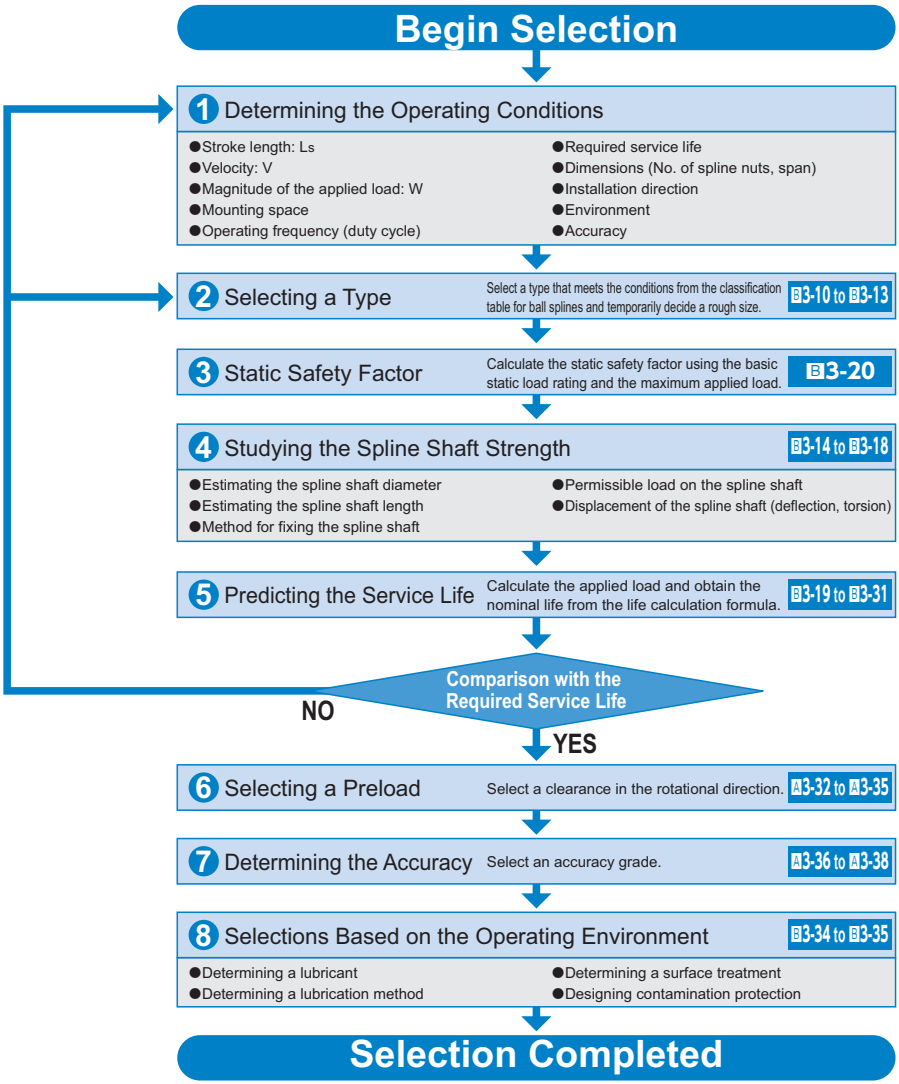
Flowchart for Selecting a Ball Spline

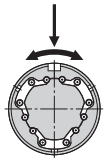
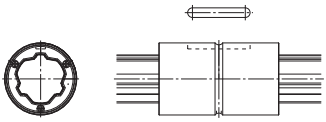
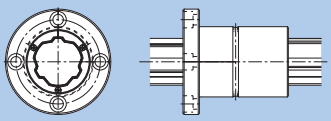
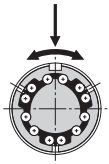
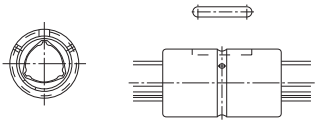
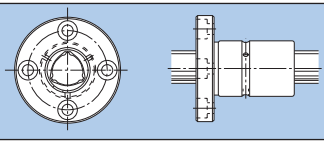
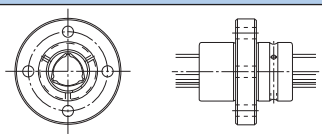
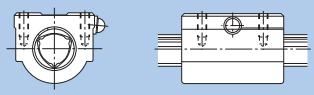
Ball Spline Selection Procedure

The following is a flowchart to reference when selecting a ball spline.



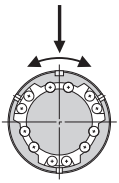
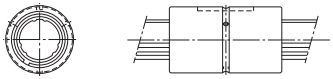
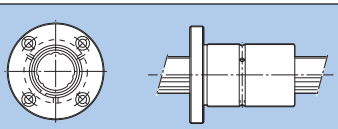
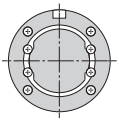
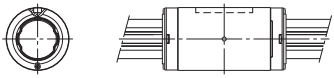
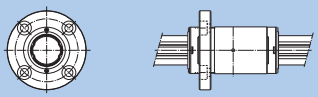
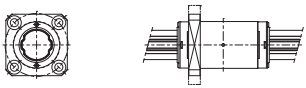
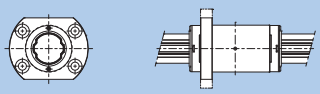
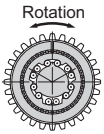
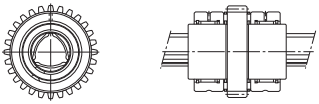
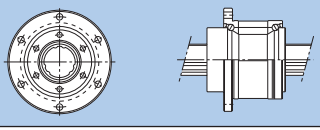
Selecting a Type

There are three types of ball spline: high-torque, medium-torque, and rotary. You can choose a type according to the intended use. In addition, a wide array of spline nut shapes are available for each type, enabling the user to choose a desired shape according to the mounting or service requirements.

Classification		Model	Shape	Shaft diameter
High-torque Caged Ball types		Model SLS Model SLS-L		Nominal shaft diameter 25 mm to 100 mm
		Model SLF		Nominal shaft diameter 25 mm to 100 mm
High-torque types		Model LBS Model LBST		Nominal shaft diameter 15 mm to 150 mm
		Model LBF		Nominal shaft diameter 15 mm to 100 mm
		Model LBR		Nominal shaft diameter 15 mm to 100 mm
		Model LBH		Nominal shaft diameter 15 mm to 50 mm

[1] For the dimensional table for each model, please see "A Product Descriptions."

Dimensional table[1]	Structure and features	Major applications
A3-46	• Making the shape more circular has significantly improved torsion and flexural rigidity compared to conventional high-torque spline shafts. • Caged Ball technology enables sustained circulation of evenly spaced balls and achieves high-speed response, improving the takt time of machines. • Caged Ball technology eliminates collision and mutual friction between balls and realizes low noise, reduced running sound, and low particle generation. • Caged Ball technology substantially increases grease retention and achieves long-term maintenance-free operation. • Caged Ball technology and a new circulation method achieve stable and smooth motion with small rolling fluctuation.	• Columns and arms of industrial robots • Automatic loaders • Transfer machines • Automatic conveyance systems • Tire molding machines • Spindles of spot-welding machines • Guide shafts of high-speed automatic coating machines • Riveting machines • Wire winders • Work heads of electric discharge machines • Grinding machine spindle drive shafts • Speed gears • Precision indexing machines
A3-48		
A3-60	• The spline shaft has three crests formed equidistantly every 120°. Six rows of balls (two for each crest) are arranged to hold the crests from both sides. The angular-contact design of the ball contact areas allows the even application of an appropriate preload. • Since the balls circulate inside the spline nut, the outer dimensions of the spline nut are compactly designed. • Even under a large preload, smooth straight motion is achieved. • Since the contact angle is large (45°) and the displacement is minimal, high rigidity is achieved. • No angular backlash occurs. • These models are capable of transmitting a large torque.	
A3-64		
A3-66		
A3-68		

Classification		Model	Shape	Shaft diameter
Medium-torque types		Model LT		Nominal shaft diameter 4 mm to 100 mm
		Model LF		Nominal shaft diameter 6 mm to 60 mm
		Model LT-X		Nominal shaft diameter 4 mm to 30 mm
		Model LF-X		Nominal shaft diameter 4 mm to 30 mm
		Model LFK-X		Nominal shaft diameter 5 mm to 30 mm
		Model LFH-X		Nominal shaft diameter 5 mm to 30 mm
Rotary types		Model LBG Model LBGT		Nominal shaft diameter 20 mm to 85 mm
		Model LTR-A Model LTR		Nominal shaft diameter 8 mm to 60 mm

[1] For the dimensional table for each model, please see "A Product Descriptions."

Dimensional table[1]	Structure and features	Major applications	
A3-82	• The spline shaft has two to three crests. On both sides of each crest, two rows (four to six rows in total) of balls are arranged to hold the crest from both sides. This design allows an appropriate preload to be evenly applied. • The contact angle of 20° and an appropriate preload level eliminate angular backlash, providing high-torque moment rigidity.	• Die-set shafts and similar applications requiring straight motion under a heavy load • Loading systems and similar applications requiring rotation to a given angle at a fixed position • Automatic gas-welding machine spindles and similar applications requiring a whirl-stop on one shaft	• Columns and arms of industrial robots • Spot-welding machines • Riveting machines • Book-binding machines • Automatic fillers • XY recorders • Automatic spinners • Optical measuring instruments
A3-84			
A3-86	• The length and external diameter of the LT-X ball spline's outer cylinder are the same as those of an LM-series linear bushing, meaning the nut can be replaced with a linear bushing.		
A3-88	• The length and external diameter of the LF-X ball spline's nut are the same as those of the Model LMF linear bushing, meaning the nut can be replaced with a linear bushing.		
A3-90	• The Model LFK-X is a lightweight and compact product designed with a lower core height than the Model LF-X.		
A3-92	• The Model LFH-X is a lightweight and compact product designed with a lower core height than the Model LFK-X.		
A3-104	• These models are unit types that have the same contact structure as Model LBS. The flange circumference on the spline nut is machined to have gear teeth, and radial and thrust needle bearings are compactly combined on the circumference of the spline nut.	• Speed gears for high torque transmission	
A3-116	• This ball spline is a lightweight and compact type featuring the same outer diameter of the nut as the Model LT-X and ball raceways that use angular contact to accommodate support bearings.	• Z axes of SCARA robots • Wire winders	

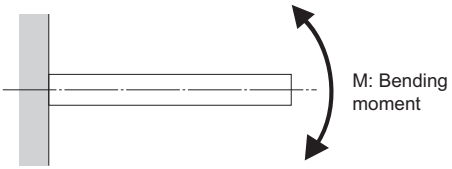
Studying the Spline Shaft Strength

The spline shaft of the ball spline is a compound shaft capable of receiving a radial load and torque. When the load and torque are large, the spline shaft strength must be taken into account.

Spline Shaft Receiving a Bending Load

When a bending load is applied to the spline shaft of a ball spline, obtain the spline shaft diameter using equation (1) below.

$$M = \sigma \cdot Z \quad \text{and} \quad Z = \frac{M}{\sigma} \qquad \cdots\cdots\cdots\cdots (1)$$



M : Maximum bending moment acting on the spline shaft (N·mm)
σ : Permissible bending stress of the spline shaft (98 N/mm²)
Z : Section modulus of the spline shaft (mm³)
(see Table 3 on A3-17, Table 4 on A3-18, Table 5 on A3-19, and Table 6 on A3-20)

Reference: Section Modulus (Solid Circle)

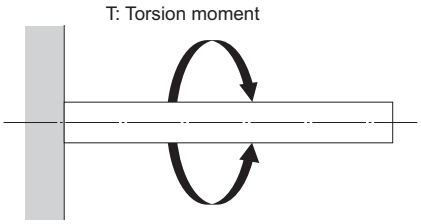
$$Z = \frac{\pi \cdot d^3}{32}$$

Z : Section modulus (mm³)
d : Shaft outer diameter (mm)

Spline Shaft Receiving a Torsion Load

When a torsion load is applied on the spline shaft of a ball spline, the spline shaft diameter is obtained using equation (2) below.

$$T = \tau_a \cdot Z_P \quad \text{and} \quad Z_P = \frac{T}{\tau_a} \qquad \cdots\cdots\cdots\cdots (2)$$



T : Maximum torsion moment (N·mm)
τ_a : Permissible torsion stress of the spline shaft (49 N/mm²)
Z_P : Polar section modulus of the spline nut (mm³)
(see Table 3 on A3-17, Table 4 on A3-18, Table 5 on A3-19, and Table 6 on A3-20)

Reference: Section Modulus (Solid Circle)

$$Z_P = \frac{\pi \cdot d^3}{16}$$

Z_P : Section modulus (mm³)
d : Shaft outer diameter (mm)

When the Spline Shaft Simultaneously Receives a Bending Load and a Torsion Load

When the spline shaft of a ball spline receives a bending load and a torsion load simultaneously, calculate two separate spline shaft diameters: one for the equivalent bending moment (M_e) and the other for the equivalent torsion moment (T_e). Then, use the greater value as the spline shaft diameter.

Equivalent bending moment

$$M_e = \frac{M + \sqrt{M^2 + T^2}}{2} = \frac{M}{2}\left\{1 + \sqrt{1 + \left(\frac{T}{M}\right)^2}\right\} \quad \cdots\cdots (3)$$

$M_e = \sigma \cdot Z$

Equivalent torsion moment

$$T_e = \sqrt{M^2 + T^2} = M \cdot \sqrt{1 + \left(\frac{T}{M}\right)^2} \quad \cdots\cdots (4)$$

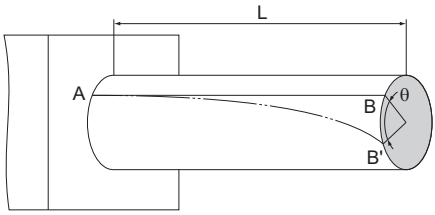
$T_e = \tau_a \cdot Z_p$

Torsional Rigidity of the Spline Shaft

The torsional rigidity of the spline shaft is expressed as a torsion angle per meter of shaft length. Its value should be limited within 1°/4.

$$\theta = 57.3 \times \frac{T \cdot L}{G \cdot I_P} \quad \cdots\cdots (5)$$

$$\text{Rigidity of the shaft} = \frac{\text{Torsion angle}}{\text{Unit length}} = \frac{\theta \cdot \ell}{L} < \frac{1°}{4}$$



θ : Torsion angle (°)
L : Spline shaft length (mm)
G : Transverse elastic modulus (7.9×10^4 N/mm^2)
ℓ : Unit length (1,000 mm)
I_p : Polar moment of inertia (mm^4)
(see Table 3 on A3-17, Table 4 on A3-18, Table 5 on A3-19, and Table 6 on A3-20)

Deflection and Deflection Angle of the Spline Shaft

The deflection and deflection angle of the ball spline shaft need to be calculated using equations that meet the relevant conditions. Table 1 and Table 2 represent these conditions and the corresponding equations.

Table 3 on **A3-17**, Table 4 on **A3-18**, Table 5 on **A3-19**, and Table 6 on **A3-20** show the section modulus of the spline shaft (Z) and the second moment of area (I). Using the Z and I values from the tables, the strength and displacement (deflection) of a typical ball spline within each model type can be obtained.

Table 1: Deflection and Deflection Angle Equations

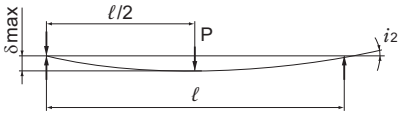
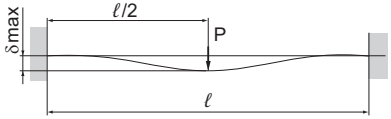
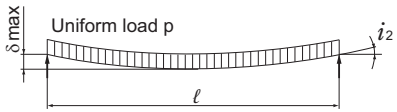
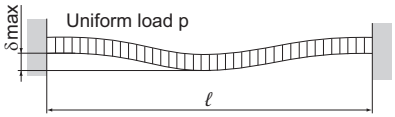
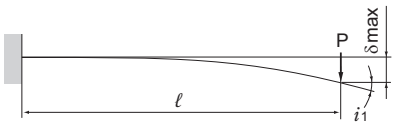
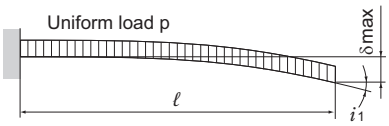
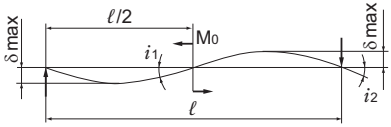
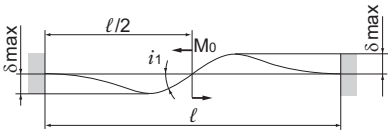
Support method	Operating conditions	Deflection equation	Deflection angle equation
Both ends free		$\delta_{max} = \frac{P\ell^3}{48EI}$	$i_1 = 0$ $i_2 = \frac{P\ell^2}{16EI}$
Both ends fastened		$\delta_{max} = \frac{P\ell^3}{192EI}$	$i_1 = 0$ $i_2 = 0$
Both ends free		$\delta_{max} = \frac{5p\ell^4}{384EI}$	$i_2 = \frac{p\ell^3}{24EI}$
Both ends fastened		$\delta_{max} = \frac{p\ell^4}{384EI}$	$i_2 = 0$

Table 2: Deflection and Deflection Angle Equations

Support method	Operating conditions	Deflection equation	Deflection angle equation
One end fastened		$\delta_{max} = \frac{P\ell^3}{3EI}$	$i_1 = \frac{P\ell^2}{2EI}$ $i_2 = 0$
One end fastened		$\delta_{max} = \frac{p\ell^4}{8EI}$	$i_1 = \frac{p\ell^3}{6EI}$ $i_2 = 0$
Both ends free		$\delta_{max} = \frac{\sqrt{3}M_0\ell^2}{216EI}$	$i_1 = \frac{M_0\ell}{12EI}$ $i_2 = \frac{M_0\ell}{24EI}$
Both ends fastened		$\delta_{max} = \frac{M_0\ell^2}{216EI}$	$i_1 = \frac{M_0\ell}{16EI}$ $i_2 = 0$

δ_{max}: Maximum deflection (mm)
M_0: Moment (N·mm)
ℓ: Span (mm)
I: Geometric moment of inertia (mm^4)
i_1: Deflection angle at loading point
i_2: Deflection angle at supporting point
P: Concentrated load (N)
p: Uniform load (N/mm)
E: Modulus of longitudinal elasticity 2.06×10^5 (N/mm^2)

Critical Speed of the Spline Shaft

When a ball spline shaft is used to transmit power while rotating, as the rotational speed of the shaft increases, the rotation cycle nears the natural frequency of the spline shaft. It may cause resonance and eventually result in loss of function. Therefore, the maximum rotational speed of the shaft must be limited to below the critical speed that does not cause resonance.
The critical speed of the spline shaft is obtained using equation (6).
(0.8 is multiplied as a safety factor.)

If the shaft's rotation cycle exceeds or nears the resonance point during operation, it is necessary to reconsider the spline shaft diameter.

● **Critical Speed**

$$N_c = \frac{60\lambda^2}{2\pi \cdot \ell_b^2} \cdot \sqrt{\frac{E \times 10^3 \cdot I}{\gamma \cdot A}} \times 0.8 \quad \cdots(6)$$

N_c : Critical speed (min^{-1})
ℓ_b : Distance between two mounting surfaces (mm)
E : Young's modulus (2.06×10^5 N/mm^2)
I : Minimum geometric moment of inertia of the shaft (mm^4)

$$I = \frac{\pi}{64} d^4 \quad \text{d: Minor diameter (mm)}$$

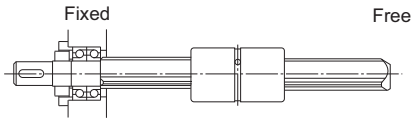
(see Table 10, Table 11, Table 12, and Table 13 on A3-26)

γ : Density (specific gravity) (7.85×10^{-6} kg/mm^3)

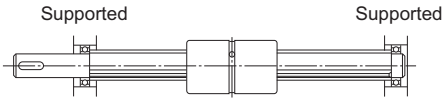
$$A = \frac{\pi}{4} d^2 \quad \text{d: Minor diameter (mm)}$$

(see Table 10, Table 11, Table 12, and Table 13 on A3-26)

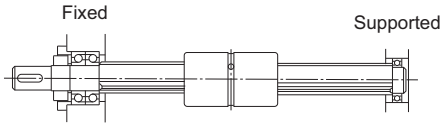
A : Spline shaft cross-sectional area (mm^2)
λ : Factor according to the mounting method
(1) Fixed - free $\lambda = 1.875$
(2) Supported - supported $\lambda = 3.142$
(3) Fixed - supported $\lambda = 3.927$
(4) Fixed - fixed $\lambda = 4.73$



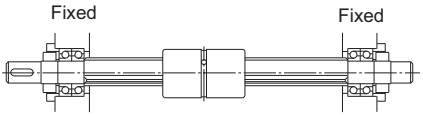
Fixed - free



Supported - supported



Fixed - supported



Fixed - fixed

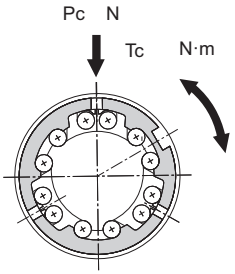
Applied Load

The load applied to the spline nut varies with the location of the center of gravity of the load, the position of the thrust, inertia generated by acceleration and deceleration during starts and stops, external forces, and torque. When selecting a ball spline, it is necessary to obtain the value of the applied load while taking these conditions into account.

● When a Torque Load and a Radial Load are Applied Simultaneously

When a torque load and a radial load are applied simultaneously, calculate the nominal life by obtaining the equivalent radial load using formula (7) below.

$$P_E = P_C + \frac{4 \cdot T_C \times 10^3}{i \cdot dp \cdot \cos\alpha} \quad \cdots\cdots\cdots (7)$$



P_E : Equivalent radial load (N)
P_C : Calculated radial load (N)
T_C : Calculated load torque (N·m)
$\cos\alpha$: Contact angle i = Number of rows of balls under a load

(Type LBSα = 45° i = 3 Type SLSα = 40° i = 3
Type LTα = 70° i = 2 (LT13 or smaller) Type LT-Xα = 65° i = 2
i = 3 (LT16 or greater))

dp : Ball center-to-center diameter (mm)
(see A3-26 Table 10, Table 11, Table 12, Table 13)

● When a Moment Load is Applied to a Single Nut or Two Nuts in Close Contact with Each Other

Calculate the nominal life by obtaining the equivalent radial load using formula (8) below.

$$P_u = K \cdot M \quad \cdots\cdots\cdots (8)$$

P_u : Equivalent radial load (with a moment applied) (N)
K : Equivalent Factors
(see A3-29 on Table 14, A3-30 on Table 15, A3-31 and Table 16 on Table 17)
M : Applied moment (N·mm)
However, M should be within the range of the static permissible moment.

● When a Moment Load and a Radial Load are Simultaneously Applied

Calculate the static safety factor and nominal life from the sum of the radial load and the equivalent radial load.

Static Safety Factor

When using a ball spline, a static safety factor derived from the basic static load rating and the calculated load in each direction must be considered. In particular, if the system starts and stops frequently, or if impact loads are applied, large moment loads or torque caused by overhanging loads may be applied to the system. When selecting a model, make sure that a sufficient static safety factor has been ensured for its maximum load (whether stationary or in motion). In addition, the static safety factor must be confirmed for each load direction using Formulas 9 and 10. For radial loads, you can obtain the static safety factor for the basic static load rating, and for moment loads, you can obtain the static safety factor for the basic static torque rating. Table 3 shows guideline values for the static safety factor.

● Static Safety Factor per Basic Static Torque Rating

$$f_s = \frac{f_T \cdot f_C \cdot C_{0T}}{T_{max}} \quad \cdots\cdots\cdots (9)$$

● Static Safety Factor per Basic Static Load Rating

$$f_s = \frac{f_T \cdot f_C \cdot C_0}{P_{max}} \quad \cdots\cdots\cdots (10)$$

f_s : Static safety factor
C_{0T} : Basic static torque rating[1] (N·m)
C_0 : Basic static load rating[2] (N)
T_{max} : Maximum load torque[3] (N·m)
P_{max} : Maximum applied load[3] (N)
f_T : Temperature factor (see A3-25 on Fig. 1)
f_C : Contact factor (see A3-25 on Table 8)

[1] The basic static torque rating is a static torque of a defined direction and size where the sum of the permanent deformation of the ball and that of the raceway at the contact area under maximum stress is 0.0001 times the ball diameter.
[2] The basic static load rating is a static load of a defined direction and size where the sum of the permanent deformation of the ball and that of the raceway at the contact area under maximum stress is 0.0001 times the ball diameter.
[3] The maximum values for T_C and P_C during a travel cycle, as given on B3-19, are applied to the maximum torque load T_{max} and the maximum load P_{max}.

Table 3: Guideline for the Static Safety Factor (f_s)

Load conditions[1]	Lower Limit of f_s
Without vibrations or impacts	3
With vibrations or impacts	5

[1] Vibrations and impacts are typically caused by factors such as acceleration and deceleration, sudden starting and stopping, vibrations and impacts from an external machine, and changes in processing power over time.

Nominal Life

A ball spline in motion under an external load receives repeated stress on its raceways and rolling elements. When the stress reaches the limit, the surface of the raceways and rolling elements flake in places due to rolling fatigue. This phenomenon is called flaking. The service life of the ball spline is the total travel distance until the first flaking occurs on any of the raceways or the rolling elements as a result of rolling fatigue of the material. The nominal life defined below is used as an estimate for the service life of a ball spline. Nominal life is the total travel distance that 90% of a group of identical ball splines independently operating under the same conditions can achieve without flaking.

Calculating the Nominal Life

The nominal life of a ball spline varies with types of loads applied during operation: torque load, radial load, and moment load. The corresponding nominal life values are obtained using the equations (11) to (15) below. (The basic load ratings in these loading directions are indicated in the specification table for the corresponding model number.)

● Calculating the Nominal Life

The nominal life of the THK ball spline is defined as 50 km. The nominal life (L_{10}) is calculated from the basic dynamic load rating (C) and the load acting on the ball spline (P_C) using the following formulas.

- When a torque load is applied

$$L_{10} = \left(\frac{C_T}{T_C}\right)^3 \times 50 \quad \cdots\cdots\cdots (11)$$

- When a radial load is applied

$$L_{10} = \left(\frac{C}{P_C}\right)^3 \times 50 \quad \cdots\cdots\cdots (12)$$

L_{10}	: Nominal life	(km)
C_T	: Basic dynamic torque rating	(N·m)
C	: Basic dynamic load rating	(N)
T_C	: Calculated torque applied	(N·m)
P_C	: Calculated radial load	(N)

Note: These nominal life formulas may not apply if the length of the stroke is less than or equal to twice the length of the ball spline nut.

When comparing the nominal life (L_{10}), you must take into account whether the basic dynamic load rating was defined based on 50 km or 100 km. Convert the basic dynamic load rating based on ISO 14728-1 as necessary.

ISO-regulated basic dynamic load rating conversion formula:

$$C_{100} = \frac{C_{50}}{1.26}$$

C_{50}	: Basic dynamic load rating based on a nominal life of 50 km
C_{100}	: Basic dynamic load rating based on a nominal life of 100 km

● Calculating the Modified Nominal Life

During use, a ball spline may be subjected to vibrations and shocks as well as fluctuating loads, which are difficult to detect. In addition, the operating temperature and having nuts arranged in close contact will significantly impact the service life. Taking these factors into account, the modified nominal life (L_{10m}) can be calculated according to the following formulas (13) and (14).

● Modified factor α

$$\alpha = \frac{f_T \cdot f_C}{f_W}$$

α : Modified factor
f_T : Temperature factor (see Fig. 1 on B3-23)
f_C : Contact factor (see Table 4 on B3-23)
f_W : Load factor (see Table 5 on B3-23)

● Modified nominal life L_{10m}

• When a torque load is applied

$$L_{10m} = \left(\alpha \times \frac{C_T}{T_C}\right)^3 \times 50 \quad \cdots\cdots\cdots (13)$$

• When a radial load is applied

$$L_{10m} = \left(\alpha \times \frac{C}{P_C}\right)^3 \times 50 \quad \cdots\cdots\cdots (14)$$

L_{10m} : Modified nominal life (km)
C_T : Basic dynamic torque rating (N·m)
C : Basic dynamic load rating (N)
T_C : Calculated torque applied (N·m)
P_C : Calculated radial load (N)

● Calculating the Service Life Time

When the nominal life (L_{10}) has been obtained in the equation above, if the stroke length and the number of reciprocations per minute are constant, the service life time is obtained using the equation (15) below.

$$L_h = \frac{L_{10} \times 10^3}{2 \times \ell_S \times n_1 \times 60} \quad \cdots\cdots\cdots (15)$$

L_h : Service life time (h)
ℓ_S : Stroke length (m)
n_1 : Number of reciprocations per minute (min^{-1})

■f_T: Temperature Factor

If the temperature of the environment surrounding the operating ball spline exceeds 100℃, take into account the adverse effect of the high temperature and multiply the basic load ratings by the temperature factor indicated in Fig. 1.
In addition, the ball spline must be of a high-temperature type.

Note: If the environmental temperature exceeds 80℃, high-temperature types of seal and retainer are required. Contact THK for details.

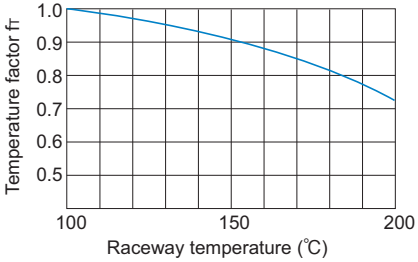


Fig. 1: Temperature Factor (f_T)

■f_C: Contact Factor

When multiple spline nuts are used in close contact with each other, their linear motion is affected by moments and mounting accuracy, making it difficult to achieve uniform load distribution. In such applications, multiply the basic load rating (C) and (C_0) by the corresponding contact factor in Table 4.

Note: If uneven load distribution is expected in a large machine, take into account the respective contact factor indicated in Table 4.

Table 4: Contact Factor (f_c)

Number of spline nuts in close contact with each other	Contact factor f_c
2	0.81
3	0.72
4	0.66
5	0.61
Normal use	1

■f_W: Load Factor

In general, reciprocating machines tend to experience vibrations or impacts during operation, and it is difficult to accurately determine the vibrations generated during high-speed operation and impacts during frequent starts and stops. When the actual load applied to a ball spline cannot be obtained, or when speed and vibrations have a significant influence, divide the basic dynamic load rating (C) by the corresponding load factor in Table 5, which has been empirically obtained.

Table 5: Load Factor (f_w)

Vibrations/ impacts	Speed (V)	f_w
Faint	Very low V ≦ 0.25 m/s	1 to 1.2
Weak	Slow 0.25 < V ≦ 1 m/s	1.2 to 1.5
Medium	Medium 1 < V ≦ 2 m/s	1.5 to 2
Strong	High V > 2 m/s	2 to 3.5

Calculating the Average Load

In cases where the load applied to each spline nut fluctuates under different conditions, such as an industrial robot advancing while holding a workpiece with its arm then retreating with its arm empty, or a machine tool handling various workpieces, it is necessary to calculate the service life of the spline nut while taking into account such fluctuating load conditions.

The average load (P_m) is a constant load under which the service life is equal to that of a ball spline operating with its spline nut receiving a fluctuating load due to varying conditions.

The following is the basic equation.

$$P_m = \sqrt[3]{\frac{1}{L} \cdot \sum_{n=1}^{n} (P_n^3 \cdot L_n)}$$

P_m	: Average load	(N)
P_n	: Varying load	(N)
L	: Total travel distance	(mm)
L_n	: Distance traveled under P_n	(mm)

- **With Stepwise Load Fluctuation**

$$P_m = \sqrt[3]{\frac{1}{L} (P_1^3 \cdot L_1 + P_2^3 \cdot L_2 \cdots\cdots + P_n^3 \cdot L_n)} \quad \cdots\cdots\cdots\cdots (16)$$

P_m	: Average load	(N)
P_n	: Varying load	(N)
L	: Total travel distance	(m)
L_n	: Distance traveled under load P_n	(m)

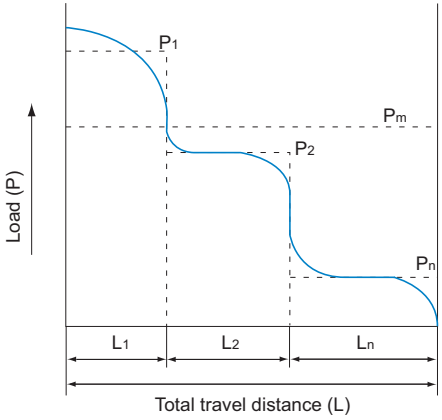


Fig. 2

● With Monotone Load Fluctuation

$$P_m \fallingdotseq \frac{1}{3}(P_{min} + 2 \cdot P_{max}) \cdots\cdots\cdots\cdots (17)$$

P_{min} : Minimum load (N)
P_{max} : Maximum load (N)

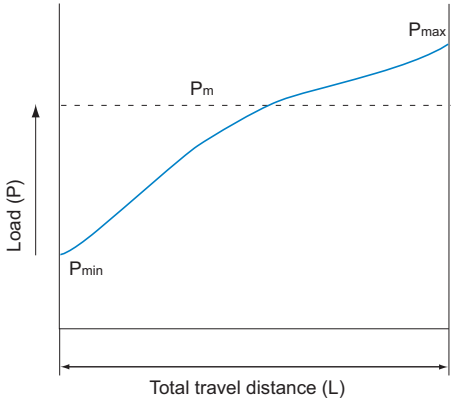
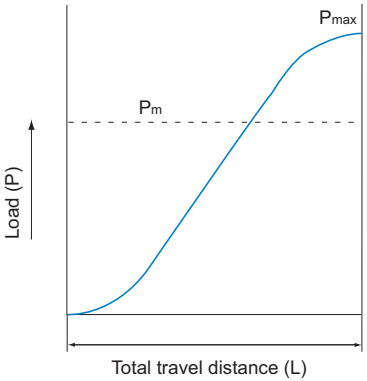


Fig. 3

● With Sinusoidal Load Fluctuation

(a) $P_m \fallingdotseq 0.65P_{max}$ ·····(18)

(b) $P_m \fallingdotseq 0.75P_{max}$ ·····(19)



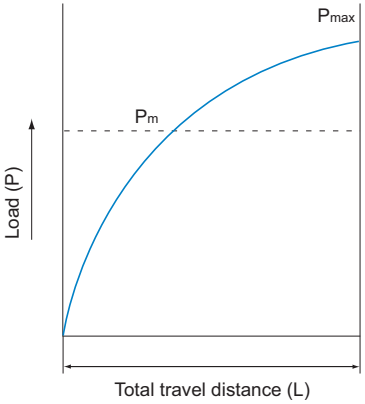


Fig. 4

Equivalent Factor

Table 14 on A3-29, Table 15 on A3-30, Table 16 and Table 17 on A3-31 show equivalent radial load factors calculated under a moment load.

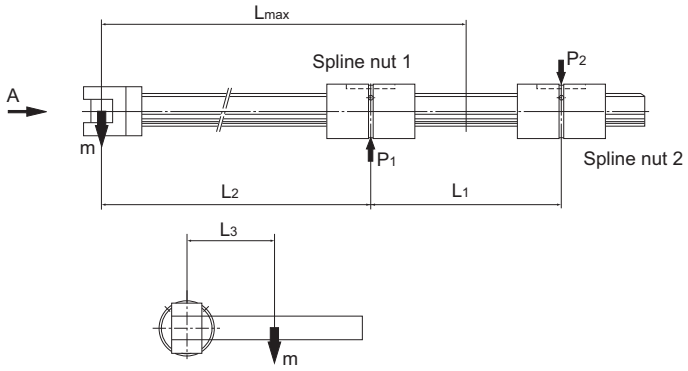
Example of Calculating the Service Life

● **Example of Calculation - 1**

An industrial robot arm (horizontal)

Operating Conditions

Mass applied to the arm end	m = 50 kg	Arm length at maximum stroke	L_{max} = 400 mm
Stroke	ℓ_S = 200 mm		L_2 = 325 mm
Spline nut mounting span (estimate)	L_1 = 150 mm		L_3 = 50 mm



View from arrow A

(The ball spline type is LBS in this example.)

Fig. 5

■Shaft Strength Calculation

Calculate the bending moment (M) and the torsion moment (T) applied on the shaft.

$M = m \times 9.8 \times L_{max} = 196{,}000$ N·mm

$T = m \times 9.8 \times L_3 = 24{,}500$ N·mm

Since the bending and torsion moments are applied simultaneously, obtain the corresponding bending moment (M_e) and torsion moment (T_e), and then determine the shaft diameter based on the greater value. From equations (3) and (4) on B3-15,

$$M_e = \frac{M + \sqrt{M^2+T^2}}{2} = 196{,}762.7 \text{ N} \cdot \text{mm}$$

$$T_e = \sqrt{M^2+T^2} = 197{,}525.3 \text{ N} \cdot \text{mm}$$

$M_e < T_e$

$\therefore T_e = \tau_a \times Z_P$ Hence,

$$Z_P = \frac{T_e}{\tau_a} = 4{,}031 \text{ mm}^3$$

Thus, judging from Table 4 on A3-18, the nominal shaft diameter that meets Z_P is at least 40 mm.

■Average Load P_m

Obtain an applied load value when the arm is extended to the maximum length (P_{max}), and another when the arm is contracted (P_{min}). Based on the values obtained, calculate the average load on the spline shaft nut.

$$P_{1max} = \frac{m \times 9.8(L_1+L_2)}{L_1} = 1{,}551.7\ N$$

$$P_{2max} = \frac{m \times 9.8 \times L_2}{L_1} = 1{,}061.7\ N$$

When the arm is contracted

$$P_{1min} = \frac{m \times 9.8 \times [(L_2-\ell_S) + L_1]}{L_1} = 898.3\ N$$

$$P_{2min} = \frac{m \times 9.8 \times (L_2-\ell_S)}{L_1} = 408.3\ N$$

As this load is monotonically varying as shown in the Fig. 3 on B3-25, calculate the average load using the equation (17) on B3-25.
The average load (P_{1m}) on spline nut 1

$$P_{1m} = \frac{1}{3}(P_{1min} + 2P_{1max}) = 1{,}333.9\ N$$

The average load (P_{2m}) on spline nut 2

$$P_{2m} = \frac{1}{3}(P_{2min} + 2P_{2max}) = 843.9\ N$$

Obtain the torque applied on one spline nut.

$$T = \frac{m \times 9.8 \times L_3}{2} = 12{,}250\ N \cdot mm$$

Since the radial load and the torque are simultaneously applied, calculate the equivalent radial load using equation (7) on B3-19.

$$P_{1E} = P_{1m} + \frac{4 \times T}{3 \times dp \times \cos\alpha} = 1{,}911.4\ N$$

$$P_{2E} = P_{2m} + \frac{4 \times T}{3 \times dp \times \cos\alpha} = 1{,}421.4\ N$$

■Nominal Life L_n

Based on the nominal life equation (12) on B3-21, each nominal life is obtained as follows.

$$\text{Nominal life of the spline nut } L_{10m1} = \left(\alpha \times \frac{C}{P_{1E}}\right)^3 \times 50 = 68{,}867.4\ km$$

$$\text{Nominal life of the spline nut } L_{10m2} = \left(\alpha \times \frac{C}{P_{2E}}\right)^3 \times 50 = 167{,}463.2\ km$$

$$\alpha = \frac{f_T \cdot f_C}{f_W}$$

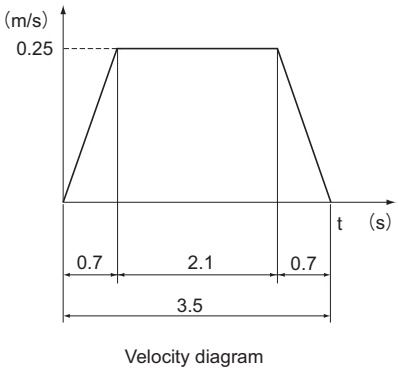
f_T: Temperature factor = 1 (from Fig. 1 on B**3-23**)
f_C: Contact factor = 1 (from Table 4 on B**3-23**)
f_W: Load factor = 1.5 (from Table 5 on B**3-23**)
C: Basic dynamic load rating = 31.9 kN (model LBS40)

Given the nominal life obtained for each spline nut above, the nominal life of the ball spline unit is equal to that of spline nut 1, which is 68,867.4 km.

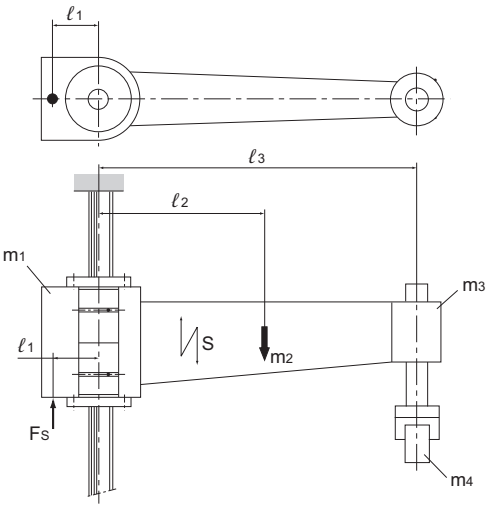
● **Example of Calculation - 2**

Operating Conditions
Thrust position: F_S
Stroke velocity: V_{max} = 0.25 m/s
Acceleration: a = 0.36 m/s^2
(from the respective velocity diagram)
Stroke: S = 700 mm
Housing mass: m_1 = 30 kg
Arm mass: m_2 = 20 kg
Head mass: m_3 = 15 kg
Work mass: m_4 = 12 kg

Distance from the thrust position to each mass
ℓ_1 = 200 mm ℓ_2 = 500 mm
ℓ_3 = 1,276 mm
Cycle (1 cycle: 30 s)
1. Descent (3.5 s) 2. Dwell (1 s): with a workpiece
3. Ascent (3.5 s) 4. Dwell (7 s)
5. Descent (3.5 s) 6. Dwell (1 s): without a workpiece
7. Ascent (3.5 s) 8. Dwell (7 s)



Velocity diagram



(The Ball Spline type is LBF in this example.)

Fig. 6

■Shaft Strength Calculation

Calculate the shaft strength while assuming the shaft diameter to be 60 mm (with double spline nuts in contact with each other).

■Calculating the Moment (M_n) Applying on the Spline Nut during Acceleration, Uniform Motion and Deceleration with Different Masses (m_n)

Applied moment during deceleration: M_1

$$M_1 = m_n \times 9.8 \left(1 \pm \frac{a}{g}\right) \times \ell_n \quad \cdots\cdots\cdots(a)$$

Applied moment during uniform motion: M_2

$$M_2 = m_n \times 9.8 \times \ell_n \quad \cdots\cdots\cdots(b)$$

Applied moment during deceleration: M_3

$$M_3 = m_n \times 9.8 \left(1 \pm \frac{a}{g}\right) \times \ell_n \quad \cdots\cdots\cdots(c)$$

m_n: Mass (kg)
a : Acceleration (m/s^2)
g : Gravitational acceleration (m/s^2)
ℓ_n : Offset from each loading point to the thrust center (mm)
Assume:

$$A = \left(1 + \frac{a}{g}\right), \quad B = \left(1 - \frac{a}{g}\right)$$

- During descent

  From equation (c), during acceleration

  $M_{m1} = m_1 \times 9.8 \times B \times \ell_1 + m_2 \times 9.8 \times B \times (\ell_1+\ell_2) + m_3 \times 9.8 \times B \times (\ell_1+\ell_3)$
  $= 398{,}105.01$ N·mm

  From equation (b), during uniform motion

  $M_{m2} = m_1 \times 9.8 \times \ell_1 + m_2 \times 9.8 \times (\ell_1+\ell_2) + m_3 \times 9.8 \times (\ell_1+\ell_3)$
  $= 412{,}972$ N·mm

  From equation (a), during deceleration

  $M_{m3} = m_1 \times 9.8 \times A \times \ell_1 + m_2 \times 9.8 \times A \times (\ell_1+\ell_2) + m_3 \times 9.8 \times A \times (\ell_1+\ell_3)$
  $= 427{,}838.99$ N·mm

- During ascent

  From equation (a), during acceleration

  $M_{m1}' = m_1 \times 9.8 \times A \times \ell_1 + m_2 \times 9.8 \times A \times (\ell_1+\ell_2) + m_3 \times 9.8 \times A \times (\ell_1+\ell_3)$
  $= 427{,}838.99$ N·mm

  From equation (b), during uniform motion

  $M_{m2}' = m_1 \times 9.8 \times \ell_1 + m_2 \times 9.8 \times (\ell_1+\ell_2) + m_3 \times (\ell_1+\ell_3)$
  $= 412{,}972$ N·mm

From equation (c), during deceleration

$M_{m3}' = m_1 \times 9.8 \times B \times \ell_1 + m_2 \times 9.8 \times B \times (\ell_1+\ell_2) + m_3 \times 9.8 \times B \times (\ell_1+\ell_3)$

$= 398{,}105.01$ N·mm

- During descent (with a workpiece loaded)

From equation (c), during acceleration

$M_{m1}'' = M_{m1} + m_4 \times 9.8 \times B \times (\ell_1 + \ell_3)$

$= 565{,}433.83$ N·mm

From equation (b), during uniform motion

$M_{m2}'' = M_{m2} + m_4 \times 9.8 \times (\ell_1 + \ell_3)$

$= 586{,}549.6$ N·mm

From equation (a), during deceleration

$M_{m3}'' = M_{m3} + m_4 \times 9.8 \times A \times (\ell_1 + \ell_3)$

$= 607{,}665.37$ N·mm

- During ascent (with a workpiece loaded)

From equation (a), during acceleration

$M_{m1}''' = M_{m1}' + m_4 \times 9.8 \times A \times (\ell_1+\ell_3)$

$= 607{,}665.37$ N·mm

From equation (b), during uniform motion

$M_{m2}''' = M_{m2}' + m_4 \times 9.8 \times (\ell_1+\ell_3)$

$= 586{,}549.6$ N·mm

From equation (c), during deceleration

$M_{m3}''' = M_{m3}' + m_4 \times 9.8 \times B \times (\ell_1+\ell_3)$

$= 565{,}433.83$ N·mm

$\therefore M_1 = M_{m1} = M_{m3}' = 398{,}105.01$ N·mm

$M_2 = M_{m2} = M_{m2}' = 412{,}972$ N·mm

$M_3 = M_{m3} = M_{m1}' = 427{,}838.99$ N·mm

$M_1' = M_{m1}'' = M_{m3}''' = 565{,}433.83$ N·mm

$M_2' = M_{m2}'' = M_{m2}''' = 586{,}549.6$ N·mm

$M_3' = M_{m3}'' = M_{m1}''' = 607{,}665.37$ N·mm

■Calculating the Equivalent Radial Load Considered to be Applied to the Spline Nut with Different Moments

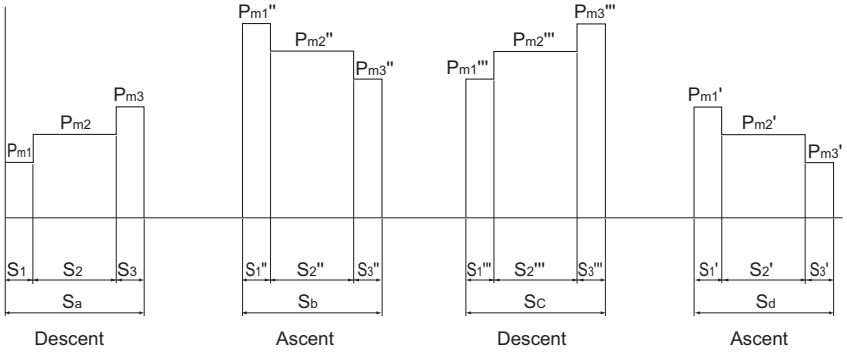
Relational expression between moment M_n and P_n

$$P_n = M_n \times K \quad \cdots\cdots\cdots(d)$$

P_n : Equivalent radial load (N)
M_n : Applied moment (N·mm)
K : Equivalent factor
(from Table 15 to A**3-30**)
(If two spline nuts of LBF60 contact with each other, K = 0.013)

Calculate the equivalent radial load with different applied moments using equation (d).

$P_{m1} = P_{m3}' = M_1 \times 0.013 \fallingdotseq 5175.4$ N
$P_{m2} = P_{m2}' = M_2 \times 0.013 \fallingdotseq 5368.6$ N
$P_{m3} = P_{m1}' = M_3 \times 0.013 \fallingdotseq 5561.9$ N
$P_{m1}'' = P_{m3}''' = M_1' \times 0.013 \fallingdotseq 7350.7$ N
$P_{m2}'' = P_{m2}''' = M_2' \times 0.013 \fallingdotseq 7625.2$ N
$P_{m3}'' = P_{m1}''' = M_3' \times 0.013 \fallingdotseq 7899.7$ N



$P_1 = P_{m1} = P_{m3}' \fallingdotseq 5175.4$ N
$P_2 = P_{m2} = P_{m2}' \fallingdotseq 5368.6$ N
$P_3 = P_{m3} = P_{m1}' \fallingdotseq 5561.9$ N

$P_4 = P_{m1}'' = P_{m3}''' \fallingdotseq 7350.7$ N
$P_5 = P_{m2}'' = P_{m2}''' \fallingdotseq 7625.2$ N
$P_6 = P_{m3}'' = P_{m1}''' \fallingdotseq 7899.7$ N

$S = S_a = S_b = S_c = S_d = 700$ mm
$S_1 = S_1 = S_1' = S_1'' = S_1''' = 87.5$ mm
$S_2 = S_2 = S_2' = S_2'' = S_2''' = 525$ mm
$S_3 = S_3 = S_3' = S_3'' = S_3''' = 87.5$ mm

■Calculating the Average Load P_m

Using equation (16) on B**3-24**,

$$P_m = \sqrt[3]{\frac{1}{4 \times S}\left[2\{(P_1^3 \times S_1) + (P_2^3 \times S_2) + (P_3^3 \times S_3)\} + 2\{(P_4^3 \times S_3) + (P_5^3 \times S_2) + (P_6^3 \times S_1)\}\right]}$$
$$\fallingdotseq 6{,}689.5 \text{ N}$$

■Calculating the Rated Life L from the Average Load

Using equation (14) on B**3-22**,

$$L_{10m} = \left(\alpha \times \frac{C}{P_m}\right)^3 \times 50$$
$$= 7{,}630 \text{ km}$$

$$\alpha = \frac{f_T \cdot f_C}{f_W}$$

f_T : Temperature factor = 1
(from Fig. 1 on B**3-23**)
f_C : Contact factor = 0.81
(from Table 4 on B**3-23**)
f_W : Load factor = 1.5
(from Table 5 on B**3-23**)
C : Basic dynamic load rating = 66.2 kN
(Model LBF60)

Given the result above, the nominal life of Model LBF60 with double spline nuts used in close contact with each other is 7,630 km.