Development of Bearing Dynamics Analysis System

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In recent years, bearings are required to be used in harsh environments such as complex loads and high-speed rotation. In order to understand the bearing behavior under actual operating conditions, JTEKT has developed a bearing dynamics analysis system (S.S.A.P./MBD). This system is based on multi-body dynamics. We used a method of constructing independent equations of motion for each part and solving them by applying the numerical integration method. PV value prediction in a contact ellipse by spin motion analysis and consideration of cage strength for needle roller bearings of planetary gears are provided here as application examples.

Key Words: *multi-body dynamics analysis, bearing, spin behavior, control, PV value*

1. Introduction

In recent years, automakers and various industrial equipment manufacturers are accelerating their efforts toward electrification and energy-saving, and this has led to demands for even lower torque and higher rotational speed in the rolling bearings used in such areas. Moreover, bearings are being used in harsher environments than ever before, such as rapid acceleration/deceleration and complex load fluctuation, therefore durability and performance in such operating environments must be guaranteed. In this way, there is an increasing number of opportunities for bearings to be used in conditions never before experienced, however at the same time, a demand has arisen for shorter bearing design lead-time in order to respond to customer requirements in a timely manner. Due to such changes in the external and internal environments, a great deal of attention is being focused on development processes utilizing analysis/simulation, as opposed to the conventional development processes which focused on prototyping and experimentation.

In an effort to respond to such environmental changes, JTEKT has independently developed Shaft System Analysis Program (S.S.A.P.) as software supporting bearing design[1), 2)]. The main functions of S.S.A.P. is shown in **Fig. 1**. S.S.A.P. enables modelling on a unit level passing through multiple shafts, gears, and bearings. Also, through bearing analysis considering power flow, housing stiffness, shaft stiffness, etc., it is possible to study bearing life, internal load distribution, friction torque and other bearing internal load conditions. Meanwhile, due to the background described at the outset, there is concern regarding problems such as damage resulting from impact and bearing internal behavior under complex operating conditions. These cannot be grasped using the current static analysis, therefore there is an increasing number of cases where dynamic analysis is required. As such, an important factor in bearing design is grasping bearing behavior in actual operating conditions from the initial design stage.

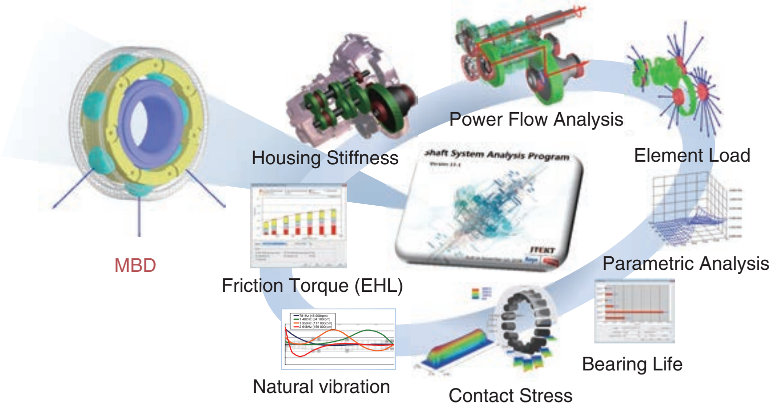

Fig. 1 Main functions of S.S.A.P.

Due to this rising need for bearing analysis utilizing dynamic analysis (simulation), JTEKT developed a bearing dynamics analysis system (S.S.A.P./MBD) as a new function of S.S.A.P. This report provides an overview of S.S.A.P./MBD and introduces use cases.

2. System Overview

2. 1 Analysis Method

S.S.A.P./MBD is a 3D dynamic analysis system for bearings built based on a multi-body dynamics (MBD) concept. In MBD, an equation of motion for each part is established for a machine system comprising multiple parts, and this makes it possible to decipher factors such as reciprocal action between parts difficult to measure, as well as the displacement and speed of individual parts. **Figure 2** shows the analysis method of S.S.A.P./MBD.

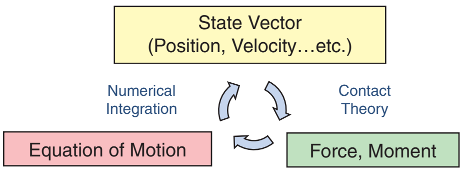

Fig. 2 Analysis method of S.S.A.P./MBD

First, an independent state vector (equation (1) is given to each part that comprises a bearing.

$$\vec{Y}(t)=\begin{bmatrix}\vec{x}(t)\\ q(t)\\ \vec{P}(t)\\ \vec{L}(t)\end{bmatrix} \qquad (1)$$

However,
$\vec{x}$: position [m]
q : orientation
$\vec{P}$: momentum [kg·m/s]
$\vec{L}$: angular momentum [N·m·s]
t : time [s]

Next, the force and moment working on individual parts in this state are calculated. One example of typical reciprocal action is contact force between parts. We used the Hertz's contact method to calculate contact force, whereby geometric interference is derived from the positional relationship of individual parts in a 3D space, and the quantity through an oil film is defined as the elastic approach. In this way, an equation of motion is established to decipher the force and moment working on each part. Equations (2) and (3) are examples of the equations of motion for rolling elements.

$$\frac{d\vec{P}_{\mathrm{Ball(i)}}}{dt}=\vec{F}_{\mathrm{Inner}}+\vec{F}_{\mathrm{Outer}}+\vec{F}_{\mathrm{Cage}}+\vec{F}_{\mathrm{Lub}}+\vec{F}_{\mathrm{G}} \qquad (2)$$

$$\frac{d\vec{L}_{\mathrm{Ball(i)}}}{dt}=\vec{N}_{\mathrm{Inner}}+\vec{N}_{\mathrm{Outer}}+\vec{N}_{\mathrm{Cage}}+\vec{N}_{\mathrm{Lub}} \qquad (3)$$

However,
$\vec{F}$: force [N]
$\vec{N}$: moment [N·m]
Superscript (indicates reciprocal action) :
Inner : inner ring
Outer : outer ring
Lub : Liquid/lubrication
G : Gravity

As equations (2) and (3) show, consideration is made toward the force and gravity that are generated due to fluid and lubricant, such as the reciprocal action, rolling viscous resistance, and mixing resistance of each part, and an equation of motion is established for the 6 degrees of freedom of translational motion and rotation for each part. By solving the equation of motion established in this way using numerical integration, the state vector of the next step is obtained. From this point onward, by repeating this cycle, it is possible to obtain the state vector of each part in every moment and calculate bearing behavior.

2. 2 Flow/function of Analysis

Figure 3 shows the basic flow when conducting a bearing study using S.S.A.P./MBD.

First, a static analysis with a shaft system model simulating the target product is used to grasp the radial and axial loads working on the bearing being studied, as well as the clearance change, raceway distortion, etc. caused by engagement. The data obtained in static analysis is carried on in the input data of dynamic analysis and the data required for dynamic analysis, such as phenomenon time, is also inputted. Moreover, in order to set the friction coefficient between the rolling element and raceway, we measured the relationship between the slip ratio against the circumferential speed under various pressure using typical lubricant oil and the friction coefficient (traction curve). We formed a database from this to enable automatic setting. **Figure 4** shows an example of traction curve.

Also, in regards to the contact damping coefficient between parts, we created a database by combining various materials, lubricant oils, and grease through independent contact damping tests and simplified data entry. This system supports the bearing varieties listed in the catalog, as well as offers functions to enable analysis of factors not easy to perform with general-purpose software, such as various bearing load and fluctuation as analysis conditions, misalignment, excitation, and other operating conditions, in addition to high-precision

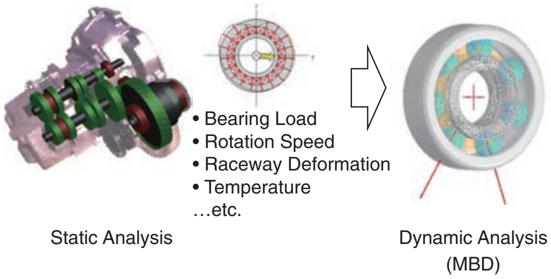

Fig. 3 Analysis flow

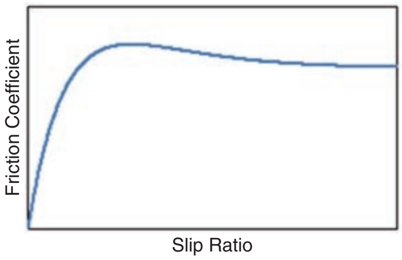

Fig. 4 Traction Curve

calculation of raceway circularity collapse and edge contact (**Fig. 5**). After completion of analysis, bearing behavior can be visually confirmed in the form of an animation analysis on the result processing window. Furthermore, data helpful toward bearing design such as impact load and friction work can be confirmed through linkage with the animation. **Figure 6** shows the result processing window for S.S.A.P./MBD.

3. Use Cases of S.S.A.P./MBD

3. 1 Spin Analysis of Ball Bearings

The first use case of S.S.A.P./MBD presented here is optimization of internal design through ball bearing spin analysis. As **Fig. 7** shows, in the case of ball bearings that operate with a contact angle (e.g. angular contact ball bearings, deep-groove ball bearings which are subjected to axial load), spin and gyro rotation of the balls is generated geometrically. This becomes friction resistance between the rolling element and raceway, and is the cause of increased ball bearing torque or raceway friction/seizure in high-speed rotation zones.[3] As such, in order to develop ball bearings that are both high-speed and low torque, it is important to accurately grasp the spin behavior of balls. The Jones' model[4] is a theory describing ball bearing kinematics, however it assumes the ideal state of the ball's slip motion, therefore a deviation occurs with reality in high-speed rotation zones as it is no longer possible to ignore gyro motion[5]. As S.S.A.P./MBD is unrestricted 3D dynamic analysis, it enables analysis of slip motion between balls and the raceway during high-speed rotation which entails spin and gyro slip.

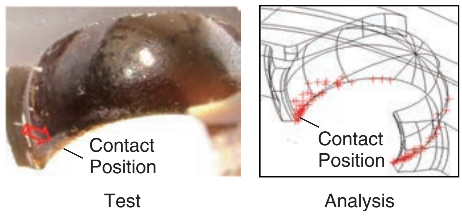

Fig. 5 High-precision calculation of edge contact

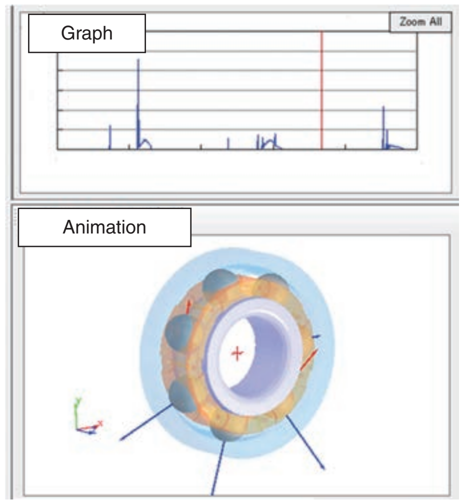

Fig. 6 Result processing window

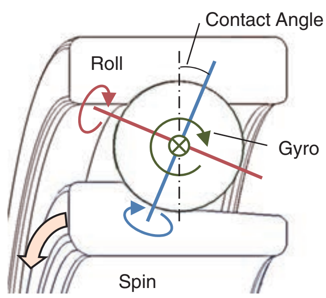

Fig. 7 Spin and gyro rotation

Figure 8 shows the calculation results for relative slip velocity distribution between the balls and raceway of a ball bearing in low-speed/high-speed rotational areas.

It is clear that, for inner ring rotating ball bearings, relative slip velocity occurs due to spin and gyro on the outer ring side during low-speed rotation, and the relative slip velocity on the inner ring side is lower than that of the outer ring side. This shows that the balls are driven on the inner ring side during low-speed rotation or, in other words, inner ring control is in effect. Meanwhile, relative slip velocity occurs on the inner ring side during high-speed rotation, and the relative slip velocity on the outer ring side is smaller, therefore it is clear that outer ring control is in effect. These analysis tendencies are confirmed to be consistent in actual measurement and theory. Furthermore, as a point that differs to conventional theory, the relative slip velocity within the contact ellipse is not even fully eliminated on the control side; rather a slight amount is generated, therefore it is believed that free ball movement is successfully analyzed.

Figure 9 shows the calculation results of PV value distribution within the contact ellipse and raceway friction actual measurement for bearings of the same size with balls made of steel and ceramic, respectively. If the balls are made from bearing steel, raceway friction occurs during high-speed rotation and the PV value increases. Meanwhile, when ceramic balls with small mass are used, raceway friction is suppressed during actual measurement, and it was even confirmed through analysis that the PV value in the raceway dropped. In this way, by using S.S.A.P./MBD to analyze ball slip motion under actual operating conditions it is possible to predict raceway friction and seizure, as well as optimize internal specifications.

3. 2 Study into Strength of Cage for Needle Roller Bearing Used in Planetary Gears

Needle roller bearings used in planetary gears as the planetary gear mechanism for transmissions (**Fig. 10**) perform complex motions not only involving self-revolution, but also centrifugal force working on the bearing so that the bearing itself can revolve, as well as entail load and moment from the gear. In such an environment, there is greater load on the cage, and a concern of cage breakage exists. It is not possible to grasp internal impact load amongst complex motion by applying the existing static analysis techniques, and there is a need for simulation using this system.

Figure 11 shows the test results for cage breakage against rotational speed of the planetary gear and carrier. As **Fig. 11** shows, we confirmed in a planetary gear durability test that cage fatigue breakage occurs on a needle roller bearing during high-speed rotation of planetary gears and carriers. Next, using S.S.A.P./MBD,

Bearing	Angular Contact Ball Bearing $\phi 20\times\phi 47\times 14$
Load	Fa=100N, Fr=0N
Rotational Speed	3 000, 10 000min^{-1}

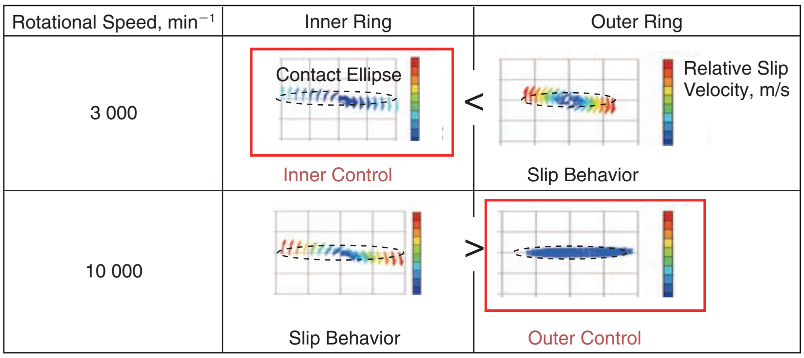

Fig. 8 Relative slip velocity distribution

Bearing	Angular Contact Ball Bearing $\phi 55\times\phi 90\times 18$
Load	Fa=160N, Fr=0N
Rotational Speed	12 000min^{-1}

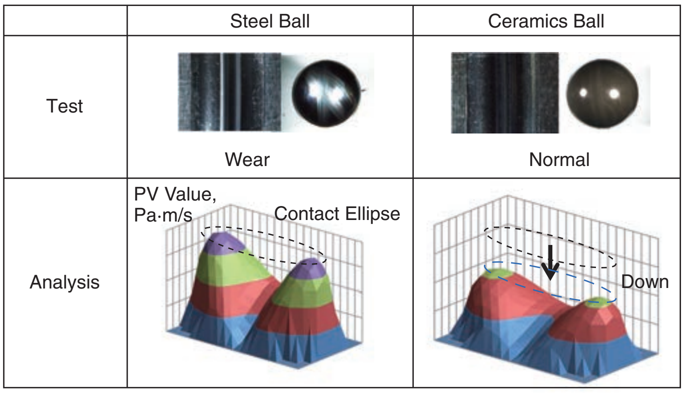

Fig. 9 PV value comparison

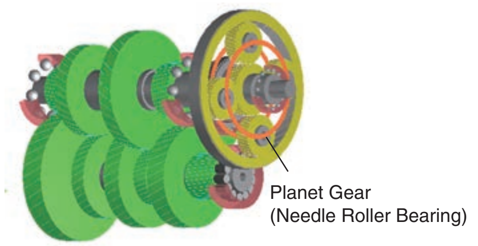

Fig. 10 Planetary gear for transmission

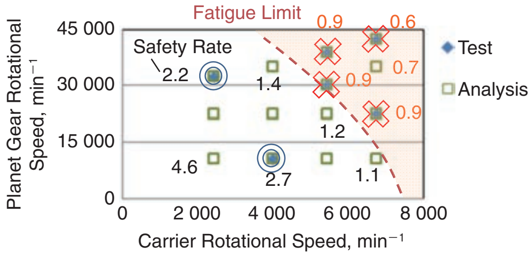

Fig. 12 Comparison of cage safety rate and test

in addition to the same conditions as the planetary gear durability test, conditions allocating parameters were used to conduct a simulation, and an analysis was made of the contact load and generated stress in the cage. **Figure 12** shows the results of the safety rate ultimately calculated relative to the fatigue limit of the target cage. As **Fig. 12** shows, the cage fatigue breakage tendency clarified through analysis was consistent with actual measurement.

Moreover, for the sake of optimal design, we confirmed bearing behavior during high-speed rotation. **Figure 13** shows bearing behavior during high-speed rotation of a planetary gear. It was clarified that, at high-speed rotation, the centrifugal force due to revolution speed is higher than gear load, and rolling element load distribution shifts toward the carrier diameter direction. At this time, in the phase that the rolling element departs the load zone, the planetary gear self-revolution direction and centrifugal force direction match, and the rolling element which has become free collides with the cage, generating a large contact load. In the reverse phase, contact load was generated by the cage pressing the rolling element. Considering this kind of mechanism, it is feasible that reducing weight of the rolling element, optimizing the operating clearance, etc. would be effective in reducing the contact load on the cages of needle roller bearings for planetary gears. Hence, as a result of reexamining internal specifications such as the cage, we discovered that contact load on the cage could be reduced by up to 79% compared to standard design, and confirmed the ability to support rotational zones of even higher speed. **Figure 14** shows the results of bearing design optimization. By utilizing S.S.A.P./MBD in this way to study cage strength under complex operating conditions, it is possible to prevent cage fatigue breakage.

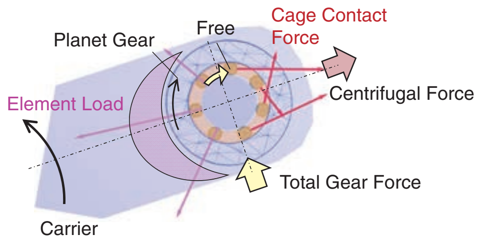

Fig. 13 Bearing behavior at high speed rotation

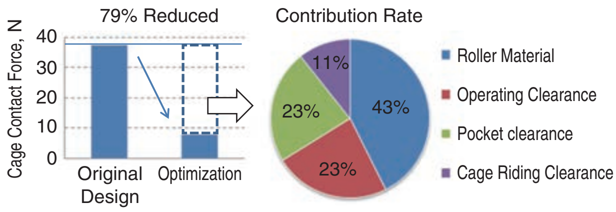

Fig. 14 Optimization of bearing design

4. Conclusion

This report has provided an overview and use cases of S.S.A.P./MBD as a bearing dynamics analysis system enabling dynamic analysis of high accuracy bearing behavior, and contributing to more efficient bearing development and optimal bearing design. JTEKT has deployed S.S.A.P. to all of its engineering divisions, including global bases, and built a system enabling all engineers to perform theoretical bearing simulations. As engineers can utilize this system to immediately verify their development ideas and actually visualize bearing behavior, there are expectations that it will contribute to the creation of innovation. In regards to simulations, depending on the various conditions selected, sometimes it is difficult to derive results precisely matching actual

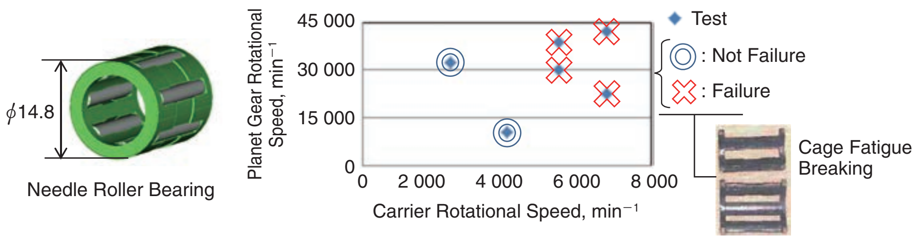

Fig. 11 Retainer breakage tendency of bearings for planetary gears

measurement, however the greatest benefit is being able to deepen understanding of phenomena by grasping the tendencies of product performance. As model-based design advances, it is anticipated that it will be utilized in an increasing number of cases moving forward. Hence, we expect that there will be greater expectations on simulation technology also in regards to demands for longer bearing life, lower torque, and higher speed.

Moreover, in regards to the S.S.A.P. static analysis function, we also offer a version of S.S.A.P. for customers with limited functions (as shown in **Fig. 15**), and this is popular amongst customers for the purpose of bearing selection. JTEKT will continue efforts to sophisticate various analysis technologies including S.S.A.P. and contribute to our customers through bearing development matching needs.

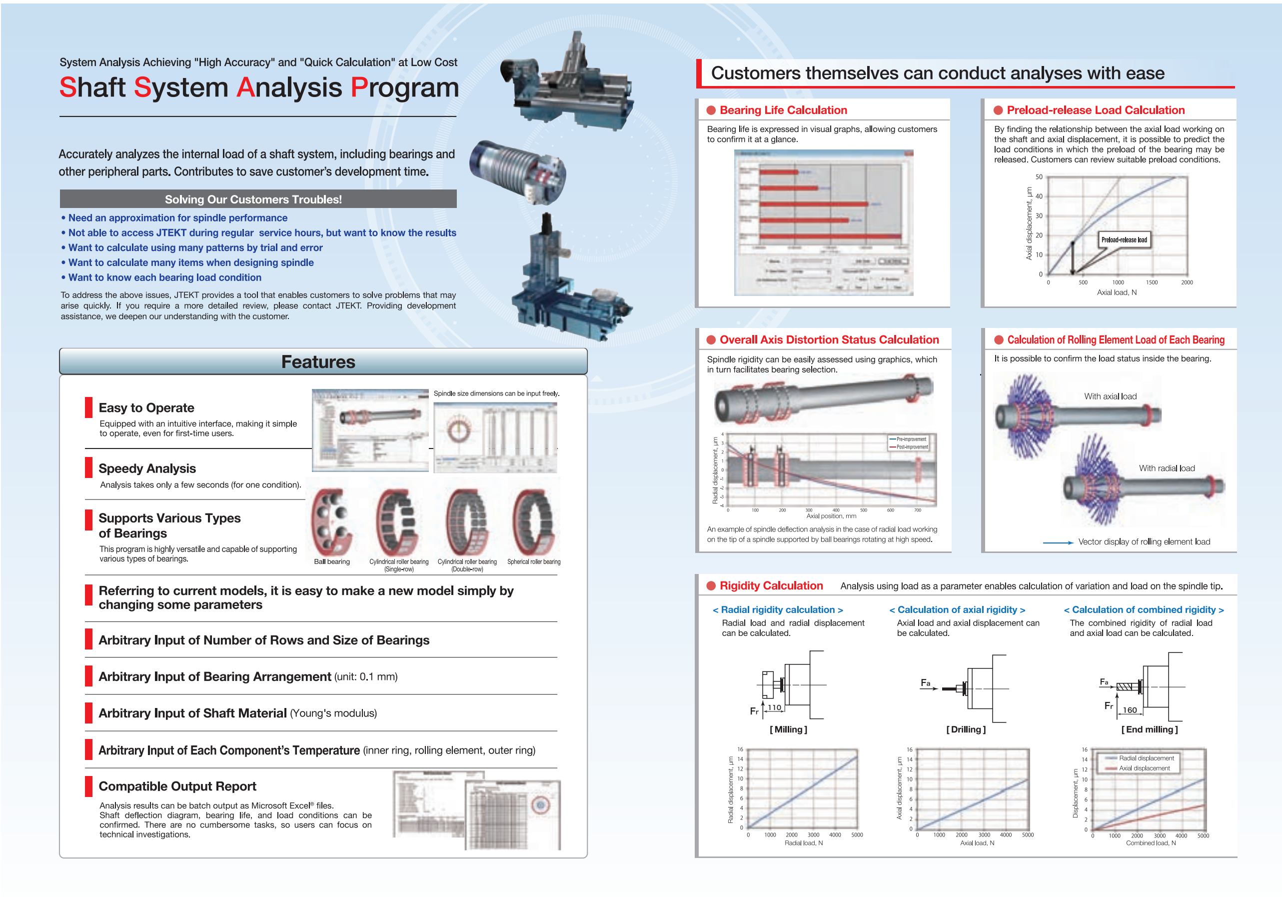

Fig. 15 Brochure of S.S.A.P. for customers

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