1. Introduction

The wide range and large amount of use of rolling bearings indicates their necessity and vital contribution to the performance of modern industries. The requirements on rolling bearings have become stricter every year. Now, user requirements include compactness and low cost in addition to maintaining or exceeding previous performance levels.

As computer technology progresses, CAE (Computer Aided Engineering) becomes popular and essential to shorten the lead time of product development. Designers of rolling bearings rely more and more on computers as a simulation tool to design better bearings to match user requirements within a short time. Rolling bearing static analysis with a computer is very popular. It is essential for the bearing designer to know basic information such as internal load distribution and deflection of the bearing. However, to create high performance bearings, the bearing designer also needs more detailed information such as bearing torque, PV value, and slip rate of rolling elements. In such cases, dynamic analysis of a rolling bearing is required. The dynamic analysis of rolling bearings encompasses friction analysis, so it is much more difficult than the static case. ADORE, which is a commercially available software package, is an example of a well-known dynamic rolling bearing analysis program. Researchers can analyze the performance of roller bearings as well as ball bearings with ADORE®. ADORE is a very advanced and sophisticated tool. Since ADORE is a program for transient analysis of bearings, it takes considerable time to find the steady state of the bearing performance.

We have developed a computer program package named “BRAIN” to simulate the kinematics and performance of rolling bearings under various running conditions. When running BRAIN on a PC, the calculation time is short, so BRAIN is expected to become a frequently used simulation tool. BRAIN consists of several separate programs that are applicable to angular contact ball bearings (including deep groove ball bearings and thrust ball bearings), four point contact ball bearings, radial cylindrical (or needle) roller bearings, thrust cylindrical (or needle) roller bearings, tapered roller bearings, and self-aligning roller bearings. Various outputs, such as running torque, roller skew angle, roller slippage, and PV values, can be calculated to assist in the design of bearings for advanced applications.

Several analyses were conducted as calculation examples of BRAIN. The orbital slippage of rollers was simulated for high speed application of a cylindrical roller bearing with the elliptic outer raceway (bi-lobe bearing). A self-aligning roller bearing was analyzed for wear on raceways. In addition, the running torque of a four point contact ball bearing and the skewing of the needle roller in a radial needle bearing were simulated and compared with experimental results.

2. Features of BRAIN

The name BRAIN came from the phrase “BeaRing Analysis In Nsk.” BRAIN is a software package to simulate the performance of rolling bearings. BRAIN consists of six programs as shown in Table 1, namely, BALTAC (Ball bearing), B4PTAC (Four point contact ball bearing), CYLTAC (Cylindrical roller bearing), TNBTAC (Thrust cylindrical roller bearing), TAPTAC (Tapered roller bearing), and SELTAC (Self-aligning roller bearing).
The performance of radial and thrust ball bearings is calculated with the BALTAC program, especially to evaluate the bearing torque (power loss) and PV value at the ring/ball contact area. For three point and four point contact ball bearings, B4PTAC, a special version of BALTAC, has been developed for the analysis of gothic-arch shaped ring grooves. B4PTAC is based on BALTAC, and it utilizes many of the same subroutines as BALTAC. CYLTAC and SELTAC can analyze double row bearings as well as single row bearings. Most types of rolling bearings can be analyzed by BRAIN.

BRAIN has several attractive features.

(1) BRAIN can analyze the attitude and the slippage of rolling elements. In conventional bearing analysis, an assumption of “pure rolling” is made, i.e. the “Jones control theory” for ball bearings and pure rolling between rollers and inner/outer rings. However, the assumption of “pure rolling” is not rigorous enough especially for high-speed applications. The gyroscopic moment and centrifugal force acting on rolling elements result in large slippage at the contact area. In addition, the skew of the rollers in a roller bearing is usually neglected in conventional analysis although it strongly influences bearing performance. In BRAIN, 4-5 degrees of freedom for rolling elements result in large slippage at the contact area. In addition, the skew of the rollers in a roller bearing is usually neglected in conventional analysis although it strongly influences bearing performance. In BRAIN, 4-5 degrees of freedom for rolling elements result in large slippage at the contact area. In addition, the skew of the rollers in a roller bearing is usually neglected in conventional analysis although it strongly influences bearing performance. In BRAIN, 4-5 degrees of freedom for rolling elements result in large slippage at the contact area.

(2) BRAIN can analyze a bearing under any loading condition. BRAIN adopts the model where the outer ring is fixed while the inner ring moves. The inner ring has all degrees of freedom except for the rotation around the shaft center line. However, it is easy to set constraint conditions on the inner ring movement in any direction. This function confers an advantage when calculating bearing performance under given misalignments and/or under a position preloading condition.

(3) BRAIN can analyze a deformed bearing. In many cases, a deformed shaft and housing influence bearing performance. For these cases, BRAIN can import the deformation data relating to the shaft and the housing calculated by FEA (finite element analysis). The deformation is considered as a change in the internal clearance of the bearing. Therefore, the internal clearance varies axially and/or circumferentially in BRAIN. This feature effects the bearing analysis especially for one that is assembled in an aluminum or thin housing.

(4) BRAIN can analyze a bearing with an arbitrary crowning profile on rollers, inner ring, and outer ring. In BRAIN, the slice method by Harris is used to consider the crowning profile. Since the slice method is an approximate calculation, it is not possible to evaluate the edge load adjacent to edge of the contact area. However, more exact analysis for edge load calculation requires inputting BRAIN’s calculation results, such as contact force, tilt, and the skew angle of the roller into software that solves contact problems.

Several important assumptions are made when using BRAIN.

(1) Inner ring and cage rotational speeds are constant.

(2) The cage center is fixed (no eccentricity).

(3) The orbital speed of rolling elements and the cage rotational speed are identical.

How BRAIN simulation works might be easily visualized in your mind’s eye as follows: Imagine that you are observing the bearing motion by a stroboscope whose frequency is adjusted to $\omega_{fc}$ (Z: number of rolling elements, fc: cage rotational frequency). The rolling elements stay at the same positions. The calculation model is similar to that of Harris.

3. Coordinate Systems

The coordinate frames defined in BRAIN are shown in Fig. 1. Inertial coordinate frame $X^*Y^*Z^*$ is fixed in space. When the outer ring is assumed to be stationary and fixed in a housing, the outer ring coordinate frame $XYZ$ is identical with the inertial coordinate frame $X^*Y^*Z^*$. External axial load is applied in the X direction. External radial loads are applied in the Y and Z directions. The shaft or housing rotates clockwise around the X axis. Each rolling element has its own local coordinate frame $x_jy_jz_j$, which rotates with the rolling element. All the coordinate systems are right handed ones. The azimuth angle of the the rolling element is defined by the angle between the Z axis and $z_j$ axis. For convenience, radial load is usually applied in the Z direction. Therefore, rolling elements have their maximum contact load at an azimuth angle equal to zero.

![Fig. 1 Coordinate systems of BRAIN](image-url)
4. Forces and Moments Acting on Rolling Elements

Fig. 2 shows forces and moments acting on rolling elements in a ball bearing. The contact forces between outer/inner ring and rolling elements are $Q_1$ and $Q_2$. The centrifugal force and gyroscopic moments are $Q_i$ and $M_{gy}, M_{gz}$, respectively. Gravity which is thought to be negligible in most applications is neglected in the analysis. Cage contact force $Q_{ca}$, friction $F_{cx}$ and $F_{cz}$ are considered at the cage pocket area.

Frictional forces work at all contact areas, such as $F_{w1}$ and $F_{w2}$. For the estimation of frictional force and moment at the ring/ball contact area, the elliptical contact area is divided into small slices. The frictional force and moment at the contact area are obtained by a numerical integration of frictional force on each slice. Coefficient of friction is determined by an empirical equation which is a function of the lubricant’s viscosity, rolling velocity, slide/roll ratio, and contact pressure. The empirical equation was derived from a friction database based on results from two-disk test machines. Coefficient of friction is also a function of oil film thickness, and it behaves like a Stribeck curve. Rolling resistance is estimated by Aihara’s equation, which uses Goksem and Hargreaves’ rolling traction force. The basis of the model about BALTAC is described in Aramaki et al. Coulomb friction or hydrodynamic friction is assumed for cage/rolling elements in contact with frictional forces $F_{cax}, F_{cay}$. Cage land friction $F_{cL}$ is estimated by a simple viscous model.

The same calculation is also conducted for three point and four point contact ball bearings though the additional contact points should be considered for these bearings.

Fig. 3 shows forces and moments acting on the rolling element in a tapered roller bearing. The half cone angle and half roller cone angle are $\alpha$ and $\beta$, respectively. The roller is sliced into thin disks. The contact forces $Q_1$ and $Q_2$ are obtained by numerically integrating the contact force of each thin disk to consider the crowning profile. In tapered roller bearings and self-aligning roller bearings, flange contact force $Q_f$ and frictional force $F_f$ are also considered. In the case of a self-aligning roller bearing with the floating ring, the floating ring has two degrees of freedom that are axial displacement and rotational speed. The frictional forces are estimated by numerical integration of the frictional force on each thin disk of the rolling element as well as by the estimation of contact forces.
5. Equilibrium of Forces and Moments

5.1 Ball bearings

From the balance of forces and moments, one can obtain the following equations for the rolling elements of the ball bearing shown in Fig. 2.

\[ 0 = -Q_{xj} \sin \alpha_{xj} + Q_{yj} \sin \alpha_{yj} + F_{w1j} \cos \alpha_{xj} - F_{w2j} \cos \alpha_{yj} + F_{xj} \]  \hspace{1em} (1)

\[ 0 = F_{y1j} - F_{y2j} + F_{d1j} + Q_{czj} \]  \hspace{1em} (2)

\[ 0 = -Q_{xj} \cos \alpha_{xj} + Q_{yj} \cos \alpha_{yj} + Q_{zj} - F_{w1j} \sin \alpha_{xj} + F_{w2j} \sin \alpha_{yj} + F_{zxj} \]  \hspace{1em} (3)

\[ I_B (\text{d} \omega_a / \text{d} t) = -M_{xj} - M_{yj} + M_{r1xj} + M_{r2yj} - D_a F_{czj} / 2 \]  \hspace{1em} (4)

\[ I_B (\text{d} \omega_a / \text{d} t) = -M_{yj} + D_a (F_{w1j} + F_{w2j}) / 2 \]  \hspace{1em} (5)

\[ I_B (\text{d} \omega_a / \text{d} t) = M_{yj} - M_{zj} - M_{r1zj} + M_{r2zj} + M_{czj} + D_a F_{czj} / 2 \]  \hspace{1em} (6)

where \( D_a \): ball diameter. Other symbols are cited on Fig.2.

Eq. (1) to Eq. (3) are force balances and Eq. (4) to Eq. (6) are moment balances with respect to \( \alpha_x, \alpha_y \), and \( \alpha_z \) directions respectively. Additional contact loads, frictional forces and moments should be considered with four point contact ball bearings. As shown in the above equations, the acceleration term (left hand side) in the force balance should be considered with four point contact ball bearings. As an example, the following equations are derived for the tapered roller bearing shown in Fig. 3.

\[ 0 = -Q_{x1j} + Q_{y1j} + Q_{z1j} + Q_{xzj} + Q_{yzj} + Q_{zzj} \]  \hspace{1em} (14)

\[ 0 = F_{y1j} + F_{y2j} + F_{d1j} + F_{d2j} + Q_{czj} \]  \hspace{1em} (15)

\[ 0 = -Q_{z2j} + Q_{x2j} + Q_{y2j} + Q_{x2zj} + Q_{y2zj} + Q_{yzj} + Q_{xzj} + F_{czj} \]  \hspace{1em} (16)

\[ I_x (\text{d} \omega_a / \text{d} t) = M_{x1j} + M_{x2j} + M_{y1j} + M_{y2j} + M_{r1j} + M_{r2j} + M_{czj} \]  \hspace{1em} (17)

\[ 0 = M_{y1j} + M_{y2j} + M_{z1j} + M_{z2j} + M_{xzj} \]  \hspace{1em} (18)

\[ 0 = M_{x1j} + M_{x2j} + M_{x2zj} + M_{y1j} + M_{y2j} + M_{xzj} \]  \hspace{1em} (19)

where \( M_t \): moment due to flange contact

\( M_c \): moment due to cage/roller contact

Excluding the \( x \) rotation, the other angular acceleration terms (on the left hand side) are ignored.

Eq. (8) is applicable even for the moment balance of a roller bearing. Double row type bearings can be calculated in CYLTAC (cylindrical roller bearings) and SELTAC (self-aligning roller bearings) so that Eq. (8) is applied to each cage when two cages are assembled in the bearing. One can also obtain the equilibrium equations for the inner ring in the same way as in Eqs. (9) – (13).

Some self-aligning roller bearings have floating rings. In this case, force and moment balance equations for the floating ring are necessary in addition to the above equations.

6. Calculation Examples

As already described, BRAIN can analyze the performance of most rolling bearings. In the present paper, several calculation results are shown as examples of BRAIN analysis.
6.1 The orbital slippage of a cylindrical roller bearing with the elliptic outer raceway for high speed application

Cylindrical roller bearings for aircraft main shaft engines often suffer from skidding damage due to large orbital slippage of rollers resulting from high speed and light load operating conditions. Therefore, radial preloading is sometimes applied to the bearing by means of an elliptic shape outer raceway (bi-lobe bearing) or triangle-like shape outer raceway (tri-lobe bearing) to prevent orbital slippage of rollers. The first example is the orbital slippage analysis of a bi-lobe cylindrical roller bearing. Cylindrical roller bearing analysis program CYLTAC was used for this calculation. The dimensions of the bearing and test condition are shown in Tables 2 and 3.

The analysis referred to the experiment is by Yamamoto and Ishihara10. The experiment by Yamamoto and Ishihara was for the circular cylindrical roller bearing specified in Table 3.

Fig. 4 shows experimental results and calculation results of the circular bearing and also calculation results of the bi-lobe bearing. The experimental results are shown with a dash line and the calculation results are shown with solid lines in the figure. The orbital slippage rate is defined as 0% when the roller orbital speed is identical to the theoretical speed (no slippage speed) and 100% when the speed is zero.

In the case of the circular bearing, both experimental and calculated orbital slippage rates increased with speed. The calculations simulated the orbital slippage very well especially at mid and high speed although some discrepancy appeared at low speed.

Then, the bi-lobe bearing was analyzed. Radial load was applied to the bi-lobe bearing along the major axis of the elliptic outer ring. The analysis was conducted under conditions of 15\(\mu\)m and -5\(\mu\)m (radial preloading) of the diametric clearance at the minor axis while keeping the diametric internal clearance at the major axis to 95\(\mu\)m. The orbital slippage rate of the bi-lobe bearing increased at low speed. After the peak of the orbital slippage rate appeared, it decreased with the increase in speed. The peak became lower and appeared at lower speed when the clearance was smaller.

Fig. 5 shows the contact load distribution between inner ring and rollers at 1 000 rpm and 15 000 rpm. The internal clearance at the minor axis is 15\(\mu\)m. The horizontal axis shows the roller position with zero at radial loading position and positive in rotating direction. The contact load at high speed was much heavier compared with that at low speed. The loading zone moved toward ±90 degrees when the speed increased. These phenomena were caused by the reduction of the clearance due to centrifugal expansion of the inner ring. Therefore, the reduction of the orbital slippage rate appeared in Fig. 4. BRAIN analysis considers the change in the internal clearance due to fitting including centrifugal expansion effect and thermal expansion effect. In the present analysis, the initial fit and temperature difference between the inner ring and the shaft were assumed to be zero. It should be noticed that the internal clearance at the minor axis in actual applications is usually set to much larger negative values to make certain heavy preload.

From this calculation example, BRAIN analysis is found to be effective for the performance analysis of the bearing which is installed with a misshapen housing or shaft caused by deformation or the rough production.

---

**Table 2** Dimensions of the calculated bearing

<table>
<thead>
<tr>
<th>Dimension</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing outside diameter</td>
<td>158 mm</td>
</tr>
<tr>
<td>Bearing bore diameter</td>
<td>111 mm</td>
</tr>
<tr>
<td>Pitch diameter</td>
<td>135 mm</td>
</tr>
<tr>
<td>Roller diameter</td>
<td>11 mm</td>
</tr>
<tr>
<td>Roller dlength</td>
<td>11 mm</td>
</tr>
<tr>
<td>The number of rollers</td>
<td>25</td>
</tr>
<tr>
<td>Diametric clearance</td>
<td>0.095 mm</td>
</tr>
</tbody>
</table>

**Table 3** Calculation (experimental) condition

<table>
<thead>
<tr>
<th>Condition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational speed</td>
<td>0 – 15 000 rpm</td>
</tr>
<tr>
<td>Radial load</td>
<td>160 N</td>
</tr>
<tr>
<td>Lubrication</td>
<td>Oil jet (9.2 mm²/s at 40°C)</td>
</tr>
<tr>
<td>Feeding rate</td>
<td>1.7 L/min</td>
</tr>
<tr>
<td>Oil temperature</td>
<td>26°C</td>
</tr>
</tbody>
</table>
6.2 Wear on the raceways of a self-aligning roller bearing

The next example is the wear on the raceways of a self-aligning roller bearing under extremely low speed and heavy load conditions. Under such conditions, the oil film between roller and raceway is not expected, so that wear would be the major problem. For this analysis, the self-aligning roller bearing program SELTAC was applied. The dimensions of the bearing and test condition are shown in Table 4.

The following procedure was conducted for the wear analysis.

1. Calculating the contact pressure and the sliding velocity distributions on the raceways at the initial stage.
2. By making the assumption that the wear depth is proportional to the pressure and the sliding velocity, and changing the raceway profile dependent upon the PV value (pressure × sliding velocity).
3. Calculating the pressure and the sliding velocity distributions on the changed raceway profile obtained at (2).
4. Repeating steps (2) and (3) 100 times.

The raceway profile of the inner ring was changed depending upon the circumferentially averaged PV since it was the rotating ring.

The calculated raceway profiles are shown in Fig. 6. The calculated bearing has a double row. However, this is the pure radial load condition so we show only one of two rows in the figure. +5mm of off-set is given as the outer ring profile. The axial distance of the horizontal axis is measured from the center of the raceway to the edge of the bearing so that it is negative at the center side (the center between the two rows) of the bearing.

Fig. 7 shows the PV at the maximum loading position of rollers before and after wear. Although the PV generally decreased after wear, it increased at the center side edge area. According to the wear progress at the center side edge area, rollers inclined along the worn profile of the raceways. Then, contact area moved toward the bearing center side so that the PV at the center side increased.

This calculation example shows that BRAIN is also applicable to the wear analysis of rolling bearings.

Table 4 Bearing dimensions and running conditions

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Bore diameter</th>
<th>160 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Outside diameter</td>
<td>270 mm</td>
</tr>
<tr>
<td></td>
<td>Roller diameter</td>
<td>20 mm</td>
</tr>
<tr>
<td>Radial load</td>
<td></td>
<td>300 kN</td>
</tr>
<tr>
<td>Rotation speed</td>
<td></td>
<td>2 rpm</td>
</tr>
<tr>
<td>Bearing temperature</td>
<td></td>
<td>80°C</td>
</tr>
<tr>
<td>Lubricant viscosity</td>
<td></td>
<td>Grease base oil 80 mm²/s</td>
</tr>
</tbody>
</table>

Fig. 5 Load distribution of rollers in an elliptical bearing

Fig. 6 Worn profile of raceways

Fig. 7 PV distribution on the outer ring raceway before and after wear
6.3 Four point contact ball bearing analysis

Four point contact ball bearings would generate high power loss and severe ring/ball wear due to large spin friction under heavy radial load and high speed conditions. However, their forward/reverse axial load supporting capability is very attractive in terms of cost and compactness. The estimation of the PV values at ball/ring contact points under radial/axial combined loading conditions is a difficult task in a conventional analysis. In this analysis, a running torque experiment was conducted to compare with the calculation results.

A four point contact ball bearing was used in the experiment. The dimensions of the test bearing are shown in Fig. 8. Two rings serve as inner rings, and each has one groove. 12 balls are inserted in the bearing. The experimental conditions are shown in Table 5. The speed range in the experiment was from 1,000 rpm to 10,000 rpm. Case A is a pure axial loading condition. Case B is a combined axial and radial loading condition.

Fig. 9 shows the PV value distribution with the ball azimuth $\varphi$ under a pure axial loading condition (case A). This PV value is the peak PV value (pressure $P \times$ slippage velocity $V$) along the major axis of elliptical contact between the ball and inner/outer ring. The horizontal axis shows a ball position that is zero at the radial loading position. Black plots in the figure show axial loading side grooves (outer ring No.2 and inner ring No.1 grooves) and white ones show unloading side grooves (outer ring No.1 and inner ring No.2 grooves). Under this load condition, a ball always makes only one contact point with the outer and with the inner ring. Therefore, PV values on the unloading side grooves are zero. In such cases, balls can rotate with the minimum spin friction. PV values at inner ring No.1 are about 0.2 [GPa m/sec].

Fig. 10 shows the PV value distribution under the combined axial and radial loading condition (case B). The PV value has its maximum at a radial loading position ($\varphi = 0$). The inner ring has a higher contact pressure and a higher PV value than the outer ring has. The maximum PV value is 1.5 [GPa m/sec], and it appears at the radial loading position of the inner ring groove No.1. The PV values under case B are much higher than those under case A. In the present calculation, the peak PV value in the ball/ring contact area appears near the edge on the major axis of the contact ellipse due to the spin velocity.

The comparisons of experiment and calculation results under case A and case B are shown in Fig. 11. Solid lines and broken lines in the figure show experimental results and calculated ones respectively. A schematic view of the experimental apparatus used to measure the running torque is shown in Fig. 12. Radial and axial loads were applied to the test bearing through hydrostatic bearings. As the inner ring rotates, the bearing's frictional torque works to turn the bearing housing. The force necessary to restrain the housing was measured and converted to bearing torque.
Since no radial load is applied under Case A, the test bearing works as a conventional angular contact ball bearing. Below about 3000 rpm, the torque increased rather sharply with speed. However, the slope of the torque became smaller as speed increased. Reduction of oil viscosity due to temperature rise seems to play an important role in this phenomenon. The experiment agrees with the calculation excellently. The discrepancy between the experiment and the calculation was below 10%.

The measured running torque under a radial/axial combined load (Case B) was about 30% higher than that under a pure axial load (Case A). Under Case B, the agreement between the experiment and the calculation is reasonable. The calculation seems to always be below the experiment. Although the maximum discrepancy between the experiment and the calculation is about 16%, the tendency of bearing torque to vary with speed is well-simulated by the calculation.

BRAIN can estimate PV value distribution at each contact area as well as bearing power loss. Therefore, optimization of four point contact ball bearings becomes feasible. The effectiveness of BRAIN will be enhanced by accumulating comparisons with experimental data.

6.4 Needle bearing analysis

Many needle bearings are used in industries thanks to their compactness. Due to the skew motion of the rollers, however, great power loss is a potential problem. In roller bearings, skewing is defined as an angular rotation of the roller axis in a plane tangent to its orbital direction with respect to the axis of the contacting ring according to Harris. The skew motion of the roller is influenced by many factors, such as lubricant and the geometry of the bearing. Although the estimation of the skew of the roller is difficult work, it is very important to analyze the bearing performance including the skew influence.

Roller skew angle and axial force generated by roller skewing in a single row needle bearing is analyzed by CYLTAC. Input data for the calculation is shown in Table 5.

A schematic view of the test part is shown in Fig. 13. Needle rollers were inserted without a cage between the housing and the shaft. The shaft was rotated and the housing was stationary. The external radial load and moment were applied to the housing. Two eddy current type gap sensors were aligned axially and mounted in the housing. The housing was fixed by a thin plate and strain gauges were attached to the thin plates. Roller skew angle was determined by the phase difference in peak signals from the two gap sensors. Axial force was measured by strain gauges on a thin plate. A positive sign for a skew angle means that a roller passed over sensor 2 before the roller passed over sensor 1. The directions of shaft rotation and axial force are also shown in Fig. 13.

![Fig. 11 Running torque of four point contact ball bearing](image1)

![Fig. 12 Running torque test rig](image2)

![Fig. 13 Schematic view of the test part](image3)
Fig. 14 shows the skew angle of the roller at the maximum loading position (azimuth = 0 degree). The black plots in the figure show the calculated skew angles and the white ones show the measured results. The absolute values of measured skew angle are also shown in the figure. In the experiment, the roller skew angle changed from positive to negative suddenly at around 3 000 rpm. The absolute values of skew angle were at around 1 degree and almost constant in the speed range of 1 000 – 6 000 rpm. In the calculation, the roller skew angle is around +1 degree. It is still not clear why the roller changed its sign at 3 000 rpm in the experiment. The rollers might have two stable positions (+1 degree and -1 degree) for the roller skew motion.

Fig. 15 shows the axial force generated by the skew motion of the roller. For reference, it also shows the negative of the measured axial forces for the region above 3 000 rpm. The absolute value of the axial force decreased with increasing speed. Though the signs differed in the region above 3 000 rpm, the magnitude of the experimental and calculated values showed good agreement.

### 7. Conclusion

A new rolling bearing analysis program package named BRAIN has been developed. BRAIN software gives results in several minutes when run on a PC. BRAIN can calculate the kinematics and the performance of most rolling bearings. The calculation results are useful, especially for advanced applications involving rolling bearings. The effectiveness of BRAIN will be enhanced by accumulating comparisons with experimental data. Expectations are very high for BRAIN to meet the requirements of continuously advancing technology.

### References:
9) Aramaki, H., Shoda, Y., Morishita, Y., Sawamoto, T.  
“The Performance of Ball Bearing with Silicon Nitride  
Ceramic Balls in High Speed Spindles for Machine Tools,”  

10) Yamamoto, S., and Ishihara, S., “Wear of Rolling Element  
Bearings-Skidding Damage of High-Speed Cylindrical Roller  