prime mover rated kW $\times K_s$ (kW)	
12. Design chain tension from the load torque: F'_ℓ Design chain tension from the prime mover: F'_m Design chain tension from the starting torque: F'_{ms} Design chain tension from the stalling torque: F'_{mb} Design chain tension of the shuttle drive: F'_c Design chain tension when accelerating: F'_s Design chain tension when decelerating: F'_b Design chain tension from the load: F'_w	$F'_\ell = F_\ell \times K_s \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_m = F_m \times K_s \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_{ms} = F_{ms} \times K \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_{mb} = F_{mb} \times K \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_c = F_c \times K_s \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_s = F_s \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_b = F_b \times K_v \times K_c \text{ { kN (kgf) }}$ $F'_w = M \times K_s \times K_v \times K_c \times \frac{G}{1000} \text{ (kN)}$	$F'_w = W \text{ (or } F_w) \times K_s \times K_v \times K_c \text{ (kgf)}$
When the mass M (weight W) is unknown, find the shaft torque $T = T_n \times i$, { kN·m (kgf·m) } from the rated torque T_n of the prime mover and use $F = 2T/d$ instead of W.		
13. Acceleration time of the prime mover: t_s	$t_s = \frac{(I_m + I_\ell) \times n_1}{375 \times (T_m - T_\ell)} \times \frac{4 \times G}{1000} \text{ (s)}$	$t_s = \frac{(GD^2_m + GD^2_\ell) \times n_1}{375 \times (T_m - T_\ell)} \text{ (s)}$
14. Deceleration time of the prime mover: t_b	$t_b = \frac{(I_m + I_\ell) \times n_1}{375 \times (T_m + T_\ell)} \times \frac{4 \times G}{1000} \text{ (s)}$	$t_b = \frac{(GD^2_m + GD^2_\ell) \times n_1}{375 \times (T_m + T_\ell)} \text{ (s)}$
15. Inertia ratio: R	$R = \frac{I_\ell}{I_m}$	$R = \frac{GD^2_\ell}{GD^2_m}$
16. Conversion of the flywheel effect (GD^2) to the moment of inertia (I)	$1 \text{ kg} \cdot \text{m}^2 \cdots (I)$	$4 \text{ kgf} \cdot \text{m}^2 \cdots (GD^2)$

All of the chain tensions in the above formulae are the tensions when using one strand of chain. When using two strands of chain or more, calculate the chain tension for one strand and multiply it by the imbalance load factor K_u (Table 5) for the number of strands being used.

5. GENERAL SELECTION



Procedure 4-5

(1) Select the chain and the number of teeth for the small sprocket:

The number of teeth for the small sprocket and a chain that satisfies the number of revolutions of the high speed shaft and design kW can be found by using the provisional selection tables (Tables 6 & 7) or the kW rating tables. When doing so, choose a chain of minimum pitch having the necessary kW rating.

When there is a shortage of performance with a single strand, choose multi-strand chain. Further, when the outside diameter of the sprocket has been made as small as possible and the distance between shafts reduced due to the space limitation, use a multi-strand roller chain with a small pitch.

(2) Select the number of teeth for the large sprocket:

If the number of teeth for the small sprocket has been determined, then multiply this value by the speed ratio and determine the number of teeth for the large sprocket.

It is appropriate to have more than 15 teeth for the small sprocket. However, if the number of teeth for the large sprocket exceeds 120 as a result, then this is not favorable. When this happens, reduce the number of teeth for the small sprocket; although, it is recommended to use more than 13 teeth.

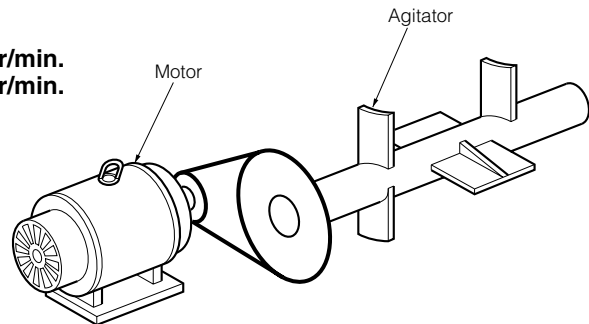
Procedure 7

If possible, try to avoid using an offset link when using an odd number of links. Instead, try adjusting the distance between the shafts until an even number of links is attained.

Example based on the general selection

Procedure 1: Data required

Machine used	: Agitator
Type of shock	: Smooth transmission
Source of power	: Motor
Rated power	: 37 kW
High speed shaft	: Shaft diameter 45 mm 750 r/min.
Low speed shaft	: Shaft diameter 60 mm 250 r/min.
Distance between shafts	: 220 mm
Space limitation	: 500 mm



Procedure 2: Use Table 2 to determine the service factor
Service factor $K_s = 1.0$

Procedure 3: Obtain design kW
 $37 \text{ kW} \times 1.0 = 37 \text{ kW}$

Procedure 4, 5: Determine the chain and the number of teeth for the sprocket.

Based on the fact that the number of revolutions of the high speed shaft is 750 r/min and the design kW is 37 kW, we can find the chain number and the number of teeth of the small sprocket.

1. According to the kilowatt rating table, the best choice would normally be a single strand of RS80-17 teeth. Since the speed ratio is 1/3 (250/750 r/min.), the necessary number of sprocket teeth would be 17 for the small sprocket and 51 for the large sprocket. However, as the outside diameters are 151 mm for 17 teeth and 427 mm for 51 teeth, it exceeds the space limitation of 500 mm. (151 + 427 > 500 mm) Therefore, these sprockets are not suitable.
2. As a single strand chain is not suitable, a double-strand RS60-2, 22 and 66 teeth would be possible. But this combination is not suitable due to the space limitation again (144 + 411 > 500 mm).
3. For triple strand, RS60-3, 15 and 45 teeth would be possible.
The sprocket's diameters are 99 mm and 284 mm respectively, the sum of which is less than 500 mm. The kilowatt rating of a 15 tooth sprocket for the RS60-3 should be confirmed by the kilowatt rating for the RS60.
The kilowatt rating of a 15 tooth sprocket is 14.1 kW at 700 r/min, and 15.9 kW at 800 r/min. The kilowatt rating at 750 r/min is about 15 kW. Since 15 kW is for a single strand chain, the kilowatt rating must be multiplied by a multi-strand factor of 2.5 for a triple strand (refer to Table 1).
Therefore, the kilowatt rating of RS60-3, 15 teeth at 750 r/min. is 37.5 kW (15 × 2.5 = 37.5)
4. This 37.5 kW rating satisfies the design kW rating.

Procedure 6: Confirm the shaft diameter

The shaft diameter is confirmed by the dimension table. The max. shaft diameter of RS60-15T is 45.5 mm and can be used for the shaft diameter of 45 mm. The maximum shaft diameter for RS60-3-45T is 63 mm and so satisfies our shaft diameter of 60 mm. The outside diameter for both sprockets is 90 mm and 284 mm respectively and fits within the prescribed space.

Procedure 7: Determine the distance between shafts

If the center distance between shafts is 220 mm, from the formula the chain length of L is as follows:

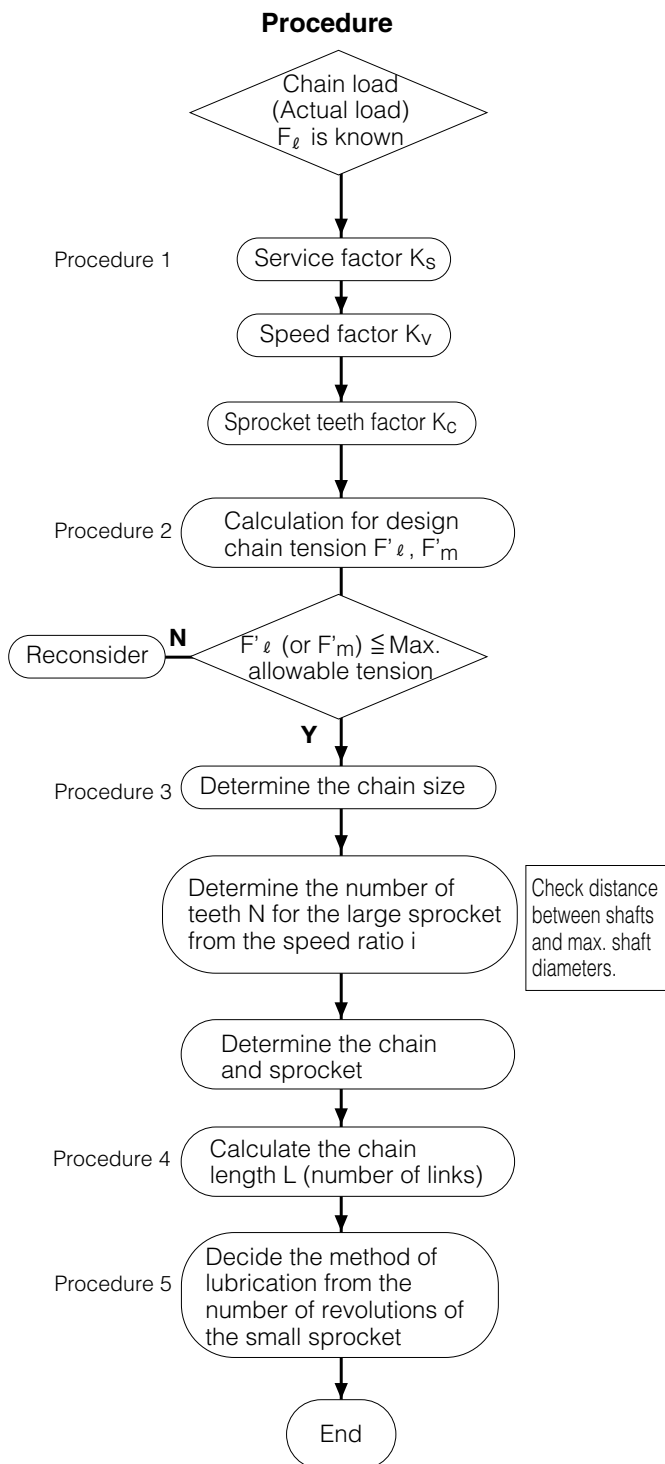
$$L = \frac{45 + 15}{2} + 2 \times \frac{220}{19.05} + \frac{\left(\frac{45 - 15}{6.28}\right)^2}{\frac{220}{19.05}} = 55.07$$

In order to have an even number of links, we raise the value below the decimal point to an integer and get 56 links.

Procedure 8: Confirm the method of lubrication

From the kW rating table, lubrication method B is selected for the small sprocket of size RS60-3-15 T at 750 r/min. Lubrication is necessary by oil bath lubrication or by slinger disc.

6. SLOW SPEED SELECTION (Starting frequency-less than 5 times/day (8hrs))

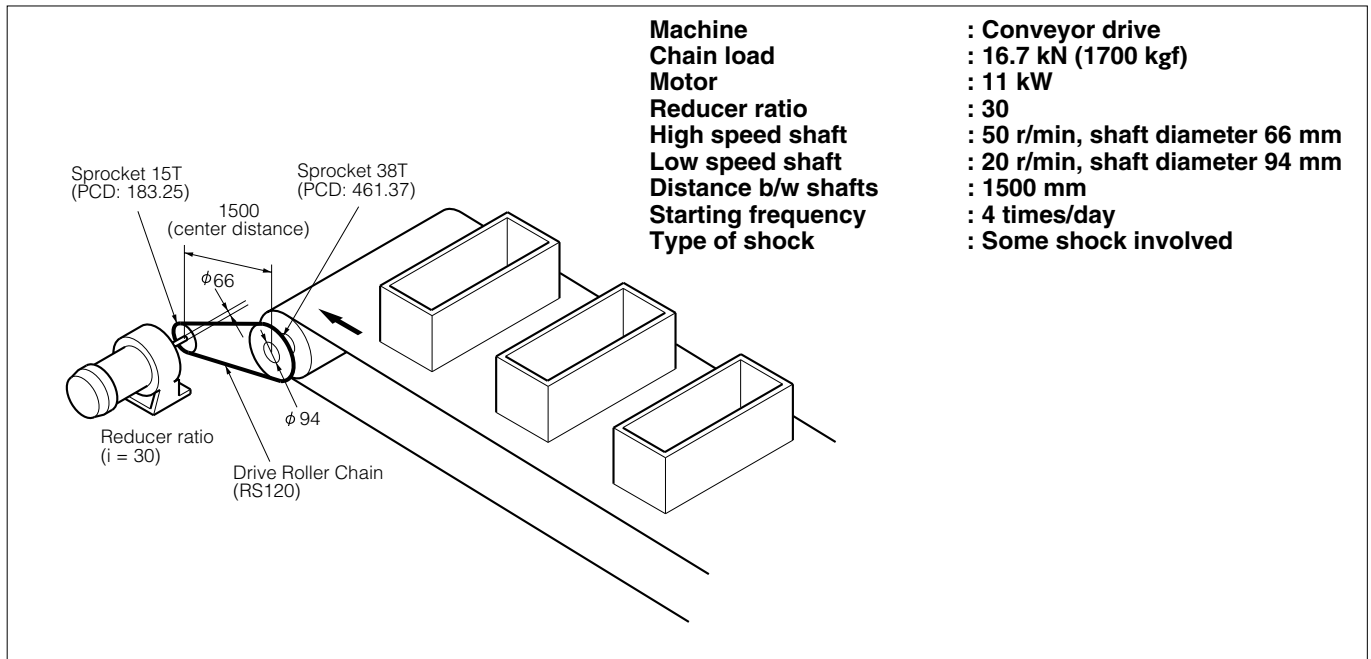


Method of selection that applies for ordinary transmission where the chain speed V is less than 50 m/min

- (1) Applicable when making a more economical selection for RS and SUPER chain.
- (2) Applicable when selecting RS-HT, SUPER-H and ULTRA SUPER chain.
- (3) In the case of severe conditions, such as transmissions with large impact, particularly from large loads and side forces, please use F-CLs and 2-pitch offset links.
- (4) When using offset and standard connecting links, allow for the following strengths as a percentage of the max. allowable tension.

M-type CL	: 100%
F-type CL	: 100%
2-pitch offset link (2POL)	: 100%
(Ref)	
1-pitch offset link (OL)	: 65%
- (5) The slow speed selection is an economical method of selection that uses the complete kW rating of the roller chain and should only be selected upon properly ascertaining the conditions of transmission. In particular it is vital that sufficient attention be given to shock loads.
- (6) Chain tension becomes large when using SUPER, RS-HT, SUPER-H and ULTRA SUPER chain, so avoid using commercially available sprockets made of cast iron since the strength of the rim and boss portions will in certain cases be insufficient. RS standard sprockets A type and B type as well as C type provide sufficient strength. (Materials such as SS400, S35C, SC450, etc. have to be used)
- (7) For the high speed side, use a sprocket with a hardening process carried out on the surface of its teeth.
- (8) Since the bearing pressure will be extremely large, be certain to lubricate the chain.

Example based on the slow speed selection



Calculate the Roller Chain speed V .

$$V = \frac{PN'n}{1000} = \frac{38.1 \times 15 \times 50}{1000} = 28.6 \text{ m/min} < 50 \text{ min}$$

This is less than 50 m/min, so slow speed selection can be used.

SI International Units	(Gravimetric Units)
------------------------	---------------------

$$F_{\ell} = 16.7 \text{ (kN)}$$

$$F_{\ell} = 1700 \text{ (kgf)}$$

Procedure 1 :

Service factor $K_s = 1.3$ some shock (Table 2)
 Speed factor $K_v = 1.06$ $V = 28.6 \text{ m/min}$ (Table 3)
 Sprocket teeth factor $K_c = 1.27$ $N' = 15^T$ (Table 3)

Procedure 2 : Calculate design chain tension F'_{ℓ}

$$\begin{aligned} F'_{\ell} &= F_{\ell} \times K_s \times K_v \times K_c \\ &= 16.7 \times 1.3 \times 1.06 \times 1.27 \\ &= 29.2 \text{ (kN)} \end{aligned}$$

Procedure 2 : Calculate design chain tension F'_{ℓ}

$$\begin{aligned} F'_{\ell} &= F_{\ell} \times K_s \times K_v \times K_c \\ &= 1700 \times 1.3 \times 1.06 \times 1.27 \\ &= 2975 \text{ (kgf)} \end{aligned}$$

Procedure 3 : Slow speed selection for RS Roller Chain

RS120 can be used since the maximum allowable tension of 30.4 kN (3100 kgf) is larger than the design chain tension 29.2 kN (2975 kgf). The driver sprocket is RS120-15T B-type (Max. shaft diameter 80 mm > Driver shaft diameter 66 mm, therefore acceptable) provided it has hardened teeth. The driven sprocket is RS120-38T B-type, provided the boss diameter is manufactured to meet the diameter of the driven shaft (94 mm).

Procedure 4 : Number of chain links

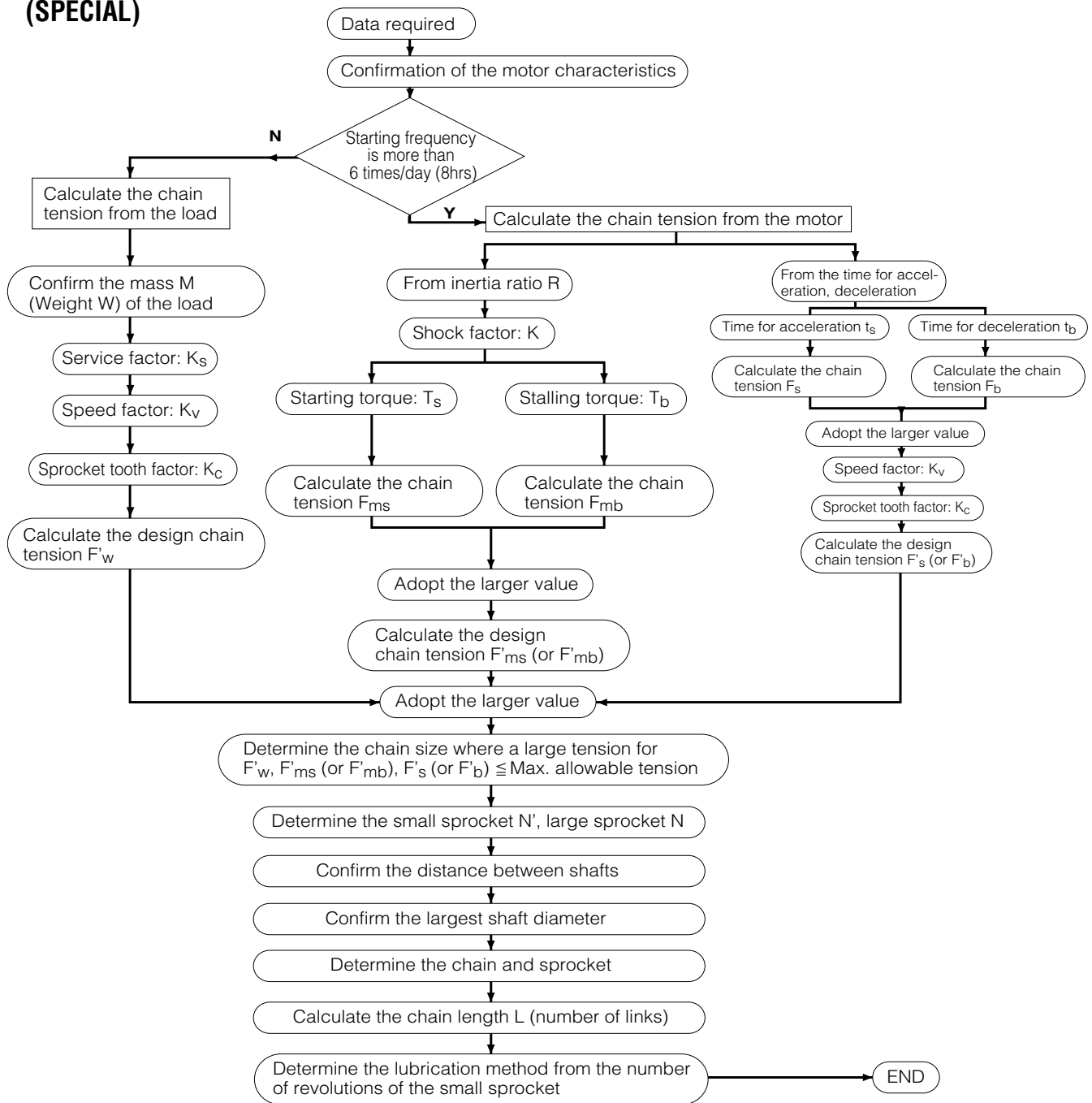
$$\begin{aligned} L &= \frac{N + N'}{2} + 2C + \frac{\left(\frac{N - N'}{6.28}\right)^2}{C} = \frac{38 + 15}{2} + 2 \times 39.37 + \frac{\left(\frac{38 - 15}{6.28}\right)^2}{39.37} \\ &= 105.58 \text{ links} \rightarrow 106 \text{ links} \quad \text{Distance between shafts} = 1508 \text{ mm} \end{aligned}$$

$$C = \frac{1500}{38.10} = 39.37$$

Procedure 5 : Lubrication method is by drip or brush

7. SLOW SPEED SELECTION (SPECIAL)

Procedure



Method of selection that applies for ordinary transmission where the chain speed V is less than 50 m/min

- (1) Applicable when making a more economical selection for RS and SUPER chain.
- (2) Applicable when selecting RS-HT, SUPER-H and ULTRA SUPER.
- (3) In the case of severe conditions, such as transmissions with large impact, particularly from large loads and side forces, please use F-CLs and 2-pitch offset links.
- (4) When using offset links and standard connecting links, allow for the following strengths as a percentage of the maximum allowable tension.

M-type CL	: 100%
F-type CL	: 100%
2-pitch offset link (2POL)	: 100%
(Ref)	
1-pitch offset link (OL)	: 65%
- (5) The slow speed selection is an economical method of

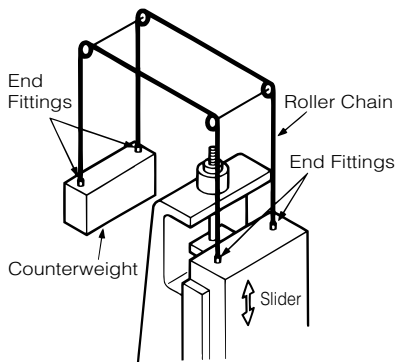
selection that uses the complete kW rating of the roller chain and should only be selected upon properly ascertaining the conditions of transmission. In particular it is vital that sufficient attention is given to shock loads.

- (6) Chain tension becomes large when using SUPER, RS-HT, SUPER-H, ULTRA SUPER chains, so avoid using commercially available sprockets made of cast iron since the strength of the rim and boss portions will, in certain cases, be insufficient. RS standard sprockets A type and B type as well as C type provide sufficient strength. (Materials such as SS400, S35C, SC450, etc. have to be used)
- (7) For the high speed side, use a sprocket with a hardening process carried out on the surface of its teeth.
- (8) Since the bearing pressure will be extremely large, make certain to lubricate the chain.

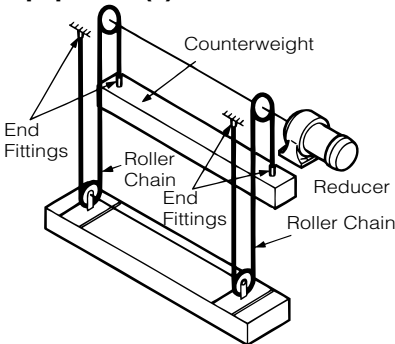
8. Selection Method for Lifting Transmissions

There are many examples of where chain is used for lifting. By making use of Roller Chain features, choosing the right chain and following the important points, it is possible to use Roller Chain for lifting transmissions. A model lifting application is illustrated below. (Please give special consideration to safety devices)

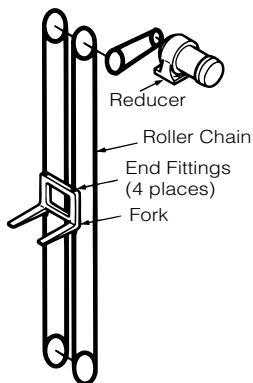
Balancing



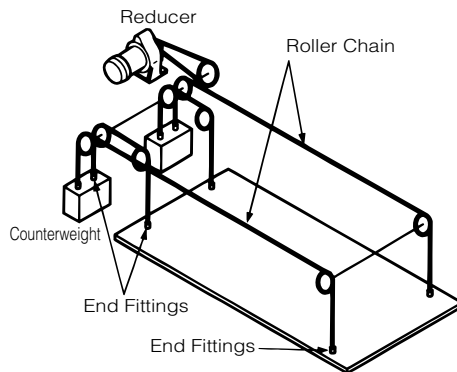
Ascending/Descending Equipment (1)



Ascending/Descending Equipment (2)

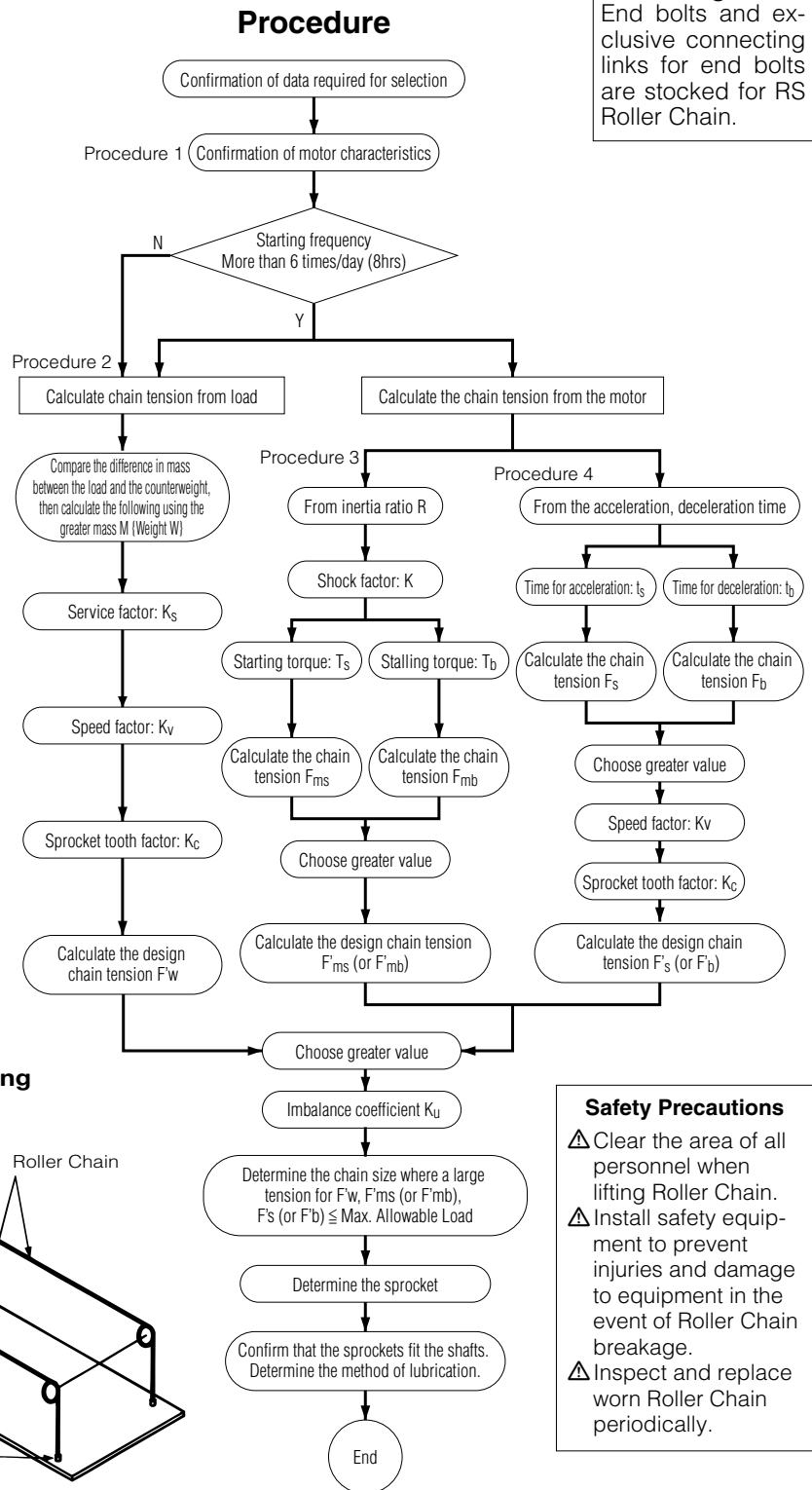


Ascending/Descending Equipment (3)



Examples of Lifting Transmissions

End Fittings
End bolts and exclusive connecting links for end bolts are stocked for RS Roller Chain.



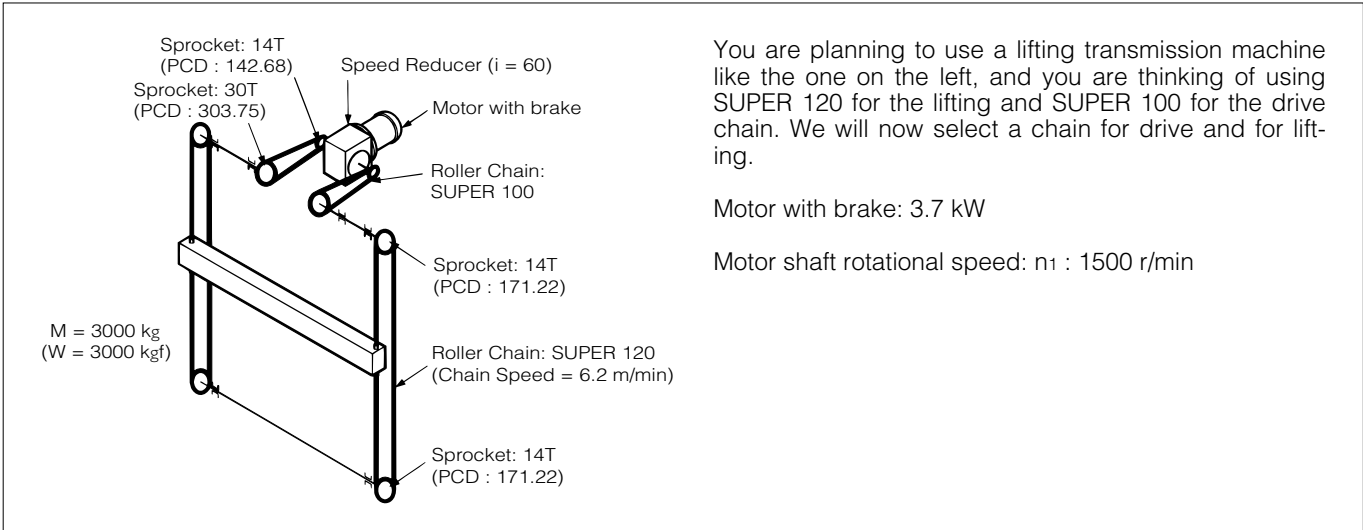
Safety Precautions

- ⚠ Clear the area of all personnel when lifting Roller Chain.
- ⚠ Install safety equipment to prevent injuries and damage to equipment in the event of Roller Chain breakage.
- ⚠ Inspect and replace worn Roller Chain periodically.

SELECTION

- ⚠ Roller Chain Selection for Lifting Applications**
- (1) When making your selection, calculate the tension from the load and from the motor and apply the greater of the two. As a rule of thumb, if the greater value is lower than the Max. Allowable Load of the chain you are thinking of choosing, then it may be selected.
 - (2) If there are any laws or guidelines for chain selection, check and calculate accordingly. Make sure to follow the manufacturer's selections and select the safer of the two selections.
 - (3) The chain speed should be less than 50m/min.
 - (4) Use F-Type (Semi Press-fit) connecting links. Offset links cannot be used.
 - (5) Lubricate the chain joints as much as possible after you reduce the loads. Sufficient lubrication is also required at end fittings (end bolts and connecting links, etc.) and connecting parts, etc.

Example of Selection for Lifting Transmission Roller Chain



You are planning to use a lifting transmission machine like the one on the left, and you are thinking of using SUPER 120 for the lifting and SUPER 100 for the drive chain. We will now select a chain for drive and for lifting.

Motor with brake: 3.7 kW

Motor shaft rotational speed: $n_1 : 1500 \text{ r/min}$

SI International Units

Procedure 1: Confirmation of motor characteristics

Rated torque: $T_n = 0.024 \text{ (kN}\cdot\text{m)}$
 Starting torque: $T_s = 0.061 \text{ (kN}\cdot\text{m)}$
 Stalling torque: $T_b = 0.073 \text{ (kN}\cdot\text{m)}$
 Motor moment of inertia: $I_m = 0.015 \text{ (kg}\cdot\text{m}^2)$

Procedure 2: Calculate chain tension from load

$$\begin{aligned} \text{Chain tension } F_w &= M \\ &= W \times \frac{G}{1000} = 3000 \times \frac{9.80665}{1000} = 29.4 \text{ (kN)} \end{aligned}$$

Chain speed $V = 6.2 \text{ m/min}$ Speed factor: $K_v = 1.02$
 14-tooth sprocket for lifting Sprocket tooth factor: $K_c = 1.28$
 Minimal shock Service factor: $K_s = 1.3$
 For double strand lifting Imbalance load coefficient $K_u = 0.6$

Design chain tension

$$\begin{aligned} F'_w &= F_w \times K_s \times K_v \times K_c \times K_u \\ &= 29.4 \times 1.3 \times 1.02 \times 1.28 \times 0.6 \\ &= 29.9 \text{ (kN)} \end{aligned} \quad \text{①}$$

Procedure 3: Calculate the chain tension from the motor

Converted moment of inertia of the loaded prime mover output shaft

$$\begin{aligned} I_\ell &= M \times \left(\frac{V}{2\pi n_1} \right)^2 \\ &= 3000 \times \left(\frac{6.2}{2 \times \pi \times 1500} \right)^2 \\ &= 0.00130 \text{ (kg}\cdot\text{m}^2) \end{aligned}$$

Moment of inertia of the prime mover output shaft (I), $I_m = 0.015 \text{ (kg}\cdot\text{m}^2)$

$$\begin{aligned} \text{Inertia ratio (R)} \quad R &= \frac{I_\ell}{I_m} = \frac{0.00130}{0.015} \\ &= 0.087 \end{aligned}$$

(Gravimetric Units)

Procedure 1: Confirmation of motor characteristics

Rated torque: $T_n = 2.4 \text{ (kgf}\cdot\text{m)}$
 Starting torque: $T_s = 6.0 \text{ (kgf}\cdot\text{m)}$
 Stalling torque: $T_b = 7.2 \text{ (kgf}\cdot\text{m)}$
 Motor GD^2 : $GD^2_m = 0.06 \text{ (kgf}\cdot\text{m}^2)$

Procedure 2: Calculate chain tension from load

$$\text{Chain tension } F_w = W = 3000 \text{ (kgf)}$$

Design chain tension

$$\begin{aligned} F'_w &= F_w \times K_s \times K_v \times K_c \times K_u \\ &= 3000 \times 1.3 \times 1.02 \times 1.28 \times 0.6 \\ &= 3055 \text{ (kgf)} \end{aligned} \quad \text{①}$$

Procedure 3: Calculate the chain tension from the motor

Converted moment of inertia of the loaded prime mover output shaft

$$\begin{aligned} GD^2_\ell &= W \times \left(\frac{V}{\pi n_1} \right)^2 \\ &= 3000 \times \left(\frac{6.2}{\pi \times 1500} \right)^2 \\ &= 0.00519 \text{ (kgf}\cdot\text{m}^2) \end{aligned}$$

Moment of inertia of the prime mover output shaft $GD^2_m = 0.06 \text{ (kgf}\cdot\text{m}^2)$

$$\begin{aligned} \text{Inertia ratio (R)} \quad R &= \frac{GD^2_\ell}{GD^2_m} = \frac{0.00519}{0.06} \\ &= 0.087 \end{aligned}$$

As there is no play ($R < 0.2$) in the system ($R = 0.2$), the coefficient of shock $K = 0.23$

Starting torque: $T_s = 0.061$ (kN·m)

Chain tension from starting torque

$$F_{ms} = T_s \times i \times \frac{30}{14} \times 1000 / \left(\frac{d}{2} \right)$$

$$= 0.061 \times 60 \times \frac{30}{14} \times 1000 / \left(\frac{171.22}{2} \right)$$

$$= 91.6 \text{ (kN)}$$

Stalling torque: $T_b = 0.073$ (kN·m)

Chain tension from stalling torque

$$F_{mb} = T_b \times i \times \frac{30}{14} \times 1000 \times 1.2 / \left(\frac{d}{2} \right)$$

$$= 0.073 \times 60 \times \frac{30}{14} \times 1000 \times 1.2 / \left(\frac{171.22}{2} \right)$$

$$= 131.6 \text{ (kN)}$$

Use the greater value of F_{mb} to calculate chain tension as $F_{mb} > F_{ms}$.

Design chain tension

$$F'_{mb} = F_{mb} \times K \times K_v \times K_c \times K_u$$

$$= 131.6 \times 0.23 \times 1.02 \times 1.28 \times 0.6$$

$$= 23.7 \text{ (kN) } \dots\dots\dots \textcircled{2}$$

Procedure 4: Calculate the chain tension from motor acceleration and deceleration.

Working torque $T_m = \frac{T_s + T_b}{2} = \frac{0.061 + 0.073}{2}$
 $= 0.067$ (kN·m)

Load torque $T_\ell = \frac{M \times d}{2 \times 1000 \times i} \times \frac{G}{1000}$
 $= \frac{3000 \times 171.22}{2 \times 1000 \times 60 \times \frac{30}{14}} \times \frac{G}{1000}$
 $= 0.02$ (kN·m)

Motor acceleration time

$$t_s = \frac{(I_m + I_\ell) \times n_1}{375 \times (T_m - T_\ell)} \times \frac{G}{1000} \times 4$$

$$= \frac{(0.015 + 0.00130) \times 1500}{375 \times (0.067 - 0.02)} \times \frac{G}{1000} \times 4$$

$$= 0.054 \text{ (s)}$$

Motor deceleration time

$$t_b = \frac{(I_m + I_\ell) \times n_1}{375 \times (T_m + T_\ell)} \times \frac{G}{1000} \times 4$$

$$= \frac{(0.015 + 0.00130) \times 1500}{375 \times (0.067 + 0.02)} \times \frac{G}{1000} \times 4$$

$$= 0.029 \text{ (s)}$$

Because t_b is smaller than t_s , chain tension from motor deceleration F_b is greater than that of acceleration, so F_b should be used.

Chain tension from acceleration

$$F_b = \frac{M \times V}{t_b \times 60 \times 1000} + F_w$$

$$= \frac{3000 \times 6.2}{0.029 \times 60 \times 1000} + 29.4$$

$$= 40.1 \text{ (kN)}$$

Starting torque: $T_s = 6.0$ (kgf·m)

Chain tension from starting torque

$$F_{ms} = T_s \times i \times \frac{30}{14} \times 1000 / (d/2)$$

$$= 6.0 \times 60 \times \frac{30}{14} \times 1000 / (171.22/2)$$

$$= 9011 \text{ (kgf)}$$

Stalling torque: $T_b = 7.2$ (kgf·m)

Chain tension from stalling torque

$$F_{mb} = T_b \times i \times \frac{30}{14} \times 1000 \times 1.2 / (d/2)$$

$$= 7.2 \times 60 \times \frac{30}{14} \times 1000 \times 1.2 / (171.22/2)$$

$$= 12976 \text{ (kgf)}$$

Design chain tension

$$F'_{mb} = F_{mb} \times K \times K_v \times K_c \times K_u$$

$$= 12976 \times 0.23 \times 1.02 \times 1.28 \times 0.6$$

$$= 2338 \text{ (kgf) } \dots\dots\dots \textcircled{2}$$

Procedure 4: Calculate the chain tension from motor acceleration and deceleration.

Working torque $T_m = \frac{T_s + T_b}{2} = \frac{6.0 + 7.2}{2}$
 $= 6.6$ (kgf·m)

Load torque $T_\ell = \frac{W \times d}{2 \times 1000 \times i}$
 $= \frac{3000 \times 171.22}{2 \times 1000 \times 60 \times \frac{30}{14}}$
 $= 2.0$ (kgf·m)

Motor acceleration time

$$t_s = \frac{(GD_m^2 + GD_\ell^2) \times n_1}{375 \times (T_m - T_\ell)}$$

$$= \frac{(0.06 + 0.00519) \times 1500}{375 \times (6.6 - 2.0)}$$

$$= 0.057 \text{ (s)}$$

Motor deceleration time

$$t_b = \frac{(GD_m^2 + GD_\ell^2) \times n_1}{375 \times (T_m + T_\ell)}$$

$$= \frac{(0.06 + 0.00519) \times 1500}{375 \times (6.6 + 2.0)}$$

$$= 0.030 \text{ (s)}$$

SELECTION

Design chain tension $F'_b = F_b \times K_v \times K_c \times K_u$
 $= 40.1 \times 1.02 \times 1.28 \times 0.6$
 $= 31.4 \text{ (kN)} \dots\dots\dots \textcircled{3}$

Design chain tension $F'_b = F_b \times K_v \times K_c \times K_u$
 $= 4054 \times 1.02 \times 1.28 \times 0.6$
 $= 3176 \text{ (kgf)} \dots\dots\dots \textcircled{3}$

When comparing the calculated design chain tensions in Steps ①, ②, and ③, note that F_b in Step ③ is the greatest.

Comparing F'_b (31.4 kN) with the maximum allowable load of SUPER 120 chain (39.2 kN), $F'_b < 39.2 \text{ kN}$. Therefore, this chain may be selected. The drive chain is

$$F'_b \times \frac{d}{d'} = 31.4 \times \frac{171.22}{303.75}$$

$$= 17.7 \text{ kN} < 30.4 \text{ kN}$$

This value is less than the maximum allowable load of SUPER 100 chain, so it may also be used.

Comparing F'_b (3176 kgf) with the maximum allowable load of SUPER 120 chain (4000 kgf), $F'_b < 4000 \text{ kgf}$. Therefore, this chain may be selected. The drive chain is

$$F'_b \times \frac{d}{d'} = 3176 \times \frac{171.22}{303.75}$$

$$= 1790 \text{ kgf} < 3100 \text{ kgf}$$

This value is less than the maximum allowable load of SUPER 100 chain, so it may also be used.

(Conclusion)

It is possible to use SUPER 120 for lifting applications and SUPER 100 for drive applications. However, if operational restrictions occur due to overload, the chains will be subjected to the following loads: Drive chain: $F_d = \frac{0.073 \times 1000 \times 60 \times 2}{142.68} = 61.4 \text{ kN}$ (6266 kgf) (per strand), $F_d \times K_u = 61.4 \text{ kN} \times 0.6 = 36.8 \text{ kN}$ (3757 kgf), Lifting chain: $F_d \times \frac{303.75}{171.22} = 65.3 \text{ kN}$ (6657 kgf). In this case, since there is a possibility of chain plastic deformation, increase the chain size by selecting SUPER 120-2 for lifting transmission and SUPER 120 for drive transmission, just to be safe.



Weight required for counterweight to prevent sprocket tooth-jumping when using Roller Chain in lifting transmission applications.

$$T_k = T_o \times \left\{ \frac{\sin \phi}{\sin (\phi + 2\alpha)} \right\}^{K-1}$$

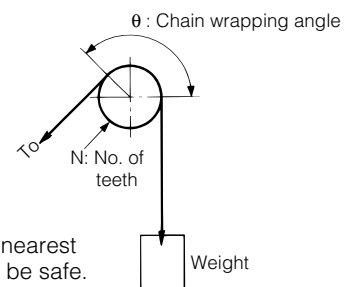
T_k : Minimum weight tension (Minimum back-tension)

T_o : Roller Chain tension

ϕ : Sprocket minimum pressure angle $\phi = 17^\circ - \frac{64^\circ}{N}$

2α : Sprocket dividing angle $2\alpha = \frac{360^\circ}{N}$

K : Engaging No. of teeth $K = \frac{\theta}{360^\circ} \times N \dots$ Round-up to the nearest whole number to be safe.



If $T_o = 1100 \text{ kgf}$, $N = 13$, and $\theta = 120^\circ$, then

$$\phi = 17^\circ - \frac{64^\circ}{N} = 17^\circ - \frac{64^\circ}{13} = 12.077$$

$$2\alpha = \frac{360^\circ}{N} = \frac{360^\circ}{13} = 27.692$$

$$K = \frac{\theta}{360^\circ} \times N = \frac{120^\circ}{360^\circ} \times 13 = 4.33 \dots K = 4$$

$$T_k = 1100 \times \left\{ \frac{\sin 12.077}{\sin (12.077 + 27.692)} \right\}^{4-1} = 38.5 \text{ (kg)}$$

Accordingly, tooth-jumping will not occur if a 39 kg weight is used. However, this will change depending on the layout and amount of wear on the Roller Chain and sprocket teeth. Please use the above as a reference.



9. Selection by Temperature

9.1 RS Roller Chain Selection by Temperature

Method of selection that allows for a decrease in strength depending on temperature. Additionally, lubrication should be carried out using a suitable lubricant according to the operating temperature.

- 1) Problems of roller chain transmission at high temperatures
 - 1) Increase in wear from a decrease in hardness
 - 2) Increase in elongation from softening
 - 3) Poor articulation and an increase in wear from depletion/carbonization of oil
 - 4) Increase in wear and poor articulation from scaling
- 2) Problems of roller chain transmission at low temperatures
 - 1) Decrease in shock resistance from brittleness at low temperatures
 - 2) Solidification of lubricant
 - 3) Poor articulation from frost and water adhesion

Table 10 Standard for transmission performance of RS Roller Chain for high and low temperatures.

Temperature	RS Roller Chain		KT Cold Resistant type*
	RS60 and under	RS80 and over	
Below -60°C (-76°F)	—	—	Unusable
-60°C ~ -50°C (-76°F ~ -58°F)	—	—	Catalog value × 1/2
-50°C ~ -40°C (-58°F ~ -40°F)	—	Unusable	Catalog value × 2/3
-40°C ~ -30°C (-40°F ~ -22°F)	Unusable	Catalog value × 1/4	Catalog value
-30°C ~ -20°C (-22°F ~ -4°F)	Catalog value × 1/4	Catalog value × 1/3	Catalog value
-20°C ~ -10°C (-4°F ~ +14°F)	Catalog value × 1/3	Catalog value × 1/2	Catalog value
-10°C ~ +60°C (+14°F ~ +140°F)	Catalog value	Catalog value	Catalog value
+60°C ~ +150°C (+140°F ~ +302°F)	Catalog value	Catalog value	Unusable
+150°C ~ +200°C (+302°F ~ +392°F)	Catalog value × 3/4	Catalog value × 3/4	—
+200°C ~ +250°C (+392°F ~ +482°F)	Catalog value × 1/2	Catalog value × 1/2	—
Above +250°C (482°F)	Unusable	Unusable	—

- Note:** 1. *KT: Made-to-order
 2. Note that the ambient temperature and the temperature of the chain itself are different.

9.2 Method of selection of SS / NS Stainless Steel Roller Chain for high temperatures (+400°C / +752°F and above)

Chain strength falls as the temperature of the chain becomes high. The temperature limit for use is decided by the temperature of the chain itself. If your operation runs at temperatures higher than +400°C (+752°F), consult the manufacturer before making your chain selection. Note that the chain cannot be used in temperatures in excess of +700°C (+1,292°F). The chain speed should be less than 50 m/min for selections by temperature.

Changes and important points regarding high temperature environments:

- 1) In order to prevent poor articulation and poor roller rotation from heat expansion, clearances in each part need to be changed.
- 2) It is possible that the chain will break (creep rupture) at lower loads when the temperature becomes higher.

10. Special Selection Method for Corrosion-Resistant Roller Chain

Slow speed selection (selection by max. allowable load) is employed for Corrosion-Resistant Roller Chain Selection.

- 1) The maximum allowable load of some Corrosion-Resistant Roller Chain is lower than that of Standard RS Roller Chain.
- 2) Avoid using offset links wherever possible.
- 3) The chain speed should be less than 50 m/min for selections made in "Special Selection Method."
- 4) Refer to the following page when substances such as acids, alkalis or chemicals come into contact with the chain.
- 5) Selection formula

$$\boxed{\text{Max. chain working load}} \times \boxed{\text{Service factor } K_S} \times \boxed{\text{Speed factor } K_V} \times \boxed{\text{Sprocket teeth factor } K_C} \leq \boxed{\text{Max. allowable load of the chain}}$$

11. Anti-Corrosion Reference Guide for Corrosion Resistant Roller Chain (Table 11)

Since corrosion resistance varies substantially according to operating conditions, this chart should not be considered as a guarantee. Using this chart as a reference, make sure to check the corrosion resistance of the chain in advance according to the actual operating conditions before deciding on the type of chain to use.

○ : Highly corrosion resistant
 △ : Corrosion resistant depending on operating conditions
 × : Not corrosion resistant
 - : Unknown

Chemical/Foodstuff	Corrosion Resistant Roller Chain						
	SS	LS	AS	NS	TI	PC	PC-SY
Acetone	20°C	○	○	○	○	○	×
Oil (Plant, Mineral)	20°C	○	○	○	○	○	○
Linseed Oil	100% 20°C	○	—	△	○	○	—
Sulphur Dioxide (wet)	20°C	○	—	×	○	○	—
Alcohol (Methyl, Ethyl, Propyl, Butyl)		○	○	○	○	○	○
Ammonia Water	20°C	○	○	○	○	○	○
Whisky	20°C	○	○	○	○	○	○
Ether (Ethyl Ether)	20°C	○	○	○	○	○	○
Zinc Chloride	50% 20°C	△	△	×	△	○	△
Ammonium Chloride	50% Boiling Point	△	△	×	○	○	—
Potassium Chloride	Saturated 20°C	○	○	△	○	○	—
Calcium Chloride	Saturated 20°C	△	—	×	○	○	△
Ferric Chloride	5% 20°C	△	△	×	△	○	—
Sodium Chloride	5% 20°C	○	○	△	○	○	○
Hydrochloric Acid	2% 20°C	×	×	×	×	○	×
Chlorine Gas (dry)	20°C	△	—	×	△	○	—
Chlorine Gas (wet)	20°C	×	×	×	△	○	—
Chlorine Water		×	×	×	○	○	×
Oleic Acid	20°C	○	○	○	○	○	—
Seawater	20°C	△	△	×	○	○	△
Sodium Perchlorate	10% Boiling Point	○	—	×	○	○	—
Hydrogen Peroxide	30% 20°C	○	—	△	○	○	×
Gasoline	20°C	○	○	○	○	○	○
Potassium Permanganate	Saturated 20°C	○	○	○	○	○	—
Formic Acid	50% 20°C	○	×	○	○	○	×
Milk	20°C	○	○	○	○	○	○
Citric Acid	50% 20°C	○	○	○	○	○	—
Glycerol	20°C	○	○	○	○	○	○
Creosote	20°C	○	—	○	○	○	—
Chromic Acid	5% 20°C	○	○	△	○	○	×
Ketchup	20°C	○	○	○	○	○	○
Developing Solution (Photo)	20°C	○	—	△	○	○	○
Synthetic Detergent		○	○	○	○	○	○
Coffee	Boiling	○	○	○	○	○	○
Cola Syrup		○	○	○	○	○	○
Acetic Acid	10% 20°C	○	○	○	○	○	○
Sugar Solution	20°C	○	○	○	○	○	○
Calcium Hypochlorite (Bleaching Powder) Available chlorine 11 - 14% 20°C		○	—	×	○	○	×
Sodium Hypochlorite	10% 20°C	×	×	×	○	○	×
Sodium Cyanide	20°C	○	○	—	○	○	—
Carbon Tetrachloride (dry)	20°C	○	○	○	○	○	○
Potassium Dichromate	10% 20°C	○	○	○	○	○	—
Oxalic Acid	10% 20°C	○	○	△	○	○	—
Tartaric Acid	10% 20°C	○	○	○	○	○	○
Nitric Acid	5% 20°C	○	—	△	○	○	×
Ammonium Nitrate	Saturated Boiling	○	○	○	○	○	△

Note: SUS304 is included in SS

Chemical/Foodstuff	Corrosion Resistant Roller Chain						
	SS	LS	AS	NS	TI	PC	PC-SY
Potassium Nitrate	25% 20°C	○	○	○	○	○	—
Potassium Nitrate	25% Boiling Point	○	—	×	○	○	—
Vinegar	20°C	△	—	×	○	○	△
Potassium Hydroxide (Caustic Potash)	20% 20°C	○	×	○	○	○	○
Calcium Hydroxide (Slaked Lime)	20% Boiling	○	—	○	○	○	○
Sodium Hydroxide (Caustic Soda)	25% 20°C	○	×	○	○	○	○
Stearic Acid	100% Boiling Point	×	×	×	○	○	×
Soft Drink	20°C	○	○	○	○	○	○
Carbolic Acid	20°C	○	—	○	○	○	×
Petroleum	20°C	○	—	○	○	○	—
Soapy Water	20°C	○	○	○	○	○	○
Carbonated Water		○	○	○	○	○	—
Sodium Hydrogen Carbonate	20°C	○	○	○	○	○	—
Sodium Carbonate	Saturated Boiling Point	○	○	○	○	○	—
Sodium Thiosulfate	25% Boiling Point	○	○	○	○	○	—
Turpentine Oil	35°C	○	—	○	○	○	—
Kerosene	20°C	○	○	○	○	○	—
Varnish		○	—	○	○	○	—
Concentrated Nitric Acid	65% 20°C	○	×	×	○	○	×
Concentrated Nitric Acid	65% Boiling	△	×	×	△	○	×
Lactic Acid	10% 20°C	○	○	△	○	○	—
Honey, Molasses		○	○	○	○	○	○
Paraffin	20°C	○	○	○	○	○	○
Beer	20°C	○	○	○	○	○	○
Picric Acid	Saturated 20°C	○	—	○	○	○	—
Fruit Juice	20°C	○	○	△	○	○	○
Benzene	20°C	○	○	○	○	○	○
Boric acid	50% 100°C	○	—	○	○	○	—
Formalin (Formaldehyde)	40% 20°C	○	○	○	○	○	—
Mayonnaise	20°C	○	○	△	○	○	○
Water		○	○	○	○	○	○
Vegetable Juice	20°C	○	○	○	○	○	○
Lard		○	—	○	○	○	—
Butyric Acid	20°C	○	—	○	○	○	—
Hydrogen Sulfide (dry)		○	—	○	○	○	○
Hydrogen Sulfide (wet)		×	×	×	×	○	×
Sulphuric Acid	5% 20°C	×	×	×	×	○	×
Zinc Sulfate	25% Saturated 20°C	○	○	○	○	○	—
Aluminium Sulfate	Saturated 20°C	○	○	×	○	○	—
Ammonium Sulfate	Saturated 20°C	○	○	△	○	○	—
Sodium Sulfate	Saturated 20°C	○	○	○	○	○	—
Malic Acid	50% 50°C	○	○	○	○	○	○
Phosphoric Acid	5% 20°C	○	—	△	○	○	×
Phosphoric Acid	10% 20°C	△	×	△	△	○	×
Wine	20°C	○	○	○	○	○	○