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.

- **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.

- **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.
The range of manufacturing for Ball Screws

The range of manufacturing for KSS Ball Screws is from \( \phi 1.8 \) to \( \phi 16 \)mm 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>Shaft nominal diameter</th>
<th>C0</th>
<th>C1</th>
<th>C3</th>
<th>C5</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>90</td>
<td>120</td>
<td>140</td>
<td>170</td>
</tr>
<tr>
<td>6</td>
<td>140</td>
<td>180</td>
<td>210</td>
<td>250</td>
</tr>
<tr>
<td>8</td>
<td>200</td>
<td>250</td>
<td>310</td>
<td>350</td>
</tr>
<tr>
<td>10</td>
<td>260</td>
<td>320</td>
<td>420</td>
<td>450</td>
</tr>
<tr>
<td>12</td>
<td>320</td>
<td>390</td>
<td>510</td>
<td>550</td>
</tr>
<tr>
<td>14</td>
<td>380</td>
<td>460</td>
<td>600</td>
<td>640</td>
</tr>
<tr>
<td>16</td>
<td>450</td>
<td>540</td>
<td>700</td>
<td>770</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Shaft nominal diameter</th>
<th>Maximum length</th>
</tr>
</thead>
<tbody>
<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 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 B1192 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 \( (\frac{2\pi}{72} \text{ rad}) \) over the Screw Shaft useful travel. Tolerance of each accuracy grades are shown in the Table A-83, 84, 85.

Note 1) If required length exceeds the number in 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.
Table A-83: Tolerance on specified travel (±e_u) and permissible travel variation (Vu) of precision Ball Screws (for positioning : C series)  Unit: μm

<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>±e_u</td>
<td>Vu</td>
</tr>
<tr>
<td>−</td>
<td>100</td>
<td>3</td>
<td>3</td>
<td>3.5</td>
</tr>
<tr>
<td>100</td>
<td>200</td>
<td>3.5</td>
<td>3</td>
<td>4.5</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>4</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 V_u, V_μ (for positioning : C series)  Unit: μm

<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>V_u</td>
<td>V_μ</td>
<td>V_u</td>
<td>V_μ</td>
</tr>
<tr>
<td>Permissible value</td>
<td>3.5</td>
<td>3</td>
<td>4</td>
<td>5</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Accuracy grade</th>
<th>Ct7</th>
<th>Ct10</th>
</tr>
</thead>
<tbody>
<tr>
<td>V_u</td>
<td>52</td>
<td>210</td>
</tr>
</tbody>
</table>

Tolerance on specified travel (ε_u) for Ct series is calculated as follows.

$$\varepsilon_u = 2 \times \frac{\text{V_u}}{\text{V_u}} \times \text{V_u}$$  (μm)

Japan Industrial Standard of Ball Screw (JIS B1192) was revised in 1997 and 2013 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 of 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.

Fig. A-86: Description of Run-out and location tolerances for Ball Screws

Japan Industrial Standard of Ball Screw (JIS B1192) was revised in 1997 and 2013 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. KSS conforms to JIS B1192-2013 and adopts C series for 0,1,3,5 grade, Cp, Ct series for 7,10 grade.
### Ball Screw

#### Technical description

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 A809～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.

#### 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 C0</td>
</tr>
<tr>
<td>8</td>
<td>3</td>
</tr>
<tr>
<td>12</td>
<td>4</td>
</tr>
</tbody>
</table>

![Fig. A-87 : Compensation of Radial Run-out](image)

#### 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 C0</td>
</tr>
<tr>
<td>8</td>
<td>2</td>
</tr>
<tr>
<td>12</td>
<td>2</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 C0</td>
</tr>
<tr>
<td>—</td>
<td>5</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 C0</td>
</tr>
<tr>
<td>—</td>
<td>20</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 C0</td>
</tr>
<tr>
<td>—</td>
<td>50</td>
</tr>
<tr>
<td>50</td>
<td>100</td>
</tr>
</tbody>
</table>

![Diagram](image)
### 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>—</th>
<th>8</th>
<th>12</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>125</td>
<td>0.015</td>
<td>0.015</td>
<td>0.015</td>
<td>0.020</td>
</tr>
<tr>
<td>125</td>
<td>200</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>200</td>
<td>315</td>
<td>0.035</td>
<td>0.025</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>315</td>
<td>400</td>
<td>—</td>
<td>0.035</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>400</td>
<td>500</td>
<td>—</td>
<td>0.045</td>
<td>0.035</td>
<td>0.020</td>
</tr>
<tr>
<td>500</td>
<td>630</td>
<td>—</td>
<td>0.060</td>
<td>0.045</td>
<td>0.020</td>
</tr>
<tr>
<td>630</td>
<td>800</td>
<td>—</td>
<td>—</td>
<td>0.060</td>
<td>0.020</td>
</tr>
<tr>
<td>800</td>
<td>1000</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.065</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>—</th>
<th>8</th>
<th>12</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>125</td>
<td>0.020</td>
<td>0.020</td>
<td>0.015</td>
<td>0.020</td>
</tr>
<tr>
<td>125</td>
<td>200</td>
<td>0.035</td>
<td>0.025</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>200</td>
<td>315</td>
<td>0.045</td>
<td>0.040</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>315</td>
<td>400</td>
<td>—</td>
<td>0.050</td>
<td>0.040</td>
<td>0.020</td>
</tr>
<tr>
<td>400</td>
<td>500</td>
<td>—</td>
<td>0.060</td>
<td>0.045</td>
<td>0.020</td>
</tr>
<tr>
<td>500</td>
<td>630</td>
<td>—</td>
<td>—</td>
<td>0.060</td>
<td>0.020</td>
</tr>
<tr>
<td>630</td>
<td>800</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.075</td>
</tr>
<tr>
<td>800</td>
<td>1000</td>
<td>—</td>
<td>—</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>—</th>
<th>8</th>
<th>12</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>125</td>
<td>0.025</td>
<td>0.025</td>
<td>0.025</td>
<td>0.020</td>
</tr>
<tr>
<td>125</td>
<td>200</td>
<td>0.035</td>
<td>0.035</td>
<td>0.035</td>
<td>0.025</td>
</tr>
<tr>
<td>200</td>
<td>315</td>
<td>0.045</td>
<td>0.040</td>
<td>0.040</td>
<td>0.030</td>
</tr>
<tr>
<td>315</td>
<td>400</td>
<td>—</td>
<td>0.060</td>
<td>0.050</td>
<td>0.040</td>
</tr>
<tr>
<td>400</td>
<td>500</td>
<td>—</td>
<td>0.065</td>
<td>0.050</td>
<td>0.040</td>
</tr>
<tr>
<td>500</td>
<td>630</td>
<td>—</td>
<td>—</td>
<td>0.070</td>
<td>0.055</td>
</tr>
<tr>
<td>630</td>
<td>800</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.070</td>
</tr>
<tr>
<td>800</td>
<td>1000</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.095</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>—</th>
<th>8</th>
<th>12</th>
<th>20</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>125</td>
<td>0.035</td>
<td>0.035</td>
<td>0.035</td>
<td>0.035</td>
</tr>
<tr>
<td>125</td>
<td>200</td>
<td>0.045</td>
<td>0.045</td>
<td>0.045</td>
<td>0.045</td>
</tr>
<tr>
<td>200</td>
<td>315</td>
<td>0.060</td>
<td>0.055</td>
<td>0.055</td>
<td>0.045</td>
</tr>
<tr>
<td>315</td>
<td>400</td>
<td>—</td>
<td>0.080</td>
<td>0.075</td>
<td>0.060</td>
</tr>
<tr>
<td>400</td>
<td>500</td>
<td>—</td>
<td>0.090</td>
<td>0.075</td>
<td>0.060</td>
</tr>
<tr>
<td>500</td>
<td>630</td>
<td>—</td>
<td>—</td>
<td>0.090</td>
<td>0.075</td>
</tr>
<tr>
<td>630</td>
<td>800</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.090</td>
</tr>
<tr>
<td>800</td>
<td>1000</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>0.120</td>
</tr>
</tbody>
</table>
### 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 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.**

### 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>8</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>—</td>
<td>0.60</td>
<td>0.055</td>
</tr>
<tr>
<td>125</td>
<td>0.075</td>
<td>0.065</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>0.100</td>
<td>0.080</td>
<td></td>
</tr>
<tr>
<td>315</td>
<td>0.100</td>
<td>0.080</td>
<td></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>0.170</td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>0.170</td>
<td>0.170</td>
<td></td>
</tr>
</tbody>
</table>

### 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>8</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>—</td>
<td>—</td>
<td>0.100</td>
<td>0.095</td>
</tr>
<tr>
<td>125</td>
<td>0.140</td>
<td>0.120</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>0.210</td>
<td>0.140</td>
<td></td>
</tr>
<tr>
<td>315</td>
<td>0.210</td>
<td>0.160</td>
<td></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>0.420</td>
<td></td>
</tr>
</tbody>
</table>

### Technical description

**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.

### Table A-99: Slenderness ratio to determine the Total Run-out

<table>
<thead>
<tr>
<th>Slenderness ratio</th>
<th>Total Run-out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Over</td>
<td>C7</td>
</tr>
<tr>
<td>—</td>
<td>0.080</td>
</tr>
<tr>
<td>40</td>
<td>0.120</td>
</tr>
<tr>
<td>60</td>
<td>0.200</td>
</tr>
<tr>
<td>80</td>
<td>0.320</td>
</tr>
</tbody>
</table>

**Slenderness ratio = L/u/d0**
- L: Useful travel (mm)
- u: 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</td>
<td>Carburizing and quenching</td>
</tr>
<tr>
<td>Nut</td>
<td>SCM415</td>
<td>Carburizing and quenching</td>
</tr>
</tbody>
</table>

Note: Hardness on table shows surface hardness of thread part.

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

Note: Hardness on table shows surface hardness of thread part.
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

\[ P = \sigma \times \frac{n \pi^2 E \cdot I}{L^2} \text{ N(kgf)} \]

- \( \sigma \) : Safety Factor 0.5
- \( E \) : Young's modulus 2.08 \times 10^5 N/mm² (MPa) (21,200kgf/mm²)
- \( I \) : Screw Shaft minimum moment of inertia of area

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

- \( d \) : 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

\[ P = \sigma \times A \text{ N(kgf)} \]

- \( \sigma \) : Permissible stress 98N/mm² (MPa) (10kgf/mm²)
- \( A \) : Screw Shaft minimum section area

\[ A = \frac{\pi}{4} d^2 \text{ mm}^2 \]

- \( d \) : 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

\[ N = \beta \times \frac{60 \cdot \lambda^2}{2\pi} \times \sqrt{\frac{E \cdot I \cdot g}{Y \cdot A \cdot L^2}} \text{ min}^{-1} (\text{rpm}) \]

- \( \beta \) : Safety Factor 0.8
- \( E \) : Young's modulus 2.08 \times 10^5 N/mm² (MPa) (21,200kgf/mm²)
- \( I \) : Screw Shaft minimum moment of inertia of area

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

- \( d \) : Screw Shaft Root diameter mm
- \( g \) : Gravity acceleration 9.8 \times 10^3 mm/sec²
- \( Y \) : Material specific gravity 7.7 \times 10^6 N/mm² (7.85 \times 10^6kgf/mm²)
- \( L \) : Mounting span distance mm
- \( A \) : Screw Shaft minimum section area

\[ A = \frac{\pi}{4} \cdot d^2 \text{ mm}^2 \]

- \( \lambda \) : Factor for Ball Screw mounting method
  - Supported—Supported \( \lambda = \pi \)
  - Fixed—Supported \( \lambda = 3.927 \)
  - Fixed—Fixed \( \lambda = 4.730 \)
  - Fixed—Free \( \lambda = 1.875 \)

● Rotation limits 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 rotation limits 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.
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>
<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>
<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>Accuracy grade</td>
<td>C0</td>
<td>C0-0</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>C1</td>
<td>C1-0</td>
<td>C1-02</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>C3</td>
<td>C3-0</td>
<td>C3-02</td>
<td>C3-05</td>
<td>C3-20</td>
</tr>
<tr>
<td></td>
<td>C5</td>
<td>-</td>
<td>-</td>
<td>C5-05</td>
<td>C5-20</td>
</tr>
<tr>
<td></td>
<td>C7</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>C7-20</td>
</tr>
<tr>
<td></td>
<td>C10</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</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)

**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.

**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.

**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.

---

*Fig. A-105 : Preload by oversized Balls*  
*Fig. A-106 : Spacer Balls*  
*Fig. A-107 : Dynamic Drag Torque measurement*
Rigidity in Linear Motion system

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

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

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

\[\text{Total Rigidity of Linear Motion system } K\]

\[
K = \frac{Fa}{\delta} \quad N/\mu m(\mu m/kgf)
\]

\(Fa\) : Axial load applied to Linear Motion system \(N(\text{kgf})\)
\(\delta\) : Elastic displacement of Linear Motion system \(\mu m\)

\textbf{Screw Shaft Rigidity } \(K_1\)

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

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

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

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

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

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

\(A\) : Screw Shaft minimum section area

\[
A = \frac{R}{4} \cdot d^2 \quad \text{mm}^2
\]

\(d\) : Screw Shaft Root diameter \(\text{mm}\)
\(E\) : Young’s modulus \(2.08 \times 10^5 \text{N/mm}^2 \text{(MPa)} \approx 21,200 \text{kgf/mm}^2\)
\(l\) : Axial distance between fixed point & Nut center \(\text{mm}\)
\(L\) : Mounting span distance \(\text{mm}\)

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

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

Fig. A-108 : Fixed-Free in axial direction

Fig. A-109 : Fixed-Fixed in axial direction
Nut Rigidity \( K_2 \)

(1) Rigidity of Single Nut with backlash

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 \( Ca \), please use the following formula. Please inquire KSS regarding theoretical Static Rigidity of model types which are not in dimension table.

\[
K'_2 = K_2 \times \left( \frac{Fa}{0.3Ca} \right)^{\frac{1}{2}} \ N/\mu m (kgf/\mu m)
\]

- \( K_2 \): Nut Rigidity in dimension table \( N/\mu m (kgf/\mu m) \)
- \( Fa \): Axial load \( N (kgf) \)
- \( Ca \): Basic Dynamic Load Rating \( N (kgf) \)

(2) Rigidity of preloaded Ball Nut

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, please use the following formula. In case of Preload type Ball Screws, Rigidity varies depending on the dispersion of Preload Dynamic Drag Torque. Therefore, please inquire KSS for details. KSS will calculate theoretical Static Rigidity of required Nut models, which are not in the dimension table.

\[
K'_2 = K_2 \times \left( \frac{Ga}{0.05Ca} \right)^{\frac{1}{2}} \ N/\mu m (kgf/\mu m)
\]

- \( K_2 \): Nut Rigidity in dimension table \( N/\mu m (kgf/\mu m) \)
- \( Ga \): Preload amount \( N (kgf) \)
- \( Ca \): Basic Dynamic Load Rating \( N (kgf) \)

Support Bearing Rigidity \( K_3 \)

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

Nut, Bearing fitting part Rigidity \( K_4 \)

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 relatively small compared to axial displacement. However, if investigation is required, the following formula may be used for calculation.

\[
\theta = \frac{32TL}{n0d^4} \times \frac{180}{\pi} \times 10^4 \ deg
\]

- \( \theta \): Torsion angle due to torsion moment \( \deg \)
- \( T \): Torsion moment \( N \cdot cm (kgf \cdot cm) \)
- \( L \): Distance between Nut & Shaft end support \( mm \)
- \( G \): Modulus of Rigidity \( 8.3 \times 10^4 N/mm^2 \) (MPa) (8,500 kgf/mm^2)
- \( d \): Screw Shaft Root diameter \( mm \)

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

\[
\delta a = \frac{\theta}{360} \times \frac{L}{\mu m}
\]

- \( \delta a \): Lead \( mm \)
Ball Screw

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 formula.

\[ L_{10} = \left( \frac{C_a}{F_a} \right)^{0.1} \times 10^6 \text{ rev} \]

Also, in place of the total number of revolutions, the Basic Rating Life can be expressed in hours \( L_{10h} \) or traveled distance \( L_{10} \) and these can be calculated through the following formulas.

\[ L_{10h} = \left( \frac{1}{60 \times N} \right) \times L_{10} \text{ hours} \]
\[ L_{10} = \left( \frac{\rho}{10^6} \right) \times L_{10} \text{ km} \]

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 \( F_{am} \), equivalent Revolution \( N_{am} \) in the following formula.

\[ F_{am} = \left( \frac{N_a \times t_1 + F_{a1} \times N_1 \times t_1 + F_{a2} \times N_2 \times t_1 + F_{a3} \times N_3 \times t_1}{N_1 \times t_1 + N_2 \times t_1 + N_3 \times t_1} \right)^{\frac{1}{5}} \text{ N[kgf]} \]
\[ N_{am} = \frac{N_1 \times t_1 + N_2 \times t_1 + N_3 \times t_1}{t_1 + t_2 + t_3} \text{ min}^{-1} (\text{rpm}) \]

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

\[ F_{am} = \left( \frac{F_{a\text{min}} + 2 \times F_{a\text{max}}}{3} \right) \text{ N[kgf]} \]

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.

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 \( F_{am} \) for the screw groove can be found by using the following formula.

\[ F_{max} = \frac{C_{oa}}{f_s} \text{ N[kgf]} \]

\( f_s \): Static safety factor

\( f_s = 1 \rightarrow 2 \) for normal operation
\( f_s = 2 \rightarrow 3 \) for vibration, impact

Hardness coefficient

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

\[ C'_{a} = \theta \times C_{a} \text{ (N)} \]
\[ C'_{oa} = \theta \times C_{oa} \text{ (N)} \]

\( \theta \), \( \theta' \): Hardness coefficient (See graph right)

![Graph of Hardness Coefficient](image)

Surface Hardness (HRC) vs. Hardness Coefficient

<table>
<thead>
<tr>
<th>Axial load ( N ) (kgf)</th>
<th>Revolution ( N ) ( \text{min}^{-1} ) (rpm)</th>
<th>Working time %</th>
</tr>
</thead>
<tbody>
<tr>
<td>( F_{a1} )</td>
<td>( N_1 )</td>
<td>( t_1 )</td>
</tr>
<tr>
<td>( F_{a2} )</td>
<td>( N_2 )</td>
<td>( t_2 )</td>
</tr>
<tr>
<td>( F_{a3} )</td>
<td>( N_3 )</td>
<td>( t_3 )</td>
</tr>
</tbody>
</table>
Driving Torque

Driving Torque $T$ in Linear Motion system is expressed according to the following formula.

$$T = T_1 + T_2 + T_3 + T_4 \quad \text{N·m (kgf·cm)}$$

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

When Motor selection, the Driving Torque in Linear Motion system is needed. $T_1 \sim T_4$ can be calculated by the following formula:

**Acceleration Torque $T_1$**

$$T_1 = \alpha \cdot I \quad \text{N·m}$$

$$\alpha = \frac{2\pi N}{60 \cdot t} \quad \text{rad/sec}^2$$

$$I = I_w \cdot \alpha^2 + I_s \cdot \alpha^2 + I_a \cdot \alpha + I_s \quad \text{kg·m}^2$$

$$I_w = m_w \cdot \left(\frac{\alpha}{2\pi N}\right)^2 \quad \text{kg·m}^2$$

$$I_s = m_s \cdot \left(\frac{d}{8}\right) \quad \text{kg·m}^2$$

$$m_w = \pi \left(\frac{d}{2}\right)^2 \cdot L \cdot y \quad \text{kg}$$

- $\alpha$: Angular acceleration $\quad \text{rad/sec}^2$
- $I$: Inertia moment $\quad \text{kg·m}^2$
- $I_w$: Inertia moment of moving object by Motor axial conversion $\quad \text{kg·m}^2$
- $I_s$: Inertia moment of Screw Shaft $\quad \text{kg·m}^2$
- $I_a$: Inertia moment of gears on screw side $\quad \text{kg·m}^2$
- $m_w$: Mass of moving object $\quad \text{kg}$
- $m_s$: Mass of Screw Shaft $\quad \text{kg}$
- $d$: Screw Shaft diameter $\quad \text{m}$
- $L$: Ball Screw length $\quad \text{m}$
- $y$: Specific gravity $\quad \text{7,850 kg/m}^3$
- $A$: Reduction ratio $\quad \text{min}^{-1}$
- $N$: Motor speed $\quad \text{sec}$

**Load Torque $T_2$**

$$T_2 = \frac{P \cdot \alpha \cdot A}{2\pi N} \times 10^{-3} \quad \frac{\alpha \cdot A \times 10^{-1}}{2\pi N} \quad \text{N·m}$$

- $P$: Axial load $\quad \text{N (kgf)}$
- $F$: Load $\quad \text{N (kgf)}$
- $W$: Weight of moving object $\quad \text{N (kgf)}$
- $\alpha$: Lead $\quad \text{mm}$
- $\mu$: Sliding surface friction coefficient $\quad \eta$: Efficiency 0.9
- $A$: Reduction ratio

**Preload Dynamic Drag Torque $T_3$**

$$T_3 = 0.05 \times (\tan \beta)^{-1} \times \frac{F \cdot \alpha \cdot A}{2\pi N} \times 10^{-3} \quad \text{N·m}$$

- $\beta$: Lead angle $\quad \text{deg}$
- $F$: Preload $\quad \text{N (kgf)}$
- $\alpha$: 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). Please refer to catalogue page B101 [Original Grease for Miniature Ball Screws].

**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.

**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 lubrication</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, swarf contamination</td>
<td>Replenish annually or as necessary, depending on Inspection results. The old or discolored grease should be wiped off before re-greasing.</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>

**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>
**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. 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 Black Chrome (BCr) coating**
  - Due to thin film thickness (2~3 μm), 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 x 150mm x 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 Cr⁴⁺ amount of KSS 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-112, or Inspection report, Photo A-113 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-114). 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.

Calculation example of characteristic for Ball Screws.

Calculation example of characteristic for Ball Screws are mentioned as follows. Each calculation example is modeled so that there is a case which is unrealistic.

Example 1 : Vertical Pick & Place

Ball Screw spec.
- Shaft dia. = Ø10mm
- Lead = 10mm
- Dynamic Capacity Ca = 3,300N
- Total length = 180mm
- Axial play = 20 μm or less

Operating Pattern
- Max Speed = 0.4m/sec
- 2,400 min^-1 because of Lead 10mm
- Acceleration & Deceleration time = 0.05 sec
- **①-②** in diagram above
- Constant speed time = 0.1 sec
- **③-④** in diagram above
- Halt time = 0.2 sec
- **⑤-⑥** in diagram above
- Cycle time = 0.8sec
Calculation of Basic Rating Life
Basic Rating Life is calculated in the following procedure.

1) Calculation of Load condition
Load condition of each operation pattern which is numbered is as follows.

① Down&Acceleration, ② Up&Deceleration :
\[ F_1 = mg - m \alpha \]
③ Constant Speed area :
\[ F_1 = mg \]
④ Down&Deceleration, ⑤ Up&Acceleration :
\[ F_2 = mg + m \alpha \]

\[ m \text{ : Mass } = 10 \text{ kg} \]
\[ g \text{ : Gravity Acceleration } = 9.807 \text{ m/sec}^2 \]
\[ \alpha \text{ : Acceleration} \]

Acceleration up to 0.4 m/sec
\[ \alpha = 0.4/0.05 = 8 \text{ m/sec}^2 \]

2) Calculation of Speed condition
Revolution of each operation pattern which is numbered is as follows.

Constant speed area (②, ③) :
\[ 0.4 \text{ m/sec} = 0.4 \times 60 \text{ m/min} = 24 \text{ m/min} \]
\[ = 2,400 \text{ min}^{-1} \text{(Lead 10mm)} \]

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

3) Calculation of equivalent Load, equivalent Revolution
Calculation based on the above, calculate the equivalent Load \( F_{\text{eq}} \) shown in catalogue page A825 and the equivalent Revolution \( N_{\text{eq}} \).

\[
F_{\text{eq}} = \left( \frac{F_1 \cdot t_1 + F_2 \cdot t_2 + F_3 \cdot t_3}{t_1 + t_2 + t_3} \right)^{\frac{1}{3}} \text{ N}
\]

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

<table>
<thead>
<tr>
<th>Operating Pattern</th>
<th>Axial load N</th>
<th>Revolution min^{-1}</th>
<th>time sec</th>
</tr>
</thead>
<tbody>
<tr>
<td>①, ⑦</td>
<td>( F_1 = 18.1 )</td>
<td>( N_1 = 1,200 )</td>
<td>( t_1 = 0.05 \times 2 = 0.1 )</td>
</tr>
<tr>
<td>③, ⑤</td>
<td>( F_2 = 98.1 )</td>
<td>( N_2 = 2,400 )</td>
<td>( t_2 = 0.1 \times 2 = 0.2 )</td>
</tr>
<tr>
<td>③, ⑤</td>
<td>( F_3 = 178.1 )</td>
<td>( N_3 = 1,200 )</td>
<td>( t_3 = 0.05 \times 2 = 0.1 )</td>
</tr>
<tr>
<td>Equivalent</td>
<td>( F_{\text{eq}} = 116.3 )</td>
<td>( N_{\text{eq}} = 1,800 )</td>
<td>0.8 sec</td>
</tr>
</tbody>
</table>

Total : 0.4 sec 1 cycle :
Halt time : 0.8 sec (50%)

4) Calculation of Basic Rating Life
Using equivalent Load and equivalent Revolution, Basic Rating Life is calculated according to the catalogue page A825.

Basic Rating Life \( L_{\text{B}10} = \left( \frac{10}{(60 \cdot N_{\text{eq}})} \right) \times (Ca/f \cdot F_{\text{eq}})^{\frac{1}{2}} \text{ hours} \)

\[ L_{\text{B}10} = 96,280 \text{ hours} \]

\[ f \text{ : Load coefficient (Assumption 1.3) } \]
\[ Ca \text{ : Basic Dynamic Load Rating } (3,300 \text{ N}) \]

Due to halt time is 50%,
\[ 96,280/0.5 = 192,560 \text{ hours operation.} \]
If 24 hours operation is premised.
\[ 192,560/24 = 8,023 \text{ days, it shows that enough life is kept.} \]
Calculation of Driving Torque for Linear Motion system

Calculate Driving Torque for Linear Motion system according to the catalogue page A827. It is important when 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 $T_2$.

$$ T = T_1 + T_2 + T_3 + T_4 \quad \text{N}\cdot\text{m} $$

$T_1$: Acceleration Torque \quad \text{N}\cdot\text{m}$

$T_2$: Load Torque \quad \text{N}\cdot\text{m}$

$T_3$: Preload Dynamic Drag Torque \quad \text{N}\cdot\text{m}$

$T_4$: Additional Torque \quad \text{N}\cdot\text{m}$

1) Calculation of acceleration Torque $T_1$

$$ T_1 = \alpha \cdot (l_w + l_s) \quad \text{N}\cdot\text{m} $$

$\alpha$ : Angular acceleration \quad \text{rad/sec}^2$

$l$ : Inertia moment \quad \text{kg}\cdot\text{m}^2$

$l_w$ : Inertia moment of moving object by motor axis conversion \quad \text{kg}\cdot\text{m}^2$

$l_s$ : Inertia moment of Screw Shaft \quad \text{kg}\cdot\text{m}^2$

$$ l_w = m_w \times (d/2\pi)^2 = 2.53 \times 10^{-3} \quad \text{kg}\cdot\text{m}^2 $$

$m_w$ : Mass of moving object = 10 kg

$d$ : Ball Screw Lead = 0.01 m

$$ l_s = m_s \times (d^2/8) = (d^2/8) \times 7850 \times 0.139 \times 10^{-3} \quad \text{kg}\cdot\text{m}^2 $$

$m_s$ : Mass of Screw Shaft

$\gamma$ : Specific gravity of Screw Shaft = 7,850 kg/m$^3$

$d$ : Shaft dia. = 0.01 m

$L$ : Shaft length = 0.18 m

$$ \alpha = (2\pi N/60N) = 5,026.5 \quad \text{rad/sec}^2 $$

$N$ : Max speed = 2,400 min$^{-1}$

$t$ : Acceleration time = 0.05 sec

$$ T_1 = 5,026.5 \times (2.53 + 0.139) \times 10^{-4} = 0.134 \quad \text{N}\cdot\text{m} $$

2) Calculation of Load Torque $T_2$

$$ T_2 = mg \times (d/2\pi) = 0.173 \quad \text{N}\cdot\text{m} $$

$m$ : Mass of moving object = 10 kg

$g$ : Gravity acceleration = 9,807 m/sec$^2$

$d$ : Ball Screw Lead = 0.01 m

$\eta$ : 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.

$$ T = T_1 + T_2 = 0.134 \quad \text{N}\cdot\text{m} + 0.173 \quad \text{N}\cdot\text{m} = 0.307 \quad \text{N}\cdot\text{m} $$
Calculation of permissible Axial load
1) Study of Buckling load
Calculate Buckling load according to the following formula in Catalogue page A815.

\[ P = \frac{q \cdot \pi^2 E \cdot I}{L^2} \text{ N} \]

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

Substitute safety factor \( q = 0.5 \),
Young’s modulus \( E = 2.08 \times 10^5 \text{ N/mm}^2 \) (MPa),
Root diameter \( d = 10.6 \text{ mm} \),
Fixed—Fixed mounting factor \( n = 4 \),
mounting span distance \( L = 400 \text{ mm} \) 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 catalogue page A815.

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

\[ A = \frac{\pi}{4} d^2 \text{ mm}^2 \]

Substitute permissible stress \( \sigma = 98 \text{ N/mm}^2 \) (MPa),
Root diameter \( d = 10.6 \text{ mm} \) 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 catalogue page A816

\[ N = \beta \times \frac{60 \cdot \lambda^2}{2 \pi} \times \sqrt{\frac{E \cdot I \cdot g}{\gamma \cdot A \cdot L^2}} \text{ min}^{-1} \text{ (rpm)} \]

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

\[ A = \frac{\pi}{4} d^2 \text{ mm}^2 \]

Substitute safety factor \( \beta = 0.8 \),
Young’s modulus \( E = 2.08 \times 10^5 \text{ N/mm}^2 \) (MPa),
gravity acceleration \( g = 9.8 \times 10^3 \text{ mm/sec}^2 \),
material specific gravity \( \gamma = 7.7 \times 10^3 \text{ N/mm}^3 \),
Root diameter \( d = 10.6 \text{ mm} \),
Fixed—Support mounting factor \( \lambda = 3.927 \),
mounting span distance \( L = 400 \text{ mm} \) 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

1) Calculation of Load condition according to the operating pattern diagram

<table>
<thead>
<tr>
<th>Operating Pattern</th>
<th>Axial load (F) N</th>
<th>Revolution (N) min⁻¹</th>
<th>time (t) sec</th>
<th>Percentage %</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Acceleration</td>
<td>216</td>
<td>1,250</td>
<td>0.25</td>
<td>5</td>
</tr>
<tr>
<td>2. Constant Speed</td>
<td>49</td>
<td>2,500</td>
<td>0.5</td>
<td>10</td>
</tr>
<tr>
<td>3. Deceleration</td>
<td>216</td>
<td>1,250</td>
<td>0.25</td>
<td>5</td>
</tr>
<tr>
<td>4. Turning</td>
<td>269</td>
<td>50</td>
<td>4.0</td>
<td>80</td>
</tr>
</tbody>
</table>

2) Calculation of equivalent Load, equivalent Revolution

According to catalogue page A825, equivalent Load Fam is as follows.

\[ Fam = \left( \frac{F_1N_1t_1 + F_2N_2t_2 + F_3N_3t_3 + F_4N_4t_4}{N_1t_1 + N_2t_2 + N_3t_3 + N_4t_4} \right)^{\frac{1}{2}} N \]

Substitute each number in table in the formula above, Fam = 166 N

In case of the equivalent Revolution, substitute each number in table in the following formula,

\[ N_{eq} = \left( \frac{N_1t_1 + N_2t_2 + N_3t_3 + N_4t_4}{t_1 + t_2 + t_3 + t_4} \right) = 415 \text{ min}^{-1} \]

3) Calculation of Basic Rating Life

Substitute the equivalent Load Fam and Revolution N_{eq} in the following formula, page A825 in catalogue.

\[ L_{10h} = \left( \frac{10^6}{60 \cdot N_{eq}} \right) \times \left( \frac{Ca}{1 \cdot \text{Fam}} \right) = 3.48 \times 10^3 \text{ hours} \]

Here, Basic Dynamic Load Rating Ca = 1900N, Load factor f = 1.2.