NB

NB LINEAR SYSTEM

The NB linear system is a linear motion mechanism which utilizes the recirculating movement of ball or roller elements to provide smooth and accurate linear travel. NB offers a wide range of linear motion products that may contribute to the size and weight reduction of machinery and other equipment, while providing dependable performance in high-precision equipment.

ADVANTAGES

Low Friction and Excellent Response:

The dynamic friction of rolling ball or roller elements is substantially lower than that of full-face surface sliding friction. Since the difference between rolling dynamic and static friction is small, motion response is excellent and results in superior dependable movement. This also allows for easy fabrication of mechanisms requiring precise positioning or high-speed acceleration.

High Precision and Smooth Movement:

The NB linear system is designed for smooth rolling movements. The rolling element's raceway contact surface is finished through high-precision grinding. The recirculating movement of the rolling elements allow for continuous high-precision linear movement without clearance.

High Load Capacity and Long Travel Life:

Although the NB linear system is designed to be compact, the use of large rolling elements and machined raceway surface results in high load capacity and long travel life.

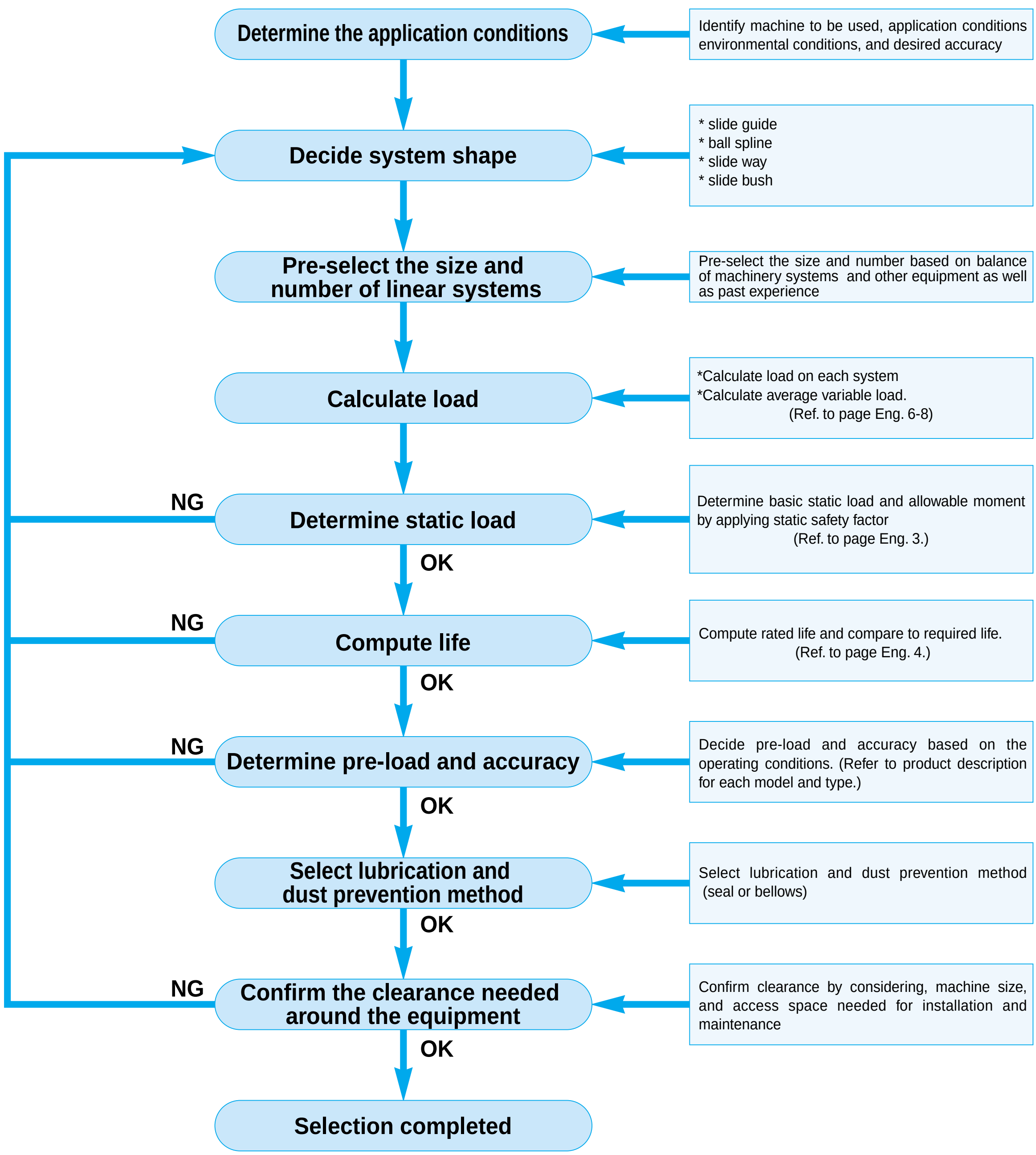
Ease of Installation:

The NB linear system shortens machining and assembly time when compared with that of a full face surface sliding bearing system.

Variety of Models:

A wide variety of models and types of NB systems are available for just about any shape or material option. This permits the selection of the best appropriate linear component for any application.

PROCESS FOR SELECTING NB LINEAR SYSTEM



ALLOWABLE LOAD

Load and Moment:

A load may be exerted to a linear system as depicted in Figure 1-1. In addition, a moment may be applied to a slide guide. Each type of load addressed by NB is described as follows.

Basic Static Load Rating and Allowable Static Moment:

Under excess or impact load conditions applied to a linear system while it is stationary or moving slowly, a permanent deformation occurs on either the rolling surface or the rolling elements. When this deformation exceeds a predictable level, it becomes a source of vibration and acoustic noise during operation and will also result in rough motion and shortens life. To prevent this permanent deformation and deterioration in movement accuracy, a basic static load rating (Co) is given as an allowable load. This basic static load rating is defined as the static load that results in the maximum allowable stress at the center of the contact surface between the rolling elements and the rolling surface. The sum of the permanent deformation of the rolling elements and that of the rolling surface is 1/10,000 the diameter of the rolling elements. In linear systems, a moment may also be present when applied in addition to the static load. The allowable static moment is defined by M_P, M_Y, and M_R, which are illustrated in Figure 1-1.

Allowable Load and Static Safety Factor:

The basic static load rating and allowable static moment define the maximum static load in each direction. These maximum static loads are not necessarily applicable depending on the operating conditions, the mounting accuracy, and the required motion accuracy. Therefore, an allowable load with a safety factor that covers these factors must be obtained. In general, the minimum static safety factor is based on the values as listed in Table 1-1.

Allowable load

$$P_{max.} \leqq C_o/f_s \cdots\cdots\cdots\cdots (1)$$

Allowable moment

$$M_{max.} \leqq (M_P, M_Y, M_R)/f_s \cdots\cdots\cdots\cdots (2)$$

fs : static safety factor　Co : basic static load rating(N)
Pmax. : allowable load(N)　M_P,M_R,M_Y : allowable static moment(N･m)
Mmax. : allowable moment(N･m)

Figure 1-1 Load and Moment

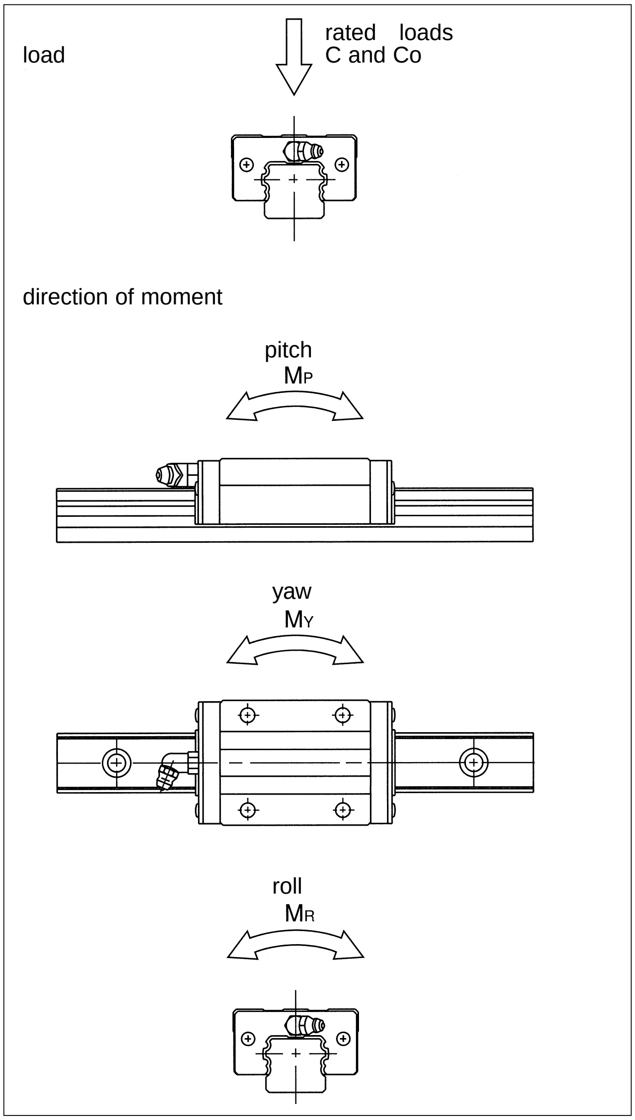


Table 1-1 Minimum Static Safety Factor

operating conditions	static safety factor
normal	1～2
smooth motion required	2～4
vibration/impact loading	3～5

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LIFE

Life of a Linear System:

If a linear system reciprocates when it is under a load, a continuous stress acts on it, ultimately causing flaking of its rolling elements and/or rolling surfaces due to material fatigue, making it inoperable. The distance a linear system travels before this flaking condition first occurs is called the life of the system. A linear system can also become inoperable due to sintering, cracking, pitting, or rusting. These factors are differentiated from those affecting the life because they are related to installation accuracy, operating environment, or the selected lubrication method of the installer.

Rated Life:

Even when two linear systems are manufactured at the same time, have the same part number, and are used under identical conditions, their lifetimes can differ due to differences in their fatigue failure characteristics. This prevents determining the life of any particular linear system. Therefore, the rated life is determined statistically and is defined as the distance 90% of linear systems travel before experiencing flaking.

Rated Basic Dynamic Load and Rated Basic Dynamic Torque:

The life of a linear system is expressed in terms of the distance traveled. Therefore, the life of a linear system is calculated using the allowable load that corresponds to a certain distance traveled. This allowable load is a measure of the system's performance relative to the applied load and is called the rated basic dynamic load. It is defined as a constant-direction load with a magnitude corresponding to a life of 50×10^3 m. In some cases or linear systems, the basic dynamic load rating may vary depending on the direction of the applied force. In the NB Linear System catalog, the value of the basic dynamic load rating is assumed when a force is applied from directly above and is indicated in the dimension tables. For ball splines, the linear motion may involve torque loading, so the basic dynamic torque rating is defined in a similar fashion.

Rated Life Estimation:

The rated lifetime estimation depends on the type of rolling element used. Both Equations (3) and (4) are used for ball and roller elements respectively. In cases when torque loading is applied, Equation (5) is to be used.

When a ball is used as the rolling element,

$$L = \left(\frac{C}{P}\right)^3 \cdot 50 \quad \cdots\cdots(3)$$

When a roller is used as the rolling element,

$$L = \left(\frac{C}{P}\right)^{10/3} \cdot 50 \quad \cdots\cdots(4)$$

When torque loading is applied,

$$L = \left(\frac{C_T}{T}\right)^3 \cdot 50 \quad \cdots\cdots(5)$$

L : rated life(km)　C : basic dynamic load rating(N)
P : applied load(N)　C_T : basic dynamic torque rating(N・m)
T : applied torque(N・m)

Numerous variables, such as guide rail accuracy, mounting conditions, operating conditions, vibration and shock while under linear motion affect an actual application. Therefore, calculating the actual applied load accurately is extremely difficult. In general, the calculation is simplified by using coefficients representing these effects. These coefficients include hardness (f_H), temperature (f_T), contact (f_C), and applied load (f_W). By using these coefficients, Equations (3) ～ (5) can be expressed by Equations (6) ～ (8).

When a ball is used as the rolling element,

$$L = \left(\frac{f_H \cdot f_T \cdot f_C}{f_W} \cdot \frac{C}{P}\right)^3 \cdot 50 \quad \cdots\cdots(6)$$

When a roller is used as the rolling element,

$$L = \left(\frac{f_H \cdot f_T \cdot f_C}{f_W} \cdot \frac{C}{P}\right)^{10/3} \cdot 50 \quad \cdots\cdots(7)$$

When torque loading is applied,

$$L = \left(\frac{f_H \cdot f_T \cdot f_C}{f_W} \cdot \frac{C_T}{T}\right)^3 \cdot 50 \quad \cdots\cdots(8)$$

L : rated life(km)　f_H : hardness coefficient
f_C : contact coefficient　f_W : applied load coefficient
P : applied load(N)　C : basic dynamic load rating (N)
C_T : basic dynamic torque rating(N・m)　T : applied torque(N・m)

If the distance traveled per unit time is known, the life can be expressed in terms of time, which may be easier to understand. The relationship between the stroke distance, the stroke frequency per minuit, and the life time is expressed by Equation (9)

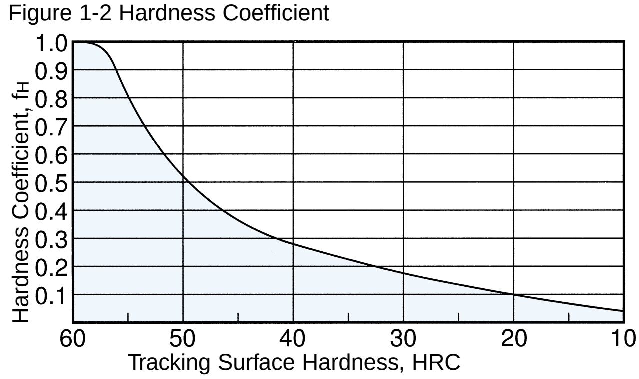
$$L_h = \frac{L \cdot 10^3}{2 \cdot \ell s \cdot n1 \cdot 60} \quad \cdots\cdots\cdots (9)$$

L_h : life time(hr) ℓs : stroke distance(m)
n1 : stroke frequency per min.(cpm)

Hardness Coefficient (f_H):

In a linear system, the guide rail serves the same purpose as an inner race of a ball bearing. Therefore, the hardness of the guide rail plays an important role in determining the rated load. If the surface hardness is less than HRC58, the rated load is reduced. NB uses an advanced heat treatment method to maintain an appropriate level of hardness. However, if guide rails with inadequate hardness must be used, the rated load must be re-calibrated based on the hardness coefficients given in Figure 1-2.

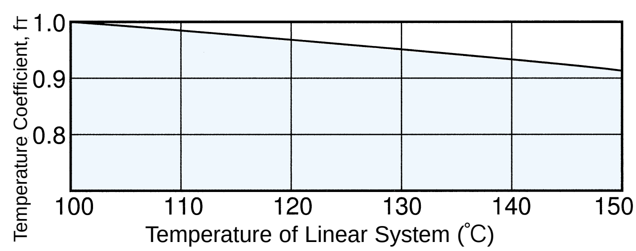
Figure 1-2 Hardness Coefficient



Temperature Coefficient (f_T):

NB linear systems are heat treated to reduce the amount of wear. Therefore, if the operating temperature exceeds 100 ˚C, hardness is reduced and the life of the system is shortened. The variation in hardness with temperature is shown in Figure 1-3.

Figure 1-3 Temperature Coefficient



Contact Coefficient (f_C):

When two or more linear systems are used in contact with each other, the variation in each system and the accuracy of the mounting surfaces must be taken into consideration. In general, the coefficient values given in Table 1-2 should be used to compute the life.

Table 1-2 Contact Coefficient

number of linear systems in contact and applied to a single shaft	contact coefficient f_C
1	1.00
2	0.81
3	0.72
4	0.66
5	0.61

Applied Load Coefficient (f_W):

When computing the applied load, the weight of the mass, inertial force, moment resulting from the motion, and the variation with time should be accurately estimated. However, it is very difficult to accurately estimate the applied load due to the existence of numerous variables, including the start/stop conditions of the reciprocating motion and of the shock/vibration. Estimation is simplified by using the values given in Table 1-3.

Table 1-3 Applied Load Coefficient

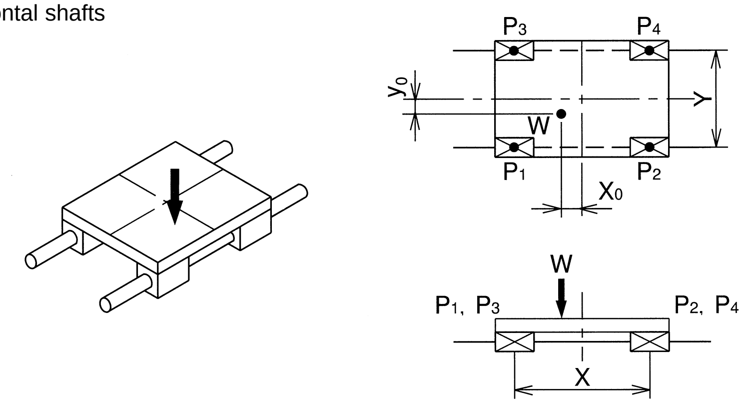
operating condition		applied load coefficient, f_W
loading condition	velocity	
no shock/vibration	15 m/min or less	1.0～1.5
low shock/vibration	60 m/min or less	1.5～2.0
high shock/vibration	60 m/min or more	2.0～3.5

Method for Determining Applied Load:

Typical cases that linear systems are set and the equations for determining the applied load for each case example are summarized in Table 1-4.

W : applied load (N) P_1 - P_4 : load applied to linear system (N) X,Y : linear system span (mm) x, y, ℓ : distance to load applied or to working center of gravity (mm) g : gravitational acceleration (9.8 x 10^3 mm/s^2) V : velocity (mm/s) t_1 : duration of acceleration (sec) t_3 : duration of deceleration (sec)

Table 1-4 Method for Determining Applied Load (1)

<table>
<tr><th></th><th>condition</th><th>applied load computation formula</th></tr>
<tr><td rowspan="3">under static conditions or constant velocity motion</td><td>2 horizontal shafts
</td><td rowspan="3">$P_1=\frac{1}{4}W+\frac{X_0}{2X}W+\frac{y_0}{2Y}W$
$P_2=\frac{1}{4}W-\frac{X_0}{2X}W+\frac{y_0}{2Y}W$
$P_3=\frac{1}{4}W+\frac{X_0}{2X}W-\frac{y_0}{2Y}W$
$P_4=\frac{1}{4}W-\frac{X_0}{2X}W-\frac{y_0}{2Y}W$

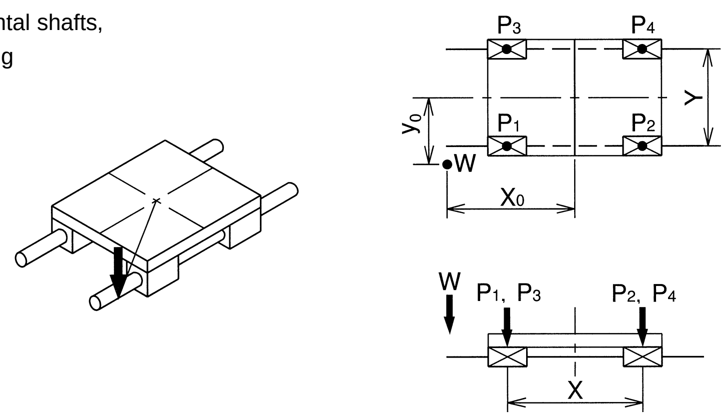
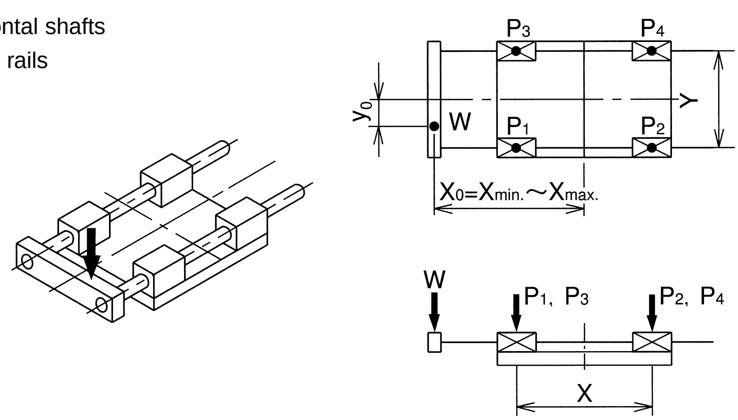
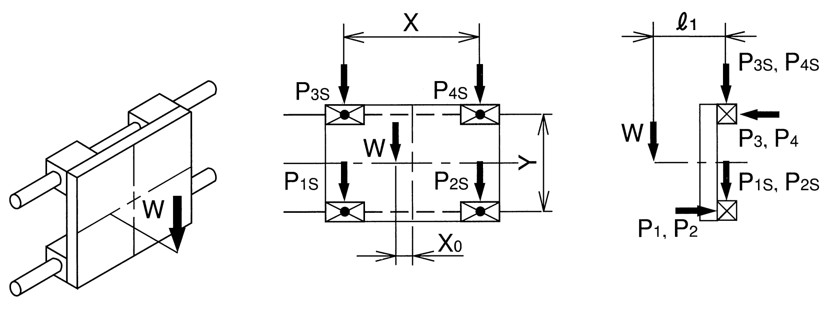
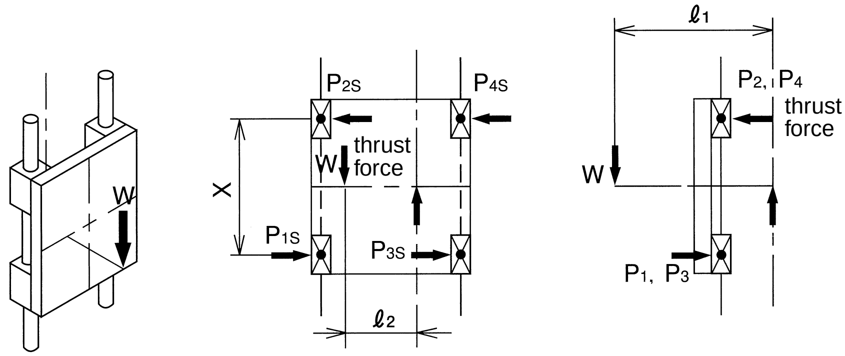
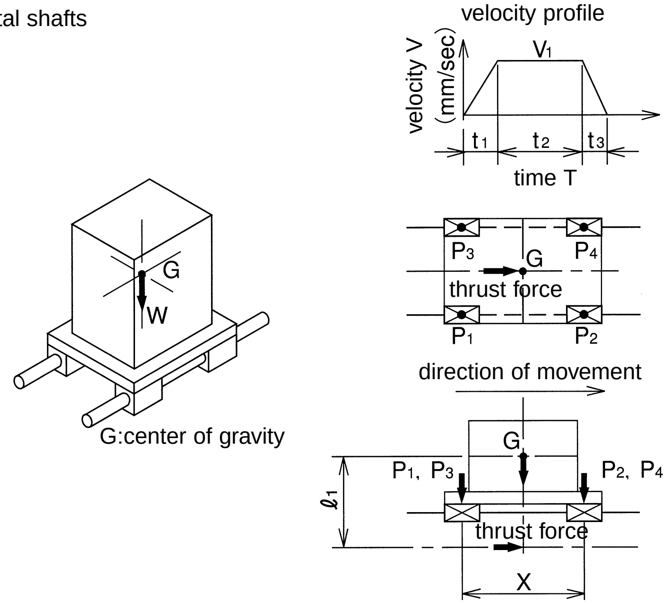
Note: If the calculation results in a negative value, the loading direction is in the opposite direction, but with the same computed magnitude.</td></tr>
<tr><td>2 horizontal shafts,
over-hang
</td></tr>
<tr><td>2 horizontal shafts
moving rails
</td></tr>
</table>

Table 1-5 Method for Determining Applied Load (2)

	condition	applied load computation formula
under static conditions or constant velocity motion	2 vertical/side shafts  	$P_1=P_2=P_3=P_4=\frac{\ell_1}{2Y}W$ $P_{1S}=P_{3S}=\frac{1}{4}W+\frac{X_0}{2X}W$ $P_{2S}=P_{4S}=\frac{1}{4}W-\frac{X_0}{2X}W$
	2 vertical shafts  	$P_1=P_2=P_3=P_4=\frac{\ell_1}{2X}W$ $P_{1S}=P_{2S}=P_{3S}=P_{4S}=\frac{\ell_2}{2X}W$
under constant acceleration conditions	2 horizontal shafts  	under acceleration $P_1=P_3=\frac{1}{4}W\left(1+\frac{2V_1\ell_1}{gt_1X}\right)$ $P_2=P_4=\frac{1}{4}W\left(1-\frac{2V_1\ell_1}{gt_1X}\right)$ under deceleration $P_1=P_3=\frac{1}{4}W\left(1-\frac{2V_1\ell_1}{gt_3X}\right)$ $P_2=P_4=\frac{1}{4}W\left(1+\frac{2V_1\ell_1}{gt_3X}\right)$ under constant velocity motion $P_1=P_2=P_3=P_4=\frac{1}{4}W$ ※g:gravitational acceleration (9.8×10^3mm/sec^2)

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Average Applied Load:

The load applied to a linear system generally varies with the distance traveled depending on how the system is used. This includes the start/stop processes of the reciprocating motion. The average applied load is used to compute the life corresponding to the actual application conditions.

1. When the load varies in a step manner with the distance traveled (Figure 1-4) and ℓ_1 is the distance traveled under load P_1, ℓ_2 is the distance traveled under load P_2, and ℓ_n is the distance traveled under load P_n, the average applied load, P_m is obtained by the following equation

$$\mathbf{Pm} = \sqrt[3]{\frac{1}{\ell}(P_1^3\ell_1 + P_2^3\ell_2 \cdots + P_n^3\ell_n)} \cdots\cdots \ (4)$$

Pm : average applied load (N)
ℓ : total distance traveled (m)

2. When the applied load varies linearly with the distance traveled (Figure 1-5), the average applied load, Pm is approximated by the following equation.

$$\mathbf{Pm} \fallingdotseq \frac{1}{3}(P_{min} + 2P_{max}) \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots \ (5)$$

Pm : minimum applied load (N)
Pmax : maximum applied load (N)

3. When the applied load draws a sine-curve as shown by Figures 1-6 (a) and (b), the average applied load, Pm, is approximated by the following equations.

$$\text{Figure 1-6(a)} \quad P_m \fallingdotseq 0.65P_{max} \cdots\cdots\cdots\cdots\cdots \ (6)$$

$$\text{Figure 1-6(b)} \quad P_m \fallingdotseq 0.75P_{max} \cdots\cdots\cdots\cdots\cdots \ (7)$$

Figure 1-4 Applied Load Varies Stepwise

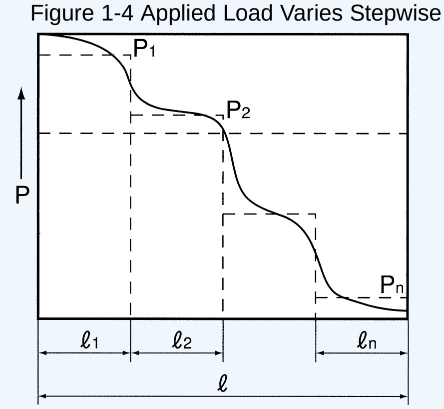


Figure 1-5 Applied Load Varies Linearly

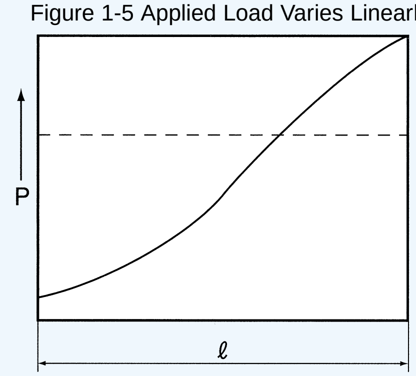
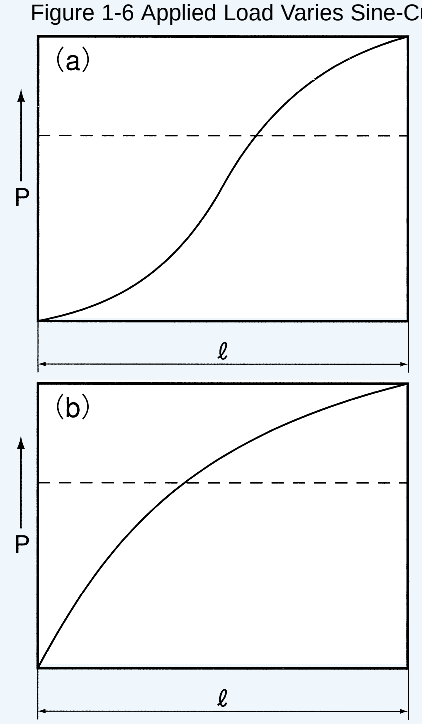


Figure 1-6 Applied Load Varies Sine-Curve



RIGIDITY AND PRE-LOAD

Effect of Pre-load and Rigidity:

The rigidity of a linear system must be taken into consideration when it is to be used in high-precision positioning devices or high-precision machinery. Pre-loaded slide guides and ball splines, which use a ball as the rolling element, are available upon request to meet the need for greater rigidity. If a force is applied to the ball elements without a pre-load, an elastic deformation proportional to the applied force to the 2/3 power will result.

$$\delta \propto W^{2/3} \quad \cdots\cdots (8)$$

δ : elastic deformation W : applied force

Therefore, the elastic deformation is relatively large during the initial loading stage, however then becomes smaller as the load increases.

The contact angles for all of the ball elements in SGL slide guides are the same as shown in Figure 1-7. Therefore, if the pre-load (P_1) applied results in an elastic deformation of $\delta 1$, the deformation will vary linearly with the applied load until the elastic deformation of the ball element on the other side cross the race becomes zero, as depicted in Figure 1-8. This permits the determination of the deformation of linear systems. The ratio between the applied load and the elastic deformation is defined as the rigidity of the system. Equation (9) can be used to determine the elastic deformation with an applied load of up to 2.8 times the pre-load.

$$2\delta_1 = 2kP_1^{2/3} = kP_2^{2/3} \qquad P_2 = 2\sqrt{2}\cdot P_1 \quad \cdots\cdots (9)$$

k : constant P_1 : pre-load
P_2 : applied load when the pre-load becomes 0

Contact NB for further information on rigidity.

Types of Pre-Load and its Specification:

Pre-load is categorized into three primary ranges: normal, light, and medium. At NB, pre-load is applied by installing rolling elements that are slightly larger than normal. Therefore, the specification of the pre-load is expressed by a negative gap value.

Figure 1-7 Contact Structure of SGL Slide Guide

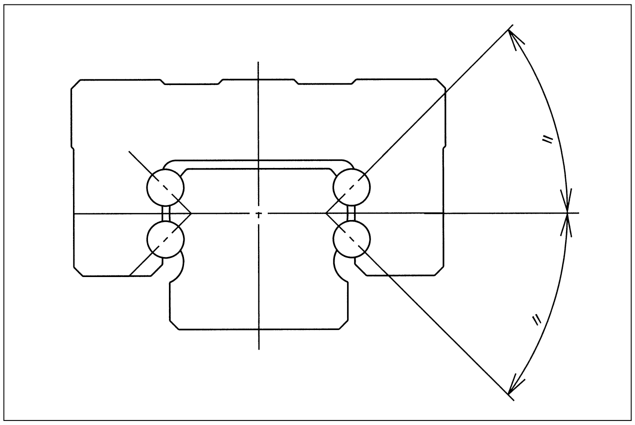
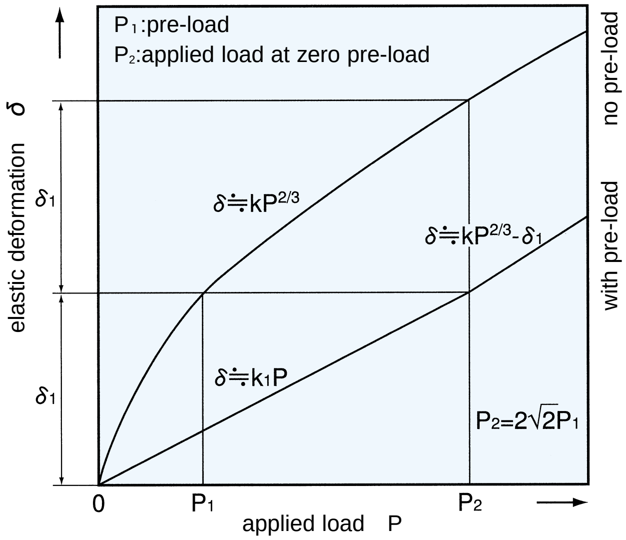


Figure 1-8 Applied Load vs. Elastic Deformation of Steel Balls



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FRICTIONAL RESISTANCE AND REQUIRED THRUST

The static friction of a linear system is extremely low. Since the difference between the static and dynamic friction is marginal, stable motion can be achieved from low to high velocity. The frictional resistance (required thrust) can be obtained from the load and the seal resistance unique to each type of system using the following equation:

$$F = \mu \cdot W + f \quad \cdots\cdots (10)$$

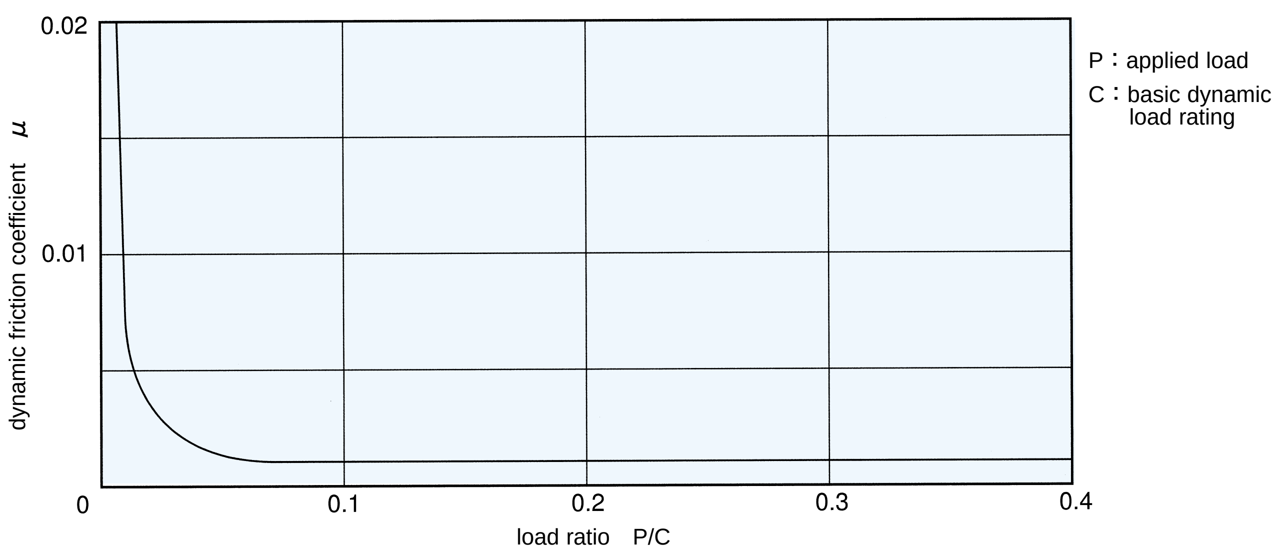
F : frictional resistance(N) μ : dynamic frictional coefficient
W : applied load(N) f : seal resistance(N)

The dynamic friction coefficient varies with the applied load, pre-load, viscosity of the lubricant, and other factors. However, the values given in Table 1-6 are used for the normal loading condition (20% of basic dynamic load rating) without any pre-load.
The seal resistance depends on the seal-lip condition as well as on the condition of the lubricant and does not change proportionally with the applied load, which commonly is expressed by a constant value of 2 to 5 N.

Table 1-6 Dynamic Friction Coefficient

type	major types	dynamic friction coefficient(μ)
slide guide	SGL・SGW	0.002～0.003
	SEB	0.004～0.006
	SER	0.004～0.006
ball spline	SSP	0.004～0.006
rotary ball spline	SPR	0.004～0.006
slide bush	GM・SM KB・SW	0.002～0.003
slide unit	SMA・SME	0.002～0.003
stroke bush	SR	0.0006～0.0012
slide rotary bush	RK	0.002～0.003
slide way	SV・SVW	0.001～0.003
slide table	SVT・SYT	0.001～0.003

Figure 1-9 Applied Load vs. Dynamic Friction Coefficient



OPERATING ENVIRONMENT

Temperature Range:

NB linear systems are heat-treated in order to harden the surface. Because of this, if the temperature of the system exceeds 100℃, the hardness and rated load will be reduced (refer to page Eng. 5, hardness coefficient). If resin is used in any one of the components, the system cannot be used in a high-temperature environment. The recommended operating temperature ranges for each type of linear system are listed in Table 1-7.

Table 1-7 Material Type and Recommended Temperature Range

component material	includes resin	steel	stainless	other
operating temperature range	−20℃ ～80℃	−20℃ ～110℃	−20℃ ～140℃	
slide guide	SEB-A/SEBS-B/SGL/SGW	SER	SEBS-BM/SERS	
ball spline	SSP/SSPF/SSPB			
rotary ball spline	SPR			
slide bush	GM/SM G/KB G/ SW G/SMS G/ KBS G/SWS G	SM/KB/SW	SMS/KBS/SWS	
slide unit	SMA G/AK G/RB/RBW/ TKA/TWA/CE/CD	SMA/AK/SWA	SMSA/AKS/SWSA	
stroke bush		SR/SRB		
slide rotary bush	RK	SRE		
slide way		SV/SVW/RV	SVS/SVWS	
slide table		SVT/SYT	SYTS/SYBS	SVTS**
slide screw		SS		

* If the system is made of stainless steel and has a seal, the temperature should be below 120℃

** Contact NB if system is to be used at temperatures beyond the recommended temperature ranges.

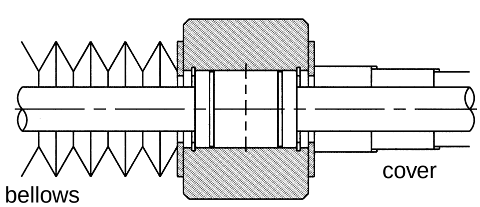
Temperature conversion equation

$$C=\frac{5}{9}(F-32) \qquad F=32+\frac{9}{5}C$$

Operating Environment:

Dust and particle debris will adversely affect the motion characteristics of a linear system. They can also cause abnormally fast wear. In a normal operating environment, the use of seals may prevent adverse effects to the linear system. However, the linear system should be well protected by using bellows or a protective cover when it is used in an extremely hostile environment. (Figure 1-10)

Figure 1-10 Example of Dust Prevention Devices



LUBRICATION

The objective of lubrication includes the reduction of friction between the rolling elements and the rolling surfaces, prevention of pitting, reduction of wear, and the prevention of rusting by forming a film over the surfaces. To maximize the performance of a linear system, a lubrication method and a lubricant appropriate for the operating environment should be selected.

Methods of lubrication include oil lubrication and grease lubrication. For oil lubrication, turbine oil conforming to ISO standard VG32-68 is also recommended. For grease lubrication, lithium soap based grease is recommended.

A rust preventative oil that does not adversely affect the lubricant is applied in a slide bush. The slide bush should be lubricated before it is placed into service. Products with raceway grooves, such as slide guides, are delivered pre-lublicated with grease and may be installed and used without further lubrication. Through operation lublicant is dispersed, so it is advisable to re-lubricate periodically based upon apprication usage.

NOTES ON HANDLING

NB linear systems are precision, machined components and should be handled with care.

To preserve the accuracy and performance expected. Any shock load caused by rough handling (such as dropping or hitting with hammer) may cause a scar on the raceway which will hinder smooth movement and shorten expected travel life. To obtain best results, it is critical to ensure accurate installation. During installation misalignment should be avoided by taking into consideration parallelism of the system.

For a linear system to deliver its best performance, ensure that the housing, shaft, and guide rail mounting surfaces all have proper and accurate dimensions.

K GREASE

K grease is a low dust generation lubricant for linear systems that provides long-term and stable dust control.

This particular type has very few impurities, yet its lubricating effect, contamination control, and rust prevention characteristics are equivalent to that of lithium soap grease.

Main Property

- Appearance : Yellow white
- Thickening agent : Ureic
- Base oil : Composition oil
- Consistency : 280(No.2)
- Operating temperature range : −30～+150℃