Ball Screw Technical Description

Feature of Ball Screws

● High mechanical efficiency
  KSS Ball Screws are fitted with steel Balls, providing rolling contact between the Nut and Screw Shaft, allowing for mechanical efficiency of about 90% and reducing the required Torque to less than one-third that of conventional Lead Screws. The design of the KSS Ball Screws also allows linear motion to be converted into rotary motion easily (Fig. A-81).

● Axial play
  With conventional Triangular and Trapezoidal Screw threads, reducing the Axial play increases the rotational Torque due to the sliding friction. KSS Ball Screws, on the other hand, are very easily rotated, even with no Axial play. The use of Double Nuts also provides increased Rigidity.

● High precision
  KSS Ball Screws are machined, assembled, and inspected using the technology of ultra-precision Lead Screw and Screw Gauge machining, under the temperature controlled room. High precision and accurate positioning ensure high reliability in use.

● Long service life
  The Ball Screw movement results in virtually no wear, as the rolling-contact design, combined with the use of carefully selected heat-treated materials, results in an extremely low friction. This is the reason that high precision can be kept over long period.

Construction of Ball Screws

● Return-plate system
  The Return-plate system uses coil-type deflectors incorporated inside the Nut to pick up the steel Balls and circulate them via the Return-plate channel. This system has the advantage of allowing the use of a Nut that is smaller in diameter than those employed in Return-tube systems. In addition, the upward-angle installation of the Return-plate ensures even smoother rotation.

● Internal-deflector system
  The Internal-deflector system employs a lightweight Miniature Ball Screw, which enables the Nut diameter and length to be reduced to the smallest possible size. The Balls bear the load while rolling along the screw groove between the Shaft and the Nut. The Balls are continuously circulated, transferred to the adjacent groove in the screw via the Internal-deflector channel and then back to the loaded groove area.

● End-cap system
  The End-cap system is a recirculating system in which the Balls advance by rolling through the screw groove between the Nut and the Screw Shaft. The Balls are then returned via the holes in the Nut and the channels in the recirculating sections of the End-caps on either end of the Nut.

● End-deflector system
  The Balls are circulated from End-deflector incorporated inside the Nut or outside the Nut through the hole in the Nut and the channels in the recirculating sections. Ball Nut diameter can be smaller than Return-plate system. This is suitable for the middle lead Ball Screws.

● Return-tube system
  In the Return-tube system, Balls rolling between the Nut and the Shaft are picked up from the screw groove by the end of the Return-tube built into the Nut. Then, they flow back through the Return-tube to the screw groove.

● Thread Groove profile
  Ball screws may have either a circular arc profile, formed of a single arc, or a gothic arc profile, formed from two arcs. KSS Ball Screws feature a gothic arc profile.

Forward Efficiency

Backward Efficiency

Fig. A-81 : Mechanical Efficiency
The range of manufacturing for Ball Screws

The range of manufacturing for KSS Ball Screws is from φ1.8 to φ16mm as Shaft nominal diameter. Maximum limit of overall lengths are shown below. Maximum limit of overall lengths will vary depending on the Shaft end configuration, materials and KSS series. Please inquire KSS for details.

<table>
<thead>
<tr>
<th>Maximum limit of overall lengths for Precision Ball Screws</th>
<th>Unit : mm</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Shaft nominal diameter</strong></td>
<td><strong>Accuracy grade</strong></td>
</tr>
<tr>
<td>4</td>
<td>C0</td>
</tr>
<tr>
<td></td>
<td>C1</td>
</tr>
<tr>
<td></td>
<td>C2</td>
</tr>
<tr>
<td></td>
<td>C3</td>
</tr>
<tr>
<td></td>
<td>C4</td>
</tr>
<tr>
<td></td>
<td>C5</td>
</tr>
<tr>
<td></td>
<td>C6</td>
</tr>
</tbody>
</table>

Note 1) If required length exceeds the number in the table above, please ask KSS representative.

<table>
<thead>
<tr>
<th>Maximum limit of overall lengths for Rolled Ball Screws (Ct7 &amp; Ct10)</th>
<th>Unit : mm</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Shaft nominal diameter</strong></td>
<td><strong>Maximum length</strong></td>
</tr>
<tr>
<td>4</td>
<td>240</td>
</tr>
<tr>
<td>5</td>
<td>300</td>
</tr>
<tr>
<td>6</td>
<td>350</td>
</tr>
<tr>
<td>8</td>
<td>450</td>
</tr>
<tr>
<td>10</td>
<td>450</td>
</tr>
<tr>
<td>12</td>
<td>700</td>
</tr>
<tr>
<td>13</td>
<td>700</td>
</tr>
<tr>
<td>14</td>
<td>700</td>
</tr>
<tr>
<td>15</td>
<td>1000</td>
</tr>
</tbody>
</table>

Note 1) If required length exceeds the number in the table above, please ask KSS representative.

Note 2) Maximum limit of overall length for Rolled Ball Screws includes 25mm of incomplete thread area at both end.

Lead accuracy of Ball Screws

Ball Screw lead accuracy conforming to JIS B 1192-3 is specified by the tolerance on specified travel over the Nut effective travel amount, or Screw Shaft useful travel, travel variation and travel variation within arbitrary 300mm, and 1 revolution (2π rad) over Screw Shaft useful travel. Tolerance of each accuracy grades are shown in the Table A-83, 84, 85.

Fig. A-82 : Travel deviation diagram

- **Nominal travel (l0)**: Travel in axial direction when rotated arbitrary number of revolution according to the Nominal lead
- **Specified Lead (Phs)**: Lead given some amount of correction to the Nominal lead in order to compensate the deformation generated due to the temperature rise or the load.
- **Travel compensation (c)**: Difference between the Specified travel and the Nominal travel within the valid travel.
- **Specified travel (ls)**: Travel in axial direction when rotated arbitrary number of revolution according to the Specified lead.
- **Actual travel (la)**: Actual travel of Ball Nut in axial direction in respect to an arbitrary angle of rotation of Ball Screw Shaft.
- **Actual mean travel (lm)**: Straight line which represents the tendency of Actual travel. It is obtained by the least square method or a simple and appropriate approximation method from the curve indicating the Valid travel of Ball Nut.
- **Tolerance on specified travel (ep)**: Difference between the Actual mean travel and the Specified travel corresponding to the Valid travel of Ball Nut or the Useful travel of Ball Screw Shaft.
- **Travel variation (V300)**: Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to Valid travel of Ball Nut or Useful travel of Ball Screw Shaft.
- **Travel variation (Vu)**: Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to arbitrary one revolution (2π rad) within Useful travel of Ball Screw Shaft.
- **Travel variation (V2K)**: Maximum width of the Actual travel curve between the two straight lines put in parallel to the Actual mean travel line, that corresponding to arbitrary 300mm taken within Useful travel of Ball Screw Shaft.
Table A-83: Tolerance on specified travel (±ευ) and permissible travel variation (Vu) of precision Ball Screws (for positioning: C series)

<table>
<thead>
<tr>
<th>Accuracy Grade</th>
<th>C0</th>
<th>C1</th>
<th>C3</th>
<th>C5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Useful travel (mm)</td>
<td>Over</td>
<td>Up to</td>
<td>±ευ</td>
<td>Vu</td>
</tr>
<tr>
<td>—</td>
<td>100</td>
<td>±3</td>
<td>±3.5</td>
<td>5</td>
</tr>
<tr>
<td>100</td>
<td>200</td>
<td>±3.5</td>
<td>±4.5</td>
<td>5</td>
</tr>
<tr>
<td>200</td>
<td>315</td>
<td>±4</td>
<td>±3.5</td>
<td>6</td>
</tr>
<tr>
<td>315</td>
<td>400</td>
<td>±5</td>
<td>±3.5</td>
<td>7</td>
</tr>
<tr>
<td>400</td>
<td>500</td>
<td>±6</td>
<td>±4</td>
<td>8</td>
</tr>
<tr>
<td>500</td>
<td>630</td>
<td>±6</td>
<td>±4</td>
<td>9</td>
</tr>
<tr>
<td>630</td>
<td>800</td>
<td>±7</td>
<td>±5</td>
<td>10</td>
</tr>
<tr>
<td>800</td>
<td>1000</td>
<td>±8</td>
<td>±6</td>
<td>11</td>
</tr>
</tbody>
</table>

Table A-84: Permissible travel variation V₃₀₀, V₂ for positioning (C series)

<table>
<thead>
<tr>
<th>Accuracy Grade</th>
<th>C0</th>
<th>C1</th>
<th>C3</th>
<th>C5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item</td>
<td>V₃₀₀</td>
<td>V₀,π</td>
<td>V₃₀₀</td>
<td>V₀,π</td>
</tr>
<tr>
<td>Permissible value</td>
<td>3.5</td>
<td>3</td>
<td>5</td>
<td>4</td>
</tr>
</tbody>
</table>

Table A-85: Permissible travel variation V₃₀₀ for Ct series (7,10 grade)

<table>
<thead>
<tr>
<th>Accuracy Grade</th>
<th>Ct7</th>
<th>Ct10</th>
</tr>
</thead>
<tbody>
<tr>
<td>V₃₀₀</td>
<td>52</td>
<td>210</td>
</tr>
</tbody>
</table>

Japan Industrial Standard of Ball Screw (JIS B1192) was revised in 1997, 2013 and 2018 in order to correspond to ISO. Regarding accuracy grade, C series (current JIS C0, 1, 3, 5) and Cp, Ct series (standard corresponding to ISO) are established. There are some differences between C series and Cp, Ct series in notation and tolerances for accuracy of Ball Screw mounting section. KSS uses notation in Fig. A-86 below and standard tolerance value, which conforms to C series standard, and KSS refers to Cp, Ct series standard in case of 7 and 10 grade. Moreover, in the revision of 2018, the notation of perpendicularity changed to ‘run-out of the mounting surface or end face’, and geometric tolerance symbols changed from ⊥ to ⊥. Fig. A-86: Description of Run-out and location tolerances for Ball Screws.
Table A-88 : Radial Run-out of Bearing seat related to the centerline of screw groove and Radial Run-out of journal diameter related to the Bearing seat

<table>
<thead>
<tr>
<th>Shaft nominal diameter (mm)</th>
<th>Permissible deviation of Radial Run-out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>Up to</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>12</td>
<td>12</td>
</tr>
</tbody>
</table>

This measurement item is affected by Total Run-out of the Screw Shaft, and so it must be corrected as follows. Find the corrected value from the Total Run-out tolerances given in Tables A-93~98 on page A811 using the ratio of the total Shaft length to the distance between the supporting point and the measuring point\(L_1, L_2\) (see Fig. A-87), and add the values obtained to the tolerance given in Table A-88.

![Compensation of Radial Run-out](image)

Compensation Value of Run-out = \frac{\text{Tolerance of total Run-out (Table A-93~98)}}{\text{Total shaft length}} \times (L_1, L_2)

\(L_1, L_2\) : Distance btw supporting pt & measuring pt (mm)

Table A-89 : Axial Run-out (Perpendicularity) of Shaft(Bearing) face related to the centerline of the Bearing seat

<table>
<thead>
<tr>
<th>Shaft nominal diameter (mm)</th>
<th>Permissible deviations of Axial Run-out (Perpendicularity)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>Up to</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>12</td>
<td>12</td>
</tr>
</tbody>
</table>

Table A-90 : Axial Run-out (Perpendicularity) of Ball Nut location face related to the centerline of Screw Shaft

<table>
<thead>
<tr>
<th>Nut outside diameter (mm)</th>
<th>Permissible deviations of Axial Run-out (Perpendicularity)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>Up to</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>32</td>
<td>32</td>
</tr>
</tbody>
</table>

Table A-91 : Radial Run-out of Ball Nut location diameter related to the centerline of Screw Shaft

<table>
<thead>
<tr>
<th>Nut outside diameter (mm)</th>
<th>Permissible deviations of Radial Run-out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>Up to</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>32</td>
<td>32</td>
</tr>
</tbody>
</table>

Table A-92 : Parallelism of rectangular Ball Nut related to the centerline of Screw Shaft

<table>
<thead>
<tr>
<th>Mounting length (mm)</th>
<th>Permissible deviations of Parallelism</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>Up to</td>
</tr>
<tr>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>100</td>
<td>100</td>
</tr>
</tbody>
</table>
### Table A-93: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C0)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Shaft nominal diameter</th>
<th>Unit ( \text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>8</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>12</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Over</td>
<td>125</td>
<td></td>
<td>0.015</td>
<td>0.015</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td></td>
<td>0.020</td>
<td>0.020</td>
</tr>
<tr>
<td></td>
<td>315</td>
<td>400</td>
<td>0.035</td>
<td>0.025</td>
</tr>
<tr>
<td></td>
<td>500</td>
<td>630</td>
<td>0.045</td>
<td>0.035</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>1000</td>
<td></td>
<td>0.050</td>
</tr>
</tbody>
</table>

### Table A-94: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C1)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Shaft nominal diameter</th>
<th>Unit ( \text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>8</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>12</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Over</td>
<td>125</td>
<td></td>
<td>0.020</td>
<td>0.020</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td></td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td></td>
<td>315</td>
<td>400</td>
<td>0.040</td>
<td>0.030</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>630</td>
<td></td>
<td>0.055</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>1000</td>
<td></td>
<td>0.075</td>
</tr>
</tbody>
</table>

### Table A-95: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C3)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Shaft nominal diameter</th>
<th>Unit ( \text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>8</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>12</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Over</td>
<td>125</td>
<td></td>
<td>0.025</td>
<td>0.035</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td></td>
<td>0.035</td>
<td>0.030</td>
</tr>
<tr>
<td></td>
<td>315</td>
<td>400</td>
<td></td>
<td>0.040</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>630</td>
<td></td>
<td>0.045</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>1000</td>
<td></td>
<td>0.070</td>
</tr>
</tbody>
</table>

### Table A-96: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C5)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Shaft nominal diameter</th>
<th>Unit ( \text{mm} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>8</td>
<td>12</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>12</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Over</td>
<td>125</td>
<td></td>
<td>0.035</td>
<td>0.040</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td></td>
<td>0.045</td>
<td>0.035</td>
</tr>
<tr>
<td></td>
<td>315</td>
<td>400</td>
<td></td>
<td>0.050</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>630</td>
<td></td>
<td>0.060</td>
</tr>
<tr>
<td></td>
<td>800</td>
<td>1000</td>
<td></td>
<td>0.120</td>
</tr>
</tbody>
</table>
Table A-97: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C7)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Unit: mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>8</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>125</td>
<td>0.640</td>
<td>0.095</td>
<td>0.055</td>
</tr>
<tr>
<td>200</td>
<td>0.075</td>
<td>0.065</td>
<td>0.060</td>
</tr>
<tr>
<td>200</td>
<td>0.100</td>
<td>0.080</td>
<td>0.070</td>
</tr>
<tr>
<td>315</td>
<td>0.100</td>
<td>0.080</td>
<td>0.070</td>
</tr>
<tr>
<td>400</td>
<td>0.120</td>
<td>0.095</td>
<td></td>
</tr>
<tr>
<td>500</td>
<td>0.150</td>
<td>0.110</td>
<td></td>
</tr>
<tr>
<td>630</td>
<td>0.140</td>
<td></td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>0.170</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Permissible deviations of total Run-out in radial direction

Table A-98: Total Run-out in radial direction of Screw Shaft related to the centerline of Screw Shaft (C10)

<table>
<thead>
<tr>
<th>Shaft total length</th>
<th>Over</th>
<th>Up to</th>
<th>Unit: mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>8</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>125</td>
<td>0.100</td>
<td>0.095</td>
<td>0.090</td>
</tr>
<tr>
<td>200</td>
<td>0.140</td>
<td>0.120</td>
<td>0.110</td>
</tr>
<tr>
<td>200</td>
<td>0.210</td>
<td>0.160</td>
<td>0.130</td>
</tr>
<tr>
<td>315</td>
<td>0.210</td>
<td>0.210</td>
<td>0.160</td>
</tr>
<tr>
<td>400</td>
<td>0.270</td>
<td>0.200</td>
<td></td>
</tr>
<tr>
<td>500</td>
<td>0.350</td>
<td>0.250</td>
<td></td>
</tr>
<tr>
<td>630</td>
<td>0.440</td>
<td>0.320</td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>0.420</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Permissible deviations of total Run-out in radial direction

Measuring method of Ball Screw Run-out and location tolerances

● Radial Run-out of Bearing seat related to the centerline of screw groove (Table A-88)

Place the Ball Screw in identical V-blocks at both Bearing seat. Place the dial gauge perpendicular to the Nut cylindrical surface. Rotate Screw Shaft slowly and record the dial gauge readings. Measurement should be done at near both ends of threaded part. Some cases, this measurement will be done by both centerhole support, and directly measured on Bearing seat.

● Radial Run-out of journal diameter related to the Bearing seat (Table A-88)

Place the Ball Screw in identical V-blocks at both Bearing seats. Place the dial gauge perpendicular to the journal cylindrical surface. Rotate the Screw Shaft slowly and record the dial gauge readings.

● Axial Run-out (Perpendicularity) of shaft (Bearing) face related to the centerline of the Bearing seat (Table A-89)

Support a Screw Shaft at both centers. Place the dial gauge perpendicular to the end face of the journal. Rotate the Screw Shaft slowly and record the dial gauge readings. **This method is equivalent to the one, which is supported at both Bearing seats, because Bearing seats are ground related to both centers.

Note: In case of C7, C10 grade, KSS may use the standard of Total Run-out based on slenderness ratio, which conforms to JIS B1192-2013.

![Diagram](image1)

Slenderness ratio Total Run-out

<table>
<thead>
<tr>
<th>Slenderness ratio</th>
<th>Over</th>
<th>C7</th>
<th>C10</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>0.080</td>
<td>0.160</td>
<td></td>
</tr>
<tr>
<td>60</td>
<td>0.120</td>
<td>0.240</td>
<td></td>
</tr>
<tr>
<td>80</td>
<td>0.200</td>
<td>0.400</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td>0.320</td>
<td>0.640</td>
<td></td>
</tr>
</tbody>
</table>

Slenderness ratio = (L / D) where L = Useful travel (mm) and D = Nominal diameter of Ball Screw (mm)
Material and Heat treatment, Surface hardness

Standard material of KSS Ball Screws, Heat treatment and Surface hardness are shown in Table A-99, 100. However, they vary depending on series or model number. Please refer to KSS drawings.

<table>
<thead>
<tr>
<th>Material</th>
<th>Heat treatment</th>
<th>Surface hardness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Screw Shaft</td>
<td>SCM415 (JIS G 4105)</td>
<td>Carburizing and quenching</td>
</tr>
<tr>
<td></td>
<td>S55C (JIS G 4051)</td>
<td>Induction hardening</td>
</tr>
<tr>
<td>Nut</td>
<td>SCM415 (JIS G 4105)</td>
<td>Carburizing and quenching</td>
</tr>
</tbody>
</table>

**Table A-100 : Material, Heat treatment & Surface hardness for stainless steel items**

<table>
<thead>
<tr>
<th>Material</th>
<th>Heat treatment</th>
<th>Surface hardness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Screw Shaft</td>
<td>SUS440C (JIS G 4303)</td>
<td>Quenching and tempering</td>
</tr>
<tr>
<td>Nut</td>
<td>SUS440C (JIS G 4303)</td>
<td>Quenching and tempering</td>
</tr>
</tbody>
</table>

Note 1: Hardness on table shows surface hardness of thread part.
Note 2: S55C is applicable for Precision Rolled Ball Screws.
Permissible Axial load

It is recommended that Ball Screw Shafts be used almost exclusively under tension load conditions. However, in some applications, compression loads may exist, and under such conditions it must be checked that Shaft buckling will not occur. Also, when the mounting span distance is short, there is a restriction on the permissible tension or compression load and the Basic Static Load Rating Coa unrelated to mounting. Buckling load, permissible tension and permissible compression load can be calculated below.

Permissible compression load calculation for buckling

\[ P = \alpha \times \frac{n \cdot E \cdot I}{L^2} \quad \text{N} \quad \text{Formula for Oiler} \]

\( \alpha \): Safety Factor 0.5
\( E \): Young's modulus 2.08 \times 10^{11} \text{N/mm}^2 (\text{MPa})
\( I \): Screw Shaft minimum moment of inertia of area
\( I = \frac{\pi}{64} d^4 \quad \text{mm}^4 \)
\( d \): Screw Shaft Root diameter mm
\( L \): Mounting span distance mm
\( n \): Factor for Ball Screw mounting method
  - Supported—Supported \( n = 1 \)
  - Fixed—Supported \( n = 2 \)
  - Fixed—Fixed \( n = 4 \)
  - Fixed—Free \( n = 1/4 \)

Permissible tension, compression load calculation for Screw Shaft yield stress

\[ P = \sigma \times A \quad \text{N} \]

\( \sigma \): Permissible stress 98N/mm\(^2\) (MPa)
\( A \): Screw Shaft minimum section area
\[ A = \frac{\pi}{4} d^2 \quad \text{mm}^2 \]
\( d \): Screw Shaft Root diameter mm

Permissible speed

For Screw Shaft rotation, the mounting method determines the established rotation limits. When this value is approached, resonance phenomenon will occur, and operation becomes impossible. There is also rotation limit which causes damages to recirculating parts. This limit is unrelated to mounting methods.

Permissible speed calculation for critical speed

\[ N = \beta \times \frac{40 \cdot \lambda^2}{2 \pi} \times \sqrt{\frac{E \cdot I \cdot g}{y} \cdot A \cdot L^4} \quad \text{min}^{-1} \]

\( \beta \): Safety Factor 0.8
\( E \): Young's modulus 2.08 \times 10^{11} \text{N/mm}^2 (\text{MPa})
\( I \): Screw Shaft minimum moment of inertia of area
\( I = \frac{\pi}{64} d^4 \quad \text{mm}^4 \)
\( d \): Screw Shaft Root diameter mm
\( g \): Gravity acceleration 9.8 \times 10^3 \text{mm/sec}^2
\( y \): Material specific gravity 7,850kg/m\(^3\) (7.7 \times 10^3 \text{N/mm}^3)
\( L \): Mounting span distance mm
\( A \): Screw Shaft minimum section area
\[ A = \frac{\pi}{4} d^2 \quad \text{mm}^2 \]
\( \lambda \): Factor for Ball Screw mounting method
  - Supported—Supported \( \lambda = \pi \)
  - Fixed—Supported \( \lambda = 3.927 \)
  - Fixed—Fixed \( \lambda = 4.730 \)
  - Fixed—Free \( \lambda = 1.875 \)

Rotational speed limit for damage on recirculating parts

Generally, regarding critical speed for damage on recirculating parts, limitation is established by \( dn \) value, which is multiplied Shaft nominal diameter of revolution, but \( dn \) value cannot be applied to Miniature Ball Screws. For KSS Ball Screws, please consider rotational speed limit by damage on recirculating parts as 3,500 to 4,000 min\(^{-1}\). This value varies depending on operating conditions and environment. Please inquire KSS for details.

Moreover, possibilities of breakage of recirculating parts will be increased when using in high acceleration / deceleration. Estimate criterion of the breakage in the recirculating section is depending on the internal specification of the Ball Screw, please ask KSS for more detail.
Ball Screw mounting methods

Typical Ball Screw’s mounting methods are shown in Fig. A-101. Mounting configuration affects permissible Axial load in relation to buckling, as well as permissible speed in relation to critical speed. Please refer to below when studying strength and speed.

Axial play and Preload

For standard Single Nut Ball Screws under normal conditions, a slight Axial play exists between the Screw Shaft and Nut. Consequently, when Axial loads act on Single Nut Ball Screws, total amount of Axial play and Elastic displacement due to Axial load becomes backlash. In order to prevent this backlash in Ball Screws, the Axial play can be reduced to a negative value. That is what we call “Preload”, which is the method of causing Elastic deformation to the Balls between the Screw Shaft and Nut in advance.

Axial play
Symbol and permissible value for Axial play are shown in Table A-102. Combination of accuracy grade and symbol are shown in Table A-103.

Table A-102 : Symbol and permissible value for Axial play

<table>
<thead>
<tr>
<th>Symbol</th>
<th>0</th>
<th>02</th>
<th>05</th>
<th>20</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial play</td>
<td>0 (Preloading)</td>
<td>0.002 max.</td>
<td>0.005 max.</td>
<td>0.02 max.</td>
<td>0.05 max.</td>
</tr>
</tbody>
</table>

Table A-103 : Combination of accuracy grade and Axial play

<table>
<thead>
<tr>
<th>Accuracy grade</th>
<th>0</th>
<th>02</th>
<th>05</th>
<th>20</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>C0</td>
<td>C0-0</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C1</td>
<td>C1-0</td>
<td>C1-02</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>C3</td>
<td>C3-0</td>
<td>C3-02</td>
<td>C3-05</td>
<td>C3-20</td>
<td>C3-50</td>
</tr>
<tr>
<td>C5</td>
<td></td>
<td></td>
<td>C5-05</td>
<td>C5-20</td>
<td>C5-50</td>
</tr>
<tr>
<td>C7</td>
<td></td>
<td></td>
<td></td>
<td>C7-20</td>
<td>C7-50</td>
</tr>
<tr>
<td>C10</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>C10-20</td>
</tr>
</tbody>
</table>

Note: When combinations other than the above are requested, please inquire KSS.
**Preload effect**  
Preload is not only used for removing Axial play, it also has the effect of reducing the amount of Axial displacement due to Axial load, and improving the Rigidity in Ball Screws. Fig. A-104 shows the difference of the amount of Elastic displacement (theoretical value) regarding Ball Screw with Axial play and Ball Screw with Preload under the Axial load.

**Fig. A-104 : Elastic displacement curve comparison between Backlash type and Preload type**

![Elastic displacement Curve](image)

- **Example**  
  - Shaft : φ14  
  - Load : 3mm

**Preload methods**  
Generally, a method of Double Nut Preload by inserting a spacer between two Nuts is adopted. KSS Ball Screw adopts (Oversized Ball Preload) by inserting Balls slightly bigger than space between Screw Shaft and Nut. As a result, it can eliminate Axial play even with a Single Nut and it is possible to maintain compact. Moreover, operating performance will never be deteriorated by using spacer Balls (Balls with slightly smaller diameter than those of the oversize Balls) alternatively with oversize Balls.

**Fig. A-105 : Preload by oversized Balls**

**Fig. A-106 : Spacer Balls**

**Preload control**  
It is difficult to control Preload amount by measuring. Therefore, Preload of Ball Screw is controlled by measuring Preload Dynamic Drag Torque, which is converted from Preload amount. Amount of Preload Dynamic Drag Torque is decided with customers by specification drawing. Preload Dynamic Drag Torque is measured under specific condition to verify the amount of Axial play is 0. Dynamic Drag Torque installed actual machine will vary depending on lubricating condition, load condition and so on. Starting torque (Torque for starting Ball Screw) is slightly bigger than Dynamic Drag Torque.

**Fig. A-107 : Dynamic Drag Torque measurement**

*Torque wave in this diagram is exaggerated for explanation.*

**Proper amount of Preload**  
Although the amount of Preload should be determined by the required Rigidity and the permissible amount of backlash, when setting Preload, there are some concerning issues as follows.  
1) Increased Dynamic Drag Torque  
2) Heat generation, lowering of positioning accuracy due to the temperature rise.  
3) Shortened life

Therefore, it is advisable to establish the amount of Preload at the lowest possible limits.
Rigidity in Linear Motion system

In precision machinery, to improve positioning accuracy of the drive screws or to increase Rigidity for load, the Rigidity of the entire Linear Motion system must be examined. Rigidity of entire Linear Motion system is as follows.

\[ \frac{1}{K} = \frac{1}{K_1} + \frac{1}{K_2} + \frac{1}{K_3} + \frac{1}{K_4} \quad \mu m/N \]

- \( K \): Total Rigidity of Linear Motion system \( N/\mu m \)
- \( K_1 \): Screw Shaft Rigidity \( N/\mu m \)
- \( K_2 \): Nut Rigidity \( N/\mu m \)
- \( K_3 \): Support Bearing Rigidity \( N/\mu m \)
- \( K_4 \): Nut, Bearing fitting part Rigidity \( N/\mu m \)

Total Rigidity of Linear Motion system \( K \)

\[ K = \frac{F_a}{\delta} \quad N/\mu m \]

- \( F_a \): Axial load applied to Linear Motion system \( N \)
- \( \delta \): Elastic displacement of Linear Motion system \( \mu m \)

Screw Shaft Rigidity \( K_1 \)

(1) In case of general mounting (Fixed-Free in axial direction) (Fig. A-108)

\[ K_1 = \frac{A \cdot E}{\delta} \times 10^{-3} \quad N/\mu m \]

(2) In case of Fixed-Fixed mounting in axial direction (Fig. A-109)

\[ K_1 = \frac{4 \cdot A \cdot E \cdot L}{\delta^2 (L - \delta)} \times 10^{-3} \quad N/\mu m \]

The max. axial displacement occurs when \( \delta = L/2 \). The formula is as follows.

\[ K_1 = \frac{4 \cdot A \cdot E}{L} \times 10^{-3} \quad N/\mu m \]

- \( A \): Screw Shaft minimum section area
- \( d \): Screw Shaft Root diameter \( mm \)
- \( E \): Young’s modulus \( 2.08 \times 10^5 \) N/mm\(^2\) (MPa)
- \( \delta \): Axial distance between fixed point & Nut center \( mm \)
- \( L \): Mounting span distance \( mm \)

Accordingly, the amount of Screw Shaft Elastic displacement \( \delta \) due to Axial load \( F_a \) is as follows.

\[ \delta = \frac{F_a}{K_1} \quad \mu m \]

Fig. A-108: Fixed-Free in axial direction
Fig. A-109: Fixed-Fixed in axial direction
**Nut Rigidity K**
Calculation formula of static Rigidity is defined by JIS B1192-4 established in 2018. KSS will use the formula which is defined by JIS to identify the static Rigidity.

1. **Rigidity of Single Nut with backlash**
   Theoretical static Rigidity (K2) of the Single Nut with backlash is calculated by the formula as follows.

   $K_2 = f_{\nu} \times (3/2) \times \frac{F_a}{\delta} \quad (N/\mu m)$

   $K_2$ : Theoretical Nut Rigidity  
   $F_a$ : Axial Load  
   $\delta$ : Amount of Elastic displacement at Axial Load $F_a$  
   $f_{\nu}$ : Correction factor = 0.67

   $\delta = k \times F_{pr}^{10} \quad (\mu m)$

   $k = \frac{2^{10} \times D_w^{10} \times (\sin \beta \times \cos \beta)^{10}}{C}$

   $k$ : Rigidity Characterization factor  
   $Z$ : Quantity of loaded Ball  
   $D_w$ : Diameter of Ball  
   $\alpha$ : Contact angle to the thread groove  
   $\beta$ : Lead angle  
   $C$ : Coefficient depending on the material, shape and dimension 0.52~0.58

The theoretical static Rigidity $K_2$ of the Nut under an Axial load equivalent to 30% of the Basic Dynamic Load Rating $Ca$ is described in dimension table. For Axial loads which are not 30% of the Basic Dynamic Load Rating $Ca$, it can be easily calculated by following formula.

$K'_{2} = K_{2} \times (\frac{F_{pr}}{0.3Ca})^{10} \quad N/\mu m$

$K_2$ : Nut Rigidity in dimension table  
$F_{pr}$ : Preload amount  
$Ca$ : Basic Dynamic Load Rating

2. **Rigidity of preloaded Ball Nut**
   Theoretical static Rigidity ($K_2$) of the preloaded single Ball Nut will become a fixed value if axial load ($F_a$) is less than $2\sqrt{2}$ times of the preload amount ($F_{pr}$) regardless of the value of the axial load ($F_a$), and this will be calculated as follows.

   $K_2 = 2^{10} \times \frac{1}{k} \times F_{pr}^{10} \quad N/\mu m$

   $k$ : Rigidity Characterization factor  
   See formula stated above  
   $F_{pr}$ : Preload amount

In case of Preload type Ball Screws, Rigidity varies depending on the dispersion of Preload Dynamic Drag Torque. Therefore, please inquire KSS for details.

If the axial load ($F_a$) will be more than $2\sqrt{2}$ times of the preload amount ($F_{pr}$), the calculation formula will be the same as the formula for single Nut theoretical static Rigidity.

The theoretical static Rigidity $K_2$ under a Preload equivalent to 5% of the Basic Dynamic Load Rating $Ca$ is described in dimension table. For Preload amounts other than the above, it can be easily calculated by following formula.

$K'_{2} = K_{2} \times (\frac{F_{pr}}{0.05Ca})^{10} \quad N/\mu m$

$K_2$ : Nut Rigidity in dimension table  
$F_{pr}$ : Preload amount  
$Ca$ : Basic Dynamic Load Rating

- **Support Bearing Rigidity $K_S$**
  Support Bearing Rigidity varies depending on the type of Bearing and amount of Preload. Please inquire Bearing manufacturers.

- **Nut, Bearing fitting part Rigidity $K_2$**
  Rigidity of Nut mounting part and Bearing mounting part vary depending on machine structure and design. KSS cannot mention the details but a design of high Rigidity must be considered.

- **Screw Shaft torsion Rigidity**
  For positioning error due to torsion, this error is a relatively small compared to axial displacement. However, if investigation is required, the following formula may be used for calculation.

  $\theta = \frac{32TL}{n6d} \times \frac{180}{\pi} \times 10 \quad \deg$

  $\theta$ : Torsion angle due to torsion moment  
  $T$ : Torsion moment  
  $L$ : Distance between Nut & Shaft end support  
  $G$ : Modulus of Rigidity $8.3 \times 10^1 \quad N/mm^2$ (MPa)  
  $d$ : Screw Shaft Root diameter

  Amount of axial displacement $\delta a$ due to torsion angle is as follows.

  $\delta a = \frac{\theta}{360} \times 10^3 \quad \mu m$

  $\delta$ : Lead mm
Basic Load Rating and Basic Rating Life

**Basic Dynamic Load Rating Ca and Basic Rating Life**
The Basic Rating Life of Ball Screws means the total number of revolutions which 90% of the Ball Screws can endure. Failure is indicated by flaking caused by rolling fatigue on the surface of grooves or Balls. These figures are valid when a group of the same type Ball Screws are operated individually under the same conditions. The Basic Dynamic Load Rating Ca is the Axial load for which the Basic Rating Life is 1,000,000 revolutions. These values are listed under Ca in the dimension tables. Ball Screw’s Basic Rating Life L₁₀ can be estimated using Basic Dynamic Load Rating Ca in the following basic formula.

\[
L₁₀ = \frac{Ca}{1.3 \times 10^4} \text{ rev.}
\]

Also, in place of the total number of revolutions, the Basic Rating Life can be expressed in hours \( L₁₀h \) or traveled distance \( L₁₀d \), and these can be calculated through the following formulas.

\[
L₁₀h = \frac{1}{60 \times N \times Ca} \times L₁₀ \text{ hours}
\]

\[
L₁₀d = \frac{d}{10^6} \times L₁₀ \text{ km}
\]

Generally, Axial load on the most machine is not constant and it can be divided into several operating pattern. In this case, Basic Rating Life can be calculated to figure up equivalent Axial load \( F_{\text{eq}} \), equivalent Revolution \( N_{\text{eq}} \), in the following formula.

\[
F_{\text{eq}} = \left( \frac{F_{a1} \cdot N_{1} \cdot t_{1} + F_{a2} \cdot N_{2} \cdot t_{2} + F_{a3} \cdot N_{3} \cdot t_{3}}{N_{1} + N_{2} + N_{3} + t_{1} + t_{2} + t_{3}} \right)^{1/2}
\]

Also, for Axial loads which vary linearly, the average Axial load \( F_{\text{mean}} \) can be calculated approximately using the following formula.

\[
F_{\text{mean}} = \frac{F_{\text{min}} + 2 \cdot F_{\text{max}}}{3} \text{ N}
\]

\[
F_{\text{min}} \text{ : Minimum Axial load}\ N
\]

\[
F_{\text{max}} \text{ : Maximum Axial load}\ N
\]

Note) As the Basic Rating Life varies due to lubricating conditions, and contaminations, Moment load or Radial load, etc., this should be considered a rough estimate only.

Load direction and Preload will be taken into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018. Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B1192-5.

**Life calculation considered the Load direction**
Contact point of the Steel Balls changes based on Load direction (see Fig. A-110), therefore it is considered the lifetime when flaking occurred at any contact points, with calculating the Rating Life at each contact point of the Steel Balls.

The calculating formula is as follows.

\[
L'₁₀ = \left( \frac{L_{\text{L10A}}^{10/9} + L_{\text{L10B}}^{10/9}}{10} \right) \times 10^{10} \text{ rev.}
\]

\[
L₁₀' \text{ : Merged Basic Rating Life of contact point A and B}
L_{\text{L10A}} \text{ : Basic Rating Life on contact point A}
L_{\text{L10B}} \text{ : Basic Rating Life on contact point B}
\]

Fig. A-110 : Ball contact condition by load direction
Life calculation considered the Preload

Preloaded Ball Screw is filled with oversized Balls, therefore each Steel Ball is contacted at four (4) points between Screw Shaft and Ball Nut. It is considered the lifetime when flaking occurred at any contact points, with calculating the Rating Life at each contact point.

The contact point of the Steel Balls is described in Fig. A-111, when Preload is effective by oversized Balls. The amount of Elastic displacement is described schematically by oval (contact ellipse). Both contact point A and B are evenly contacted under no load from outside.

Using the value calculated by the above formula, calculate the Rating Life at each contact point A and B (L(A10), L(B10)), then merge both value to calculate the merged Basic Rating Life.

\[
L_{(A10)} = \left(\frac{Ca}{F_{\text{max}}}\right)^{1/10} \text{ rev.}
\]

\[
L_{(B10)} = \left(\frac{Ca}{F_{\text{max}}}\right)^{1/10} \text{ rev.}
\]

\[
L'_{(10)} = (L_{(A10)} - v^A + L_{(B10)} - v^B)^{1/10} \text{ rev.}
\]

Note) As a rough estimation of Basic Rating Life, we consider the Axis load as external load added by preload amount Fpr, for some cases.

Basic Static Load Rating Coa

The Basic Static Load Rating Coa is the Axial Static load at which the amount of permanent deformation (Ball + Raceway) occurring at the maximum stress contact point between the Ball and Raceway surfaces is 1/10,000 times the Ball diameter. These values are listed under Coa in the dimension tables. The Basic Static Load Rating Coa values apply to investigation of stationary state or extremely low Revolution load conditions (less than 10 min-1). However, in most cases the amount of permanent deformation causes absolutely no problems under the general conditions. The maximum permissible load Fa max for the screw groove can be found by using the following formula.

\[
F_{\text{max}} = \frac{\text{Coa}}{f_s} \times N
\]

fs : Static safety factor
f_s = 1~2 for normal operation
f_s = 2~3 for vibration, impact

Hardness coefficient

For Surface hardness of less than HRC58 (654 Hv10), the Basic Dynamic Load Rating Ca and the Basic Static Load Rating Coa must be adjusted. Adjustment is made by the following formula.

\[
\text{Ca}' = f_h \cdot \text{Ca} \quad \text{(N)}
\]

\[
\text{Coa}' = f_h \cdot \text{Coa} \quad \text{(N)}
\]

\[
f_h = \left(\frac{H_v}{654}\right)^{1/2} \leq 1
\]

\[
f_i = \left(\frac{H_v}{654}\right)^{1/2} \leq 1
\]

H_v : Vickers hardness Hv10

See formula above and graph right.

Note) Load direction of A and B is opposite.
Driving Torque

Driving Torque in Linear Motion System \( T \) is expressed according to the following formula.

\[
T = T_1 + T_2 + T_3 + T_4 \quad \text{N·m}
\]

- \( T_1 \): Acceleration Torque \( \text{N·m} \)
- \( T_2 \): Load Torque \( \text{N·m} \)
- \( T_3 \): Preload Dynamic Drag Torque \( \text{N·m} \)
- \( T_4 \): Additional Torque \( \text{N·m} \)

When Motor selection, Driving Torque in Linear Motion System is needed. \( T_1 \sim T_3 \) can be calculated by the following formula.

### Acceleration Torque \( T_1 \)

\[
T_1 = \frac{2lN}{60 \cdot t} \quad \text{rad/sec}^2
\]

\[
l = lw \cdot A^2 + ls \cdot A^2 + la \cdot A^2 + la \quad \text{kg·m}^2
\]

\[
lw = m_w \times \left( \frac{\rho}{2\eta} \right)^2 \quad \text{kg·m}^2
\]

\[
ls = m_s \times \left( \frac{d}{8} \right) \quad \text{kg·m}^2
\]

\[
m_w = \pi \left( \frac{d}{2} \right)^2 \times L \times \gamma \quad \text{kg}
\]

### Load Torque \( T_2 \)

\[
T_2 = \frac{P \cdot Q \cdot A}{2 \pi n} \times 10^{-3} \quad \text{N·m}
\]

\[
P \quad \text{Axial load} \quad \text{N}
\]

\[
Q \quad \text{Load} \quad \text{N}
\]

\[
m \quad \text{Mass of moving object} \quad \text{kg}
\]

\[
g \quad \text{Gravity acceleration}=9.8 \times 10^3 \text{mm/sec}^2
\]

\[
\rho \quad \text{Lead} \quad \text{mm}
\]

\[
\mu \quad \text{Sliding surface friction coefficient}
\]

\[
\eta \quad \text{Efficiency}=0.9
\]

\[
A \quad \text{Reduction ratio}
\]

### Preload Dynamic Drag Torque \( T_3 \)

\[
T_3 = 0.05 \times (\tan \beta)^{-1} \times \frac{F_{p}}{2 \pi} \times 10^{-3} \quad \text{N·m}
\]

\[
\beta \quad \text{Lead angle} \quad \text{deg}
\]

\[
F_p \quad \text{Preload} \quad \text{N}
\]

\[
\beta \quad \text{Lead} \quad \text{mm}
\]

### Additional Torque \( T_4 \)

Described as Torque which occurs in addition to those listed above. For example, support Bearing friction Torque, oil seal resistance Torque, etc.
Rust prevention and Lubrication

- Rust prevention
  KSS Ball Screws are applied anti-rust oil when shipping in case of no specific instruction. This oil should be removed before use. Wash Ball Screws with cleaned Kerosine and apply lubricant (Grease or Oil) on Ball Screws. As customer’s request, specified Grease or Oil can be applied, but it should be noted that they are not suitable for long term storage purpose and rust might occur.
  Note) Anti-rust oil is focused on anti-rust performance and it does not have lubricating function. Therefore, when using Ball Screws with anti-rust oil coating, the problems such as shortened life, increase of Torque and abnormal heat generation occurs.

- Lubrication
  In Ball Screw use, lubricant should be required. If lubricant is not applied with, the problem such as increase of Torque and shortened life occurs. Applying lubricant can minimize temperature increases, decline of mechanical efficiency due to friction, and deterioration of accuracy caused by wear. Ball Screw lubrication is divided into Greasing and Oiling. A regular lithium-soap-based Grease and ISO VG32-68 Oil (turbine Oil #1 to #3) are recommended. It is highly important to choose lubricant depending on customer’s usage. Especially in case of Miniature Ball Screws, malfunction such as increase of Torque are caused by the stir resistance. KSS original Greases which maintains Ball Screw’s smooth movement and have high lubricating performance are prepared. MSG No.1 is appropriate for high smooth requirement and high positioning usage (consistency 1). MSG No.2 is suitable for high speed and general usage (consistency 2).

Recommended lubricants for normal operating conditions

<table>
<thead>
<tr>
<th>Lubricant</th>
<th>Type</th>
<th>Product name</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grease</td>
<td>Lithium-based Grease</td>
<td>KSS original Grease MSG No.2</td>
</tr>
<tr>
<td>Lubricating Oil</td>
<td>Sliding surface Oil or turbine Oil</td>
<td>Super Multi 68</td>
</tr>
</tbody>
</table>

- Inspection and replenishment
  Grease inspection should be performed once every two to three months, and Oil inspection should be performed approximately weekly. Check the Oil or Grease amount and contamination at each inspection and replenish if needed.

  When re-greasing, the old or discolored one should be wiped off as much as you can.

  1) It is desirable to wear rubber gloves, not to handle Ball Screw by bare hand.
  2) Wipe off discolored Grease on the Screw Shaft by using cloth or paper exclusive for wiping Grease or oil (e.g.: Kim Wipes by Kimberly-Clark Corp.). Move the Ball Nut to wipe off remaining Grease inside the Ball Nut as much as possible.
  3) There is no oil hole on the flange for KSS Ball Screws as standard design, apply Grease entirely throughout the Screw Shaft. Please use the brush exclusive for applying Grease, or apply directly to the Screw Shaft by hand with wearing rubber gloves. If the Ball Nut has an oil hole, utilize it to fill in the new Grease.
  4) In order to apply Grease entirely on the Screw Shaft, move the Ball Nut over full travel manually, or install in the device and do running-in. Remove any remaining Grease on either end of the Screw Shaft.

Please consult KSS for details.

Inspection and replenishment Interval of lubricant

<table>
<thead>
<tr>
<th>Lubrication</th>
<th>Inspection frequency</th>
<th>Inspection Items</th>
<th>Replenishment and replacement frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Automatic intermittent</td>
<td>Weekly</td>
<td>Oil level, contamination</td>
<td>Replenish at each inspection, depending on tank capacity</td>
</tr>
<tr>
<td>Grease</td>
<td>Every 2 to 3 months initially</td>
<td>Contamination, smart contamination</td>
<td>Replenish annually or as necessary, depending on Inspection results</td>
</tr>
<tr>
<td>Oil bath</td>
<td>Daily before operation</td>
<td>Oil surface check</td>
<td>Set a rule for replenishment as necessary, depending on amount of wear</td>
</tr>
</tbody>
</table>

Grease-up Procedure (Example)

1) It is desirable to wear rubber gloves, not to handle Ball Screw by bare hand.

2) Wipe off discolored Grease on the Screw Shaft by using cloth or paper exclusive for wiping Grease or oil (e.g.: Kim Wipes by Kimberly-Clark Corp.). Move the Ball Nut to wipe off remaining Grease inside the Ball Nut as much as possible.

3) There is no oil hole on the flange for KSS Ball Screws as standard design, apply Grease entirely throughout the Screw Shaft. Please use the brush exclusive for applying Grease, or apply directly to the Screw Shaft by hand with wearing rubber gloves. If the Ball Nut has an oil hole, utilize it to fill in the new Grease.

4) In order to apply Grease entirely on the Screw Shaft, move the Ball Nut over full travel manually, or install in the device and do running-in. Remove any remaining Grease on either end of the Screw Shaft.

Please consult KSS for details.
Dust prevention

In Ball Screws, if dust or other contaminations intrude into the Ball Nut, wear is accelerated, the screw groove will be damaged, circulation will be obstructed due to Ball fracture, damage of recirculation parts and so on. Eventually, the Ball Screws will cease to function. Where the possibility of dust or other contaminant exists, the screw thread section cannot be left exposed, and dust prevention measure such as a bellows or Telescopic pipe must be taken.

KSS Ball Screws are concentrated on compact design for a feature of Miniature Ball Screw. Therefore, all models in the catalogue are the dimension without seals. Please inquire KSS if seals are required. Please note that Nut dimension may change due to seal installation. Some models cannot install the seals.

Surface treatment

Surface treatment can be possible for the purpose of rust prevention. Very Low temp. Black Chrome treatment (BCr) is KSS standard surface treatment for the purpose of rust prevention. Please inquire KSS if other surface treatments are needed.

- **Feature of KSS Ball Screws with Very Low temp. Black Chrome (BCr) coating**
  - Due to thin film thickness, mating part can be applicable with BCr.
  - Due to strict production management, film thickness can be treated equally and smoothness is kept.
  - High anti-rust ability is possible.
  - To improve sliding characteristics, BCr+fluorine resin coating is also available.

- **Examination data of anti-rust ability**
  Based on the salt spray corrosion test (JIS Z2371), anti-rust ability has been evaluated, as follows.

  - Standard test piece : 70mm × 150mm × 1mm (material: SPCC )
  - Data : Evaluated by appearance and rating number method after 24 hours of salt spray corrosion test. (The less number, the more corrosion)

<table>
<thead>
<tr>
<th>Sample</th>
<th>Rating number (Average)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sample A (BCr coating)</td>
<td>9.3</td>
</tr>
<tr>
<td>Sample B (R coating)</td>
<td>9~8</td>
</tr>
<tr>
<td>Sample C (M coating)</td>
<td>3~4</td>
</tr>
</tbody>
</table>

- **About RoHS compliance**
  The amount of hexavalent Chromium in KSS Very Low temp. Black Chrome (BCr) coating is less value than the based on RoHS regulation.
Traceability

KSS Ball Screws are manufactured from rigidly selected materials in our temperature controlled factory. They are manufactured using the latest production equipment, with consistent quality control supervision ranging from the production process to inspection and shipping. Certificate of inspection, Photo A-115, or Inspection report, Photo A-116 can be provided as your request. The Ball Screws produced by KSS have a serial number which is marked on the Nut (refer to the Photo A-117). Record of inspection and production trail which is in correspondence to a production number, are stored in KSS and inspection data can be retrieved by inquiry of a serial number.

However, some products may not be applicable for serial number, please ask KSS for more detail.

Calculation example of characteristic for Ball Screws.

Load direction and Preload will be taking into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018. Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B 1192-5.

Example 1 : Vertical Pick & Place

Ball Screw model and operating condition

Load direction (Ball contact point) should be considered in calculation of lifetime for Vertical axis application. Load direction is defined as plus for downward, and as minus for upward. The status of Ball contact point is indicated in Fig. A-118.

Calculation:

<table>
<thead>
<tr>
<th>Ball Screw spec.</th>
<th>Shaft dia. = φ10mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Lead = 10mm</td>
</tr>
<tr>
<td></td>
<td>Dynamic Capacity Ca = 3,300N</td>
</tr>
<tr>
<td></td>
<td>Total length = 180mm</td>
</tr>
<tr>
<td></td>
<td>Axial play = 20μm or less</td>
</tr>
</tbody>
</table>

Operating Pattern

Max Speed = 0.4m/sec

** 2,400 min−1 because of Lead 10mm

Acceleration & Deceleration time = 0.02 sec

**①②③④ in diagram above

Constant speed time = 0.2 sec

**⑤⑥ in diagram above

Halt time = 0.1 sec

**④⑤⑥ in diagram above

Cycle time = 0.68sec

Fig. A-118 : Load direction and Ball Contact condition
### Calculation of Basic Rating Life

Basic Rating Life is calculated in the following procedure.

1) Calculation of Load condition from Operating pattern

Load condition of each operating pattern which is numbered is as follows.

- **Down & Acceleration**
  \[ F_a = mg - ma = 10 \times 9.807 - 10 \times 20 = -101.9 \text{ (N)} \]

- **Down & Constant speed area**
  \[ F_a = mg = 10 \times 9.807 = 98.1 \text{ (N)} \]

- **Down & Deceleration**
  \[ F_a = mg + ma = 10 \times 9.807 + 10 \times 20 = 298.1 \text{ (N)} \]

- **Halt**
  \[ F_a = 0 \]

- **Up & Acceleration**
  \[ F_a = mg + ma = 10 \times 9.807 + 10 \times 20 = 298.1 \text{ (N)} \]

- **Up & Constant speed area**
  \[ F_a = mg = 10 \times 9.807 = 98.1 \text{ (N)} \]

- **Up & Deceleration**
  \[ F_a = mg - ma = 10 \times 9.807 - 10 \times 20 = -101.9 \text{ (N)} \]

- **Halt**
  \[ F_a = 0 \]

Here, 
- \( m \): Mass = 10 kg
- \( g \): Gravity Acceleration = 9.807 m/sec\(^2\)
- \( a \): Acceleration
  - Acceleration up to 0.4 m/sec\(^2\)
  - \( a = 0.4/0.02 = 20 \text{ m/sec}^2 \)

2) Calculation of Speed condition from Operating pattern

Speed condition (Revolution condition) of each operating pattern which is numbered is as follows.

Constant speed area(1, 3, 5, 7, 9): 
0.4 m/sec = 400 x 60 mm/min = 24,000 mm/min = 2,400 min\(^{-1}\)(Lead 10 mm)

Acceleration and deceleration area(1, 3, 5, 7, 9): 
as an average revolution above, 2,400/2 = 1,200 min\(^{-1}\)

### Calculation result of the load condition and speed condition (revolution) for each operating patterns are as below.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Axial load Fa(N)</th>
<th>Revolution NI(min(^{-1}))</th>
<th>Frequency of use t/sec</th>
</tr>
</thead>
<tbody>
<tr>
<td>Down &amp; Acceleration</td>
<td>—101.9</td>
<td>2,400</td>
<td>0.02</td>
</tr>
<tr>
<td>Down &amp; Constant speed</td>
<td>98.1</td>
<td>—</td>
<td>0.2</td>
</tr>
<tr>
<td>Down &amp; Deceleration</td>
<td>298.1</td>
<td>1,200</td>
<td>0.02</td>
</tr>
<tr>
<td>Halt</td>
<td>0</td>
<td>0</td>
<td>0.1</td>
</tr>
<tr>
<td>Up &amp; Acceleration</td>
<td>298.1</td>
<td>1,200</td>
<td>0.02</td>
</tr>
<tr>
<td>Up &amp; Constant speed</td>
<td>98.1</td>
<td>2,400</td>
<td>0.2</td>
</tr>
<tr>
<td>Up &amp; Deceleration</td>
<td>—101.9</td>
<td>1,200</td>
<td>0.02</td>
</tr>
<tr>
<td>Halt</td>
<td>0</td>
<td>0</td>
<td>0.1</td>
</tr>
</tbody>
</table>

### Table of Equivalent load, Equivalent revolution for in each Load direction (Ball contact point)

As we could calculate the applying load and direction in each operating pattern, now we calculate the Equivalent load and Equivalent revolution for each Load direction.

Calculation formula shown in page A825 will be used for calculating Equivalent load and Equivalent revolution.

\[
F_{am} = \left( \frac{F_a \cdot t_1 + F_a \cdot t_2 + F_a \cdot t_3 + \cdots + F_a \cdot t_n}{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + \cdots + N_n \cdot t_n} \right)^{1/2} N
\]

\[
N_{re} = \frac{N_1 \cdot t_1 + N_2 \cdot t_2 + N_3 \cdot t_3 + \cdots + N_n \cdot t_n}{t_1 + t_2 + t_3 + \cdots + t_n} \text{ min}^{-1}
\]

Now calculation table should be re-arranged as below by load direction, and Equivalent load and Equivalent revolution in each load direction are as follows.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Downward load</th>
<th>Upward load</th>
<th>Frequency of use t/sec</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial load Fa(N)</td>
<td>Revolution NI(min(^{-1}))</td>
<td>Axial load Fa(N)</td>
<td>Revolution NI(min(^{-1}))</td>
</tr>
<tr>
<td>Down &amp; Acceleration</td>
<td>—</td>
<td>101.9</td>
<td>1,200</td>
</tr>
<tr>
<td>Down &amp; Constant speed</td>
<td>98.1</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Down &amp; Deceleration</td>
<td>298.1</td>
<td>1,200</td>
<td>—</td>
</tr>
<tr>
<td>Halt</td>
<td>0</td>
<td>0</td>
<td>0.1</td>
</tr>
<tr>
<td>Up &amp; Acceleration</td>
<td>298.1</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>Up &amp; Constant speed</td>
<td>98.1</td>
<td>2,400</td>
<td>—</td>
</tr>
<tr>
<td>Up &amp; Deceleration</td>
<td>—101.9</td>
<td>—101.9</td>
<td>1,200</td>
</tr>
<tr>
<td>Halt</td>
<td>0</td>
<td>0</td>
<td>0.1</td>
</tr>
</tbody>
</table>

**Equivalence**

- Fam(d) = 129.3
- Nm(d) = 2,290.9
- Fam(u) = 101.9
- Nm(u) = 1,200

Working duration: 0.48 sec
Halt time: 0.2 sec
1 cycle: 0.68 sec
4) Calculation of Basic Rating Life for each Load direction (Ball contact point)
Then calculate the Basic Rating Life for downward load, upward load by using the value of Equivalent load, Equivalent revolution in each load direction (Ball contact point).

[Downward load]
Substitute the Equivalent Load Fam(d) and Revolution Nm(d) in the following formula in page A825.

\[ L_{\text{down}} = \left( \frac{C_a}{F_{\text{arm}}(d)} \right)^{1/3} \times \left( \frac{10^6}{60 \cdot Nm(d)} \right) = 69,991 \text{ hours} \]

Here, Basic Dynamic Load Rating Ca = 3,300N, Load factor f = 1.2.

[Upward load]
Calculate the upward load as same method as above.

\[ L_{\text{up}} = \left( \frac{C_a}{F_{\text{arm}}(u)} \right)^{1/3} \times \left( \frac{10^6}{60 \cdot Nm(u)} \right) = 272,988 \text{ hours} \]

5) Calculation of merged Basic Rating Life in each Load direction (Ball contact point)
Calculate the merged Basic Rating Life by combining the Basic Rating Life of each Load direction (\( L_{\text{down}} \), \( L_{\text{up}} \)), with the calculation formula of page A826.

\[ L'_{\text{down}} = (L_{\text{down}} - 10^3 + L_{\text{up}} - 10^3)^{-1/3} = 58,504 \text{ hours} \]

5) Calculation of total Life with taking halt time into consideration
Above calculation is only for the working duration, therefore calculate the total Life with taking halt time in each cycle into consideration.

\[ L''_{\text{down}} = L'_{\text{down}} \times \text{cycle time} / \text{working duration} = 58,504 \times \left( \frac{0.68}{0.48} \right) = 82,881 \text{ hours} \]

Calculation of Driving Torque for Linear Motion system
Calculate Driving Torque for Linear Motion system according to page A829. It is important for motor selection. In the above case, due to backlash type Ball Screw, Preload Dynamic Drag Torque does not occur. Therefore, calculate acceleration Torque \( T_1 \) and Load Torque \( T_2 \).

\[ T = T_1 + T_2 + T_3 + T_4 \text{ N m} \]

\[ T_1: \text{ Acceleration Torque } \text{ N m} \]
\[ T_2: \text{ Load Torque } \text{ N m} \]
\[ T_3: \text{ Preload Dynamic Drag Torque } \text{ N m} \]
\[ T_4: \text{ Additional Torque } \text{ N m} \]

1) Calculation of acceleration Torque \( T_1 \)
\[ T_1 = \alpha \cdot I = \alpha \left( \frac{\theta_2 + \theta_3}{2} \right) \text{ N m} \]
\[ \alpha : \text{ Angular acceleration rad/sec}^2 \]
\[ I : \text{ Inertia moment kg m}^2 \]
\[ \theta_2 : \text{ Inertia moment of moving object by motor axis conversion kg m}^2 \]
\[ \theta_3 : \text{ Inertia moment of Screw Shaft kg m}^2 \]
\[ \theta = \text{ Ball Screw Lead} = 0.01 \text{ m} \]
\[ \theta_2 = m_0 \times \left( \frac{d(2\pi)^2}{2} \right) = 2.53 \times 10^{-5} \text{ kg m}^2 \]
\[ m_0 : \text{ Mass of moving object} = 10 \text{ kg} \]
\[ \theta_3 = \text{ Ball Screw Lead} = 0.01 \text{ m} \]
\[ \theta_3 = m_0 \times \left( \frac{d}{2} \right)^2 \times \gamma \times L \times \left( \frac{d}{2} \right) = 0.139 \times 10^{-5} \text{ kg m}^2 \]
\[ m_0 : \text{ Mass of Screw Shaft} = 7.850 \text{ kg/m}^3 \]
\[ d : \text{ Shaft dia.} = 0.01 \text{ m} \]
\[ L : \text{ Shaft length} = 0.18 \text{ m} \]
\[ \theta = \frac{(2\pi N)}{60t} = 12,566.4 \text{ rad/sec}^2 \]
\[ N : \text{ Max speed} = 2,400 \text{ min}^{-1} \]
\[ t : \text{ Acceleration time} = 0.02 \text{ sec} \]
\[ T_1 = 12,566.4 \times (2.53 + 0.139) \times 10^{-5} = 0.335 \text{ N m} \]

2) Calculation of Load Torque \( T_2 \)
\[ T_2 = mg \theta / (2\pi N) = 0.173 \text{ N m} \]
\[ m : \text{ Mass of moving object} = 10 \text{ kg} \]
\[ g : \text{ Gravety acceleration} = 9.807 \text{ m/sec}^2 \]
\[ \theta : \text{ Ball Screw Lead} = 0.01 \text{ m} \]
\[ \eta : \text{ Ball Screw efficiency} = 0.9 \]

3) Calculation of Driving Torque \( T \) for Linear Motion system
In case without consideration of Torque by support Bearings, Driving Torque of Ball Screw is as follows.

\[ T = T_1 + T_2 = 0.335 \text{ N m} + 0.173 \text{ N m} = 0.508 \text{ N m} \]
Example 2: Horizontal desk top small lathe
Ball Screw model and operating condition

Calculation of permissible Axial load
1) Study of Buckling Load
Calculate Buckling load according to the following formula in page A815.

\[ P = \frac{a \times \pi^2 E I}{l} \]

Substitute permissible stress \( a = 0.5 \), Young's modulus \( E = 2.08 \times 10^5 \) MPa, Root diameter \( d = 10.6 \) mm, Fixed - Fixed mounting factor \( n = 4 \), mounting span \( l = 400 \) mm in formula above.

\[ P = 15,900 \text{ N} \]

It is more than maximum Load so that there is no problem.

2) Study of permissible Load for yield stress
Calculate permissible Load for yield stress based on the formula in page A815.

\[ P = a \times A \]

Substitute permissible stress \( a = 98 \) N/mm², Root diameter \( d = 10.6 \) mm in the formula above.

\[ P = 8,650 \text{ N} \]

It is more than maximum Load and there is no problem.

Calculation of permissible Revolution
Calculate permissible Revolution based on the formula in page A816

\[ N = B \times \frac{40 - \Delta^2}{2\pi} \times \sqrt{\frac{E I g}{l}} \text{ min}^{-1} \]

\[ I = \frac{\pi}{64} d^4 \text{ mm}^4 \]

Substitute safety factor \( B = 0.8 \), Young's modulus \( E = 2.08 \times 10^5 \) MPa, gravity acceleration \( g = 9.8 \times 10^{-2} \) m/sec², material specific gravity \( \gamma = 7.7 \times 10^{-3} \) N/mm³, Root diameter \( d = 10.6 \) mm, Fixed - Support mounting factor \( \lambda = 3.927 \), mounting span \( l = 400 \) mm in formula above.

\[ N = 10,000 \text{ min}^{-1} \]

Therefore, it is more than maximum Revolution and there is no problem.
Calculation of Basic Rating Life
Load direction and Preload will be taken into consideration when calculate the Basic Rating Life by JIS B1192-5, which was established in 2018. Therefore, KSS uses a calculation formula of Basic Rating Life for Miniature Ball Screws that is conformed to JIS B1192-5.

In case when preload is effective by oversized Ball, the contact condition of the Ball is 4 points as per Fig. A-111. As explained in page A827, total Life can be calculated after calculation of Rating Life at contact point A and B due to the change of initial contact condition under the preload caused by external load.

Calculation of Load condition for each operating pattern
Calculation of Speed condition for each operating pattern
Calculation of the Load condition, speed condition at left/right contact point.
Calculation of Equivalent load, Equivalent revolution at left/right contact point.
Calculation of total Life with taking halt time into consideration.
Calculation of merged Basic Rating Life in each contact point.
Calculation of Rating Life at each contact point.

1) Calculation of Load condition from Operating pattern
Load condition of each operating pattern which is numbered is as follows.

1) Forward Acceleration
   \[ F_{ai} = \mu mg + ma = 0.05 \times 10 \times 9.807 + 10 \times 0.25 = 74 (N) \]
2) Forward at constant speed area
   \[ F_{ai} = \mu mg = 0.05 \times 10 \times 9.807 = 4.9 (N) \]
3) Forward Deceleration
   \[ F_{ai} = \mu mg - ma = 0.05 \times 10 \times 9.807 - 10 \times 0.25 = 2.4 (N) \]
4) Halt
   \[ F_{ai} = 0 \]
5) Sat Turning
   \[ F_{ai} = \mu mg + Fa = 0.05 \times 10 \times 9.807 + 200 = 204.9 (N) \]
6) Halt
   \[ F_{ai} = 0 \]
7) Backward Acceleration
   \[ F_{ai} = - (\mu mg + ma) = - (0.05 \times 10 \times 9.807 + 10 \times 0.25) = -74 (N) \]
8) Backward at constant speed area
   \[ F_{ai} = - \mu mg = -0.05 \times 10 \times 9.807 = -4.9 (N) \]
9) Backward Deceleration
   \[ F_{ai} = - \mu mg + ma = -0.05 \times 10 \times 9.807 + 10 \times 0.25 = -2.4 (N) \]
10) Halt
    \[ F_{ai} = 0 \]

Here,
- \( m \): Mass = 10 kg
- \( g \): Gravity Acceleration = 9.807 m/sec^2
- \( a \): Acceleration
  - Acceleration which reaches up to 50mm/sec^2
  - \( a = 0.05/0.2 = 0.25 \) m/sec^2

2) Calculation of Speed condition from Operating pattern
Speed condition (Revolution condition) of each operating pattern which is numbered is as follows.

Constant speed area \((\circ, \bullet)\):
50mm/sec=50 \times 60 mm/min = 3,000 mm/min = 1,500 min^-1 (Lead 2mm)

Acceleration and deceleration area \((\circ, \bullet, \oplus, \ominus)\):
As above average revolution, 1,500/2 \approx 750 min^-1

Calculation result of the load condition and speed condition (revolution) for each operating patterns are as below.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Axial load Fail(N)</th>
<th>Revolution N(min^-1)</th>
<th>Frequency of use (sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1) Forward Acceleration</td>
<td>7.6</td>
<td>750</td>
<td>0.2</td>
</tr>
<tr>
<td>2) Forward at constant speed</td>
<td>4.9</td>
<td>1,500</td>
<td>3.0</td>
</tr>
<tr>
<td>3) Forward Deceleration</td>
<td>2.6</td>
<td>750</td>
<td>0.2</td>
</tr>
<tr>
<td>4) Halt</td>
<td>0</td>
<td>0</td>
<td>1.0</td>
</tr>
<tr>
<td>5) Sat Turning</td>
<td>204.9</td>
<td>60</td>
<td>7.5</td>
</tr>
<tr>
<td>6) Halt</td>
<td>0</td>
<td>0</td>
<td>0.5</td>
</tr>
<tr>
<td>7) Backward Acceleration</td>
<td>-7.4</td>
<td>750</td>
<td>0.2</td>
</tr>
<tr>
<td>8) Backward at constant speed</td>
<td>-4.9</td>
<td>1,500</td>
<td>3.3</td>
</tr>
<tr>
<td>9) Backward Deceleration</td>
<td>-2.4</td>
<td>750</td>
<td>0.2</td>
</tr>
<tr>
<td>10) Halt</td>
<td>0</td>
<td>0</td>
<td>0.5</td>
</tr>
</tbody>
</table>

3) Calculation of the Load condition at left/right contact point
Ball contact condition in 4 point between Balls and thread grooves by preload may changes by external load as shown in page 827(Fig. A-112). Based on the changed Elastic displacement, load applying on the contact point A and B will be calculated by formula below.

\[
F_{ai(A)} = F_{ai} \times (1 + \frac{|F_{ai}|}{2N \times F_{pr}})^{3/2} \\
F_{ai(B)} = F_{ai} - |F_{ai}|
\]

Here,
- \( F_{pr} \): Preloaded load = 95 N
- \( F_{ai} \): Axial load in each condition (N)
  - \( (A), (B) \): This means contact point

The calculation result of each load condition and revolution condition as per contact point A and B is shown in table A-120.
4) Calculation of Equivalent load, Equivalent revolution at left and right contact point

Load applying on contact point A and B is calculated under each operating condition, then Equivalent load and Equivalent revolution at each contact point will be calculated. However, the speed and frequency of use stay the same, only the load condition will be different. Calculation formula shown in page A825 will be used for calculating Equivalent load and Equivalent revolution.

\[
F_{\text{eq}} = \left( \frac{F_{1} \cdot t_{1} + F_{2} \cdot t_{2} + \ldots}{t_{1} + t_{2} + \ldots} \right)^{1/2} N
\]

\[
N_{\text{eq}} = \left( \frac{N_{1} \cdot t_{1} + N_{2} \cdot t_{2} + \ldots}{t_{1} + t_{2} + \ldots} \right) \text{min}^{-1}
\]

The axial load applying on contact point A and B for each condition, Equivalent load and Equivalent revolution are as follows.

Table A-120 : Load & Revolution condition at each contact point

<table>
<thead>
<tr>
<th>Condition</th>
<th>Axial load at contact pt. A</th>
<th>Axial load at contact pt. B</th>
<th>Axial load at contact pt. A</th>
<th>Axial load at contact pt. B</th>
<th>Revolution Nm (min(^{-1}))</th>
<th>Frequency of use ti (sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>E: Forward Acceleration</td>
<td>7.4</td>
<td>99.0</td>
<td>91.6</td>
<td>750</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>E: Forward at Constant speed</td>
<td>4.9</td>
<td>97.6</td>
<td>92.7</td>
<td>1,500</td>
<td>3.0</td>
<td></td>
</tr>
<tr>
<td>E: Forward Deceleration</td>
<td>2.4</td>
<td>96.3</td>
<td>93.9</td>
<td>750</td>
<td>0.2</td>
<td></td>
</tr>
<tr>
<td>E: Halt</td>
<td>0</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>0.5</td>
<td>0.2</td>
</tr>
<tr>
<td>E: Turning</td>
<td>204.9</td>
<td>222.3</td>
<td>17.4</td>
<td>60</td>
<td>7.5</td>
<td></td>
</tr>
<tr>
<td>E: Halt</td>
<td>0</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>0.5</td>
<td></td>
</tr>
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<td>0</td>
<td>-</td>
<td>0</td>
<td>0</td>
<td>0.5</td>
<td></td>
</tr>
<tr>
<td>Equivalence</td>
<td>Fam_{A}=109.0</td>
<td>Fam_{B}=94.0</td>
<td>N_{eq}=719.2</td>
<td>Working duration : 16.6 sec</td>
<td>Halt time : 2.0 sec</td>
<td>1 cycle : 16.6 sec</td>
</tr>
</tbody>
</table>

Note: Results of applying load at contact point A and B are all absolute number.

5) Calculation of Rating Life at each contact point

Calculate the Basic Rating Life at contact point A and B by using the value of Equivalent load, Equivalent revolution in each contact point A, B.

- Contact point A
  
  \[
  L_{10h}^{(A)} = \left( \frac{C_{A}}{F_{\text{eq}}^{(A)}} \right)^{1.5} \times \left( \frac{10^{6}}{60 \cdot N_{\text{eq}}} \right) = 71,029 \text{ hours}
  \]

- Contact point B
  
  \[
  L_{10h}^{(B)} = \left( \frac{C_{B}}{F_{\text{eq}}^{(B)}} \right)^{1.5} \times \left( \frac{10^{6}}{60 \cdot N_{\text{eq}}} \right) = 110,747 \text{ hours}
  \]

Here, Basic Dynamic Load Rating \(C_{A}=1,900\text{N}, \text{Load factor } f=1.2.\)

6) Calculation of merged Basic Rating Life in each contact point

Calculate merged Basic Rating Life \(L_{10h}^{(A/B)}\) by using formula in page A 826.

\[
L_{10h}^{(A/B)} = \left( \frac{L_{10h}^{(A)}}{60} \right)^{10/9} \left( \frac{L_{10h}^{(B)}}{60} \right)^{10/9} 
\]

\[
= 46,257 \text{ hours}
\]

7) Calculation of total Life with taking halt time into consideration

Above calculation is only for the working duration, therefore calculate the total Life with taking halt time into consideration.

\[
L_{10h}^{(A/B)} = 46,257 \times \left( \frac{46.6 \text{ sec}}{16.6 \text{ sec}} \right) = 52,594 \text{ hours}
\]