INTERNATIONAL JOURNAL OF PRECISION ENGINEERING AND MANUFACTURING Vol. 15, No. 2, pp. 201-207 FEBRUARY 2014 / 201
DOI: 10.1007/s12541-014-0326-9

Development of a Variable Preload Spindle by using an Electromagnetic Actuator

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KEYWORDS: Variable preload spindle, High efficiency spindle, Electromagnetic actuator, Machine tool, Rolling bearing

Variable preload technology involving varying preloads with respect to the spindle speeds and machining conditions is the most suitable preload method for spindles that require extensive rotation. The necessity of variable preload technology will continue to grow as the spindle speed and efficiency increase in order to improve the productivities of machine tools. This study attempts to develop a variable preload spindle that varies the preload with respect to the rotation number, allowing high speed rotation, and improves the stiffness characteristics. The authors designed and fabricated a variable preload device composed of an electromagnetic actuator and other components. We then evaluated the performance of the fabricated variable preload device with an experiment using load cells and subsequently determined the performance of the device and identified areas requiring improvement through a performance evaluation. Finally, we developed a variable preload spindle by applying the produced variable preload device. Smooth operation was verified through performance evaluations of the variable preload spindle, and stiffness improvement was verified through preload variation.

Manuscript received: July 23, 2013 / Accepted: December 23, 2013

NOMENCLATURE

F_p = preload
F_s = spring force
F_a = actuating force
F_f = friction force
B = magnetic flux density in tesla
μ_o = the permeability of free space or air
N = number of coil turns
I = electric current
g = air gap
A_g = pole surface area

1. Introduction

Loss of work from frequent movement of processes has to be minimized in order to improve machining precision and productivity in modern production systems. Errors from reinstallation should also be minimized by completing as many processes as possible in the installation of a workpiece and the non-cutting time from movement and installation of a workpiece has to be minimized in order to achieve better productivity of mass production in small quantity batch production. In order to fulfill such needs, machine tool manufacturers are releasing precision machines with high and ultrahigh speed spindles, and multi-tasking machines capable of multi-processes. High efficiency spindles with improved processing capability from enhanced stiffness and high speed characteristics, through the application of variable preload technology, have also been released.

Variable preload technology is used to change appropriate preload applied on spindle bearings based on the spindle speeds and machining conditions, and is known as the most suitable preload method for a spindle requiring extensive rotation.[1-6] Machine tool manufacturers, such as IBAG, MAZAK, GMN, YASDA, etc. are introducing high efficiencies of spindles through variable preload technology.[7-10] The necessity of variable preload technology will continue to grow as the spindle speed and efficiency increase.

Recent studies on variable preload structures applied to machine tool spindle bearings are as follows. Tsutsui et al. introduced variable preload structure using a piezoelectric actuator.[2,11,12] Kitamura et al. introduced a structure that can vary the preload applied on spindle bearings with a shape memory alloy.[13] Zverev et al. and machine tool manufacturers, such as IBAG, MAZAK, etc., have introduced variable preload structures using hydraulic or pneumatic pressure.[1,7,8,14,15] Hwang and Lee conducted research on variable preload structures using centrifugal and magnetic forces.[4,5]

In the summary of these studies, hydraulic actuators, with large driving range and force, are the most popular variable preload structures for applying variable preload. However, the use of hydraulic actuators results in high costs and there are difficulties in establishing the correlation between the accurate number of rotations and the preload. In addition, a feedback system is sometimes separately installed due to the inaccurate feedback process of lowering increased preload.[16] In variable preload structures using a hydraulic actuator, introduced by MAZAK and others, the variation of preload is usually limited to 2~3 levels.

Hence, Hwang et al. introduced a variable preload device that utilizes an electromagnetic actuator, which has a faster response and is more accurate than a hydraulic actuator, using electric current as the driving source. Previous studies focused on the introduction of novel structures and verification of the principle.[5] Taking a step further from such studies, this study researched application of the proposed variable preload device on a commercial spindle, and carried out the finite element analysis to verify the structural stiffness and the electromagnetic force of the actuator. An electromagnetic actuator capable of arbitrary control of preload depending on the number of rotations of the spindle and a controller was developed and applied to a built-in spindle. The performance of the variable preload device was verified through experiments and the variable preload spindle effects were verified through a stiffness experiment.

2. Development of a Variable Preload Spindle

2.1 Concept and operation principles of the variable preload device

An electromagnetic actuator was applied to the variable preload device of the new structure for control of the preload on the rolling bearings for machine tool spindles. Fig. 1 shows the concept of the variable preload device proposed in this paper for control of the preload.

The variable preload device is composed of an electromagnetic actuator, preload spring, spring cage, and linear motion bearings. The preload on a bearing is determined by Eq. (1).

$$F_p = F_s - F_a + F_f \fallingdotseq F_s - F_a \tag{1}$$

Preload (F_p) in Eq. (1) is a range of values determined by the life of the bearing, the maximum allowable bearing contact stress, and other conditions depending on the spindle rotation number, load, etc. and determined from a bearing analysis. Spring force (F_s) is a fixed value determined at the design phase. Friction force (F_f) can be negligible with smooth motion along the axis of the disc with the role of linear motion bearing. Therefore, the preload (F_p) on bearings can be controlled by varying the actuating force (F_a) of the electromagnetic actuator.

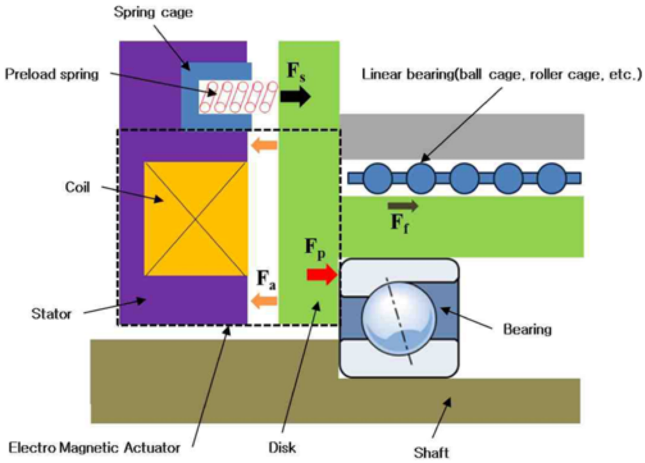

Fig. 1 Concept drawing of the variable preload device

Table 1 Specifications of the spindle

Item	Specification
Max. rotational speed	24,000 rpm
Shaft interface	HSK-A63
Driving method	Built-in motor
Lubricating method	Oil-air
Preload method	Constant pressure preload (1,400 N)
Bearing arrangement	DBB

The actuating force is determined by Eq. (2), using the Maxwell stress tensor method.[17,18]

$$F_a = \frac{1}{2\mu_o}\iint |B|^2 da = \frac{\mu_o N^2 I^2 A_g}{4g^2} \tag{2}$$

The area of pole surface (A_g) and number of coil turns (N) are fixed by the design values of the electromagnetic actuator. Therefore, the actuating force (F_a) can be controlled by varying the amount of supplied electric current (I). The desired actuating force is, therefore, obtained by controlling the electric current supplied to the coil.

2.2 Specifications of the variable preload spindle

The proposed variable preload device was applied to a high speed spindle of H company. The specifications of the applied spindle are listed in Table 1.

The maximum rotational speed is 24,000 rpm and the type of spindle interface is HSK-A63. The preload type is a constant pressure preload, which applies 1,400 N of preload by spring force.

This study only replaced the existing spindle preload method with the variable preload method in order to verify the effect of the proposed variable preload device. The recommended preload was determined by comprehensively considering the life and stiffness of the bearing, contact stresses within the bearing, and sliding friction of the rolling element, and the values recommended by the manufacturer were used. The recommended preloads for different rotational speeds are listed in Table 2. The preload was varied within the spindle rotational speed range using the proposed variable preload device.

Table 2 Recommended preloads according to rotational speeds

Rotational speed [rpm]	Preload [N]
0 ~ 4,000	2,100 ~ 2,300
4,000 ~ 8,000	2,000 ~ 2,200
8,000 ~ 12,000	1,900 ~ 2,100
12,000 ~ 16,000	1,700 ~ 1,900
16,000 ~ 20,000	1,500 ~ 1,700
20,000 ~ 24,000	1,300 ~ 1,500

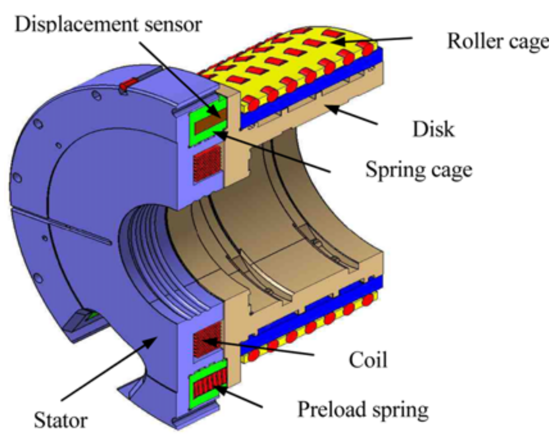

Fig. 2 The assembly structure of the AMA

2.3 Manufacture and testing of the variable preload device

A prototype of the variable preload device was produced and an experiment device was designed in order to verify the operating principle and evaluate the performance of the proposed variable preload device prior to its actual application to the spindle.

Fig. 2 illustrates the final form of the variable preload device this paper proposes.

The roles of each part are as follows. The electromagnetic actuator controls the preload on the bearings by creating electromagnetic force proportional to the electric current from an external source between a stator and a disc. A disc acts as a stator of the electromagnetic actuator and the housing of the bearing simultaneously, and creates a cooling channel around the outer surface to cool the bearing. The outer surface of the disc is guided by the roller cage to facilitate the motion along the axis of the disc and grant the radial stiffness. The material of the disc is a SM45C. A stator is composed of a spring cage, which fixes the preload spring, and a coil. A sealing and cooling function are required in order to achieve a cooling effect on the stator and to prevent the bearing lubricant from seeping into the motor. Compressed air was supplied to a grooved labyrinth seal to perform cooling and sealing functions simultaneously. The spring cage is composed of a processed radial hole, a preload spring, and a displacement sensor. In order to minimize flux leakage of the electromagnetic actuator, the material is non-magnetic substance. Preload spring applies load proportional to the compressed length, which depends on the relative distance of the disc and the stator, to the bearing. The preload spring plays the same role as the preload spring in the constant pressure preload method. The displacement sensor measures the relative distance between the disc and the stator to sense the change in the air gap.

The design theory of Maslen and Knospe was used in order to determine the dimensions of the stator and the disc of the electromagnetic actuator.[18] The specifications of the developed variable preload device are listed in Table 3.

Table 3 Specifications of the developed variable preload device

Parameter		Specification
Coil spring	Free length	25 mm
	Compression length	6.9 mm
	Spring force	2,430 N
Electro-magnetic actuator	Max. actuating force	2,500 N
	Copper diameter	1 mm
	Coil turns	175
	Packing factor	0.715
Disp. sensor	Measurement range	±0.25 mm
	Accuracy	0.5 μm
	Repeatability	0.1 μm

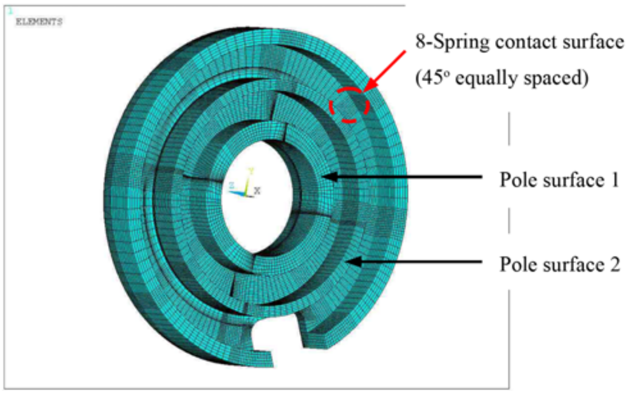

Fig. 3 Finite element model of the stator

A finite element analysis was conducted using the commercial software ANSYS to verify the validity of the designs of the variable preload device. The validity of the design was conducted in two categories: the verification of the structural stiffness of the stator and disc, and the verification of the electromagnetic force of the electromagnetic actuator.

The validity of the structural design of the stator and disc was reviewed from the perspective of stiffness design. The influence of the strains of the stator and disc on the change of the air gap under the load condition and the allowable range of the change in the air gap due to the strain for operation were determined.

Fig. 3 shows the finite element model of the stator. For the convenience of the analysis, holes and the cooling channel that do not have a significant effect on the result of the analysis were neglected in the modeling.

The loading conditions were set to be 2,430 N of spring force and pole triggered a maximum of 1,030 N of electromagnetic force. The axial degree of freedom of the contact area of the stator due to the assembly and the radial degree of freedom of the outer surface of the stator was constrained as the constrain condition.

From the analysis, axial strain of 0.8 μm of the stator was identified at the 1 pole that affects the air gap.

Similar to the stator, the disc was modeled without holes and cooling channels that are not significant to the results. Pressures of 2,430 N and 1,030 N were loaded on the spring assembled side and electromagnetic force response side, respectively, as the loading condition. The axial

Table 4 Calculation results of the electromagnetic force

Displacement [μm]	0	1.03
Magnetic force [N]	2,752	2,765

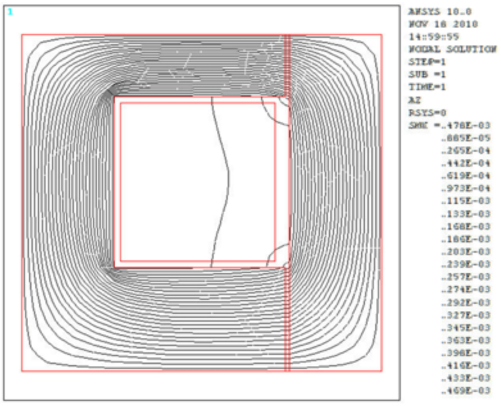

Fig. 4 Electromagnetic flux lines of the electromagnetic actuator

degree of freedom of the area of contact with the bearing outer ring and the radial degree of freedom of the outer surface assembled with the linear motion bearing was constrained as the constrain condition. Maximum axial strain of 3 μm of the disc occurred at the spring support side and axial strain of 0.23 μm of the disc occurred at the side affected by the electromagnetic force that affects the air gap.

From the results above, a 1.03 μm change in the air gap was observed from the changes in the stator and disc. Change in the electromagnetic force was computed to identify the effects of change of the air gap on the electromagnetic force. Cases in which the strains are 0 and 1.03 μm were calculated using Eq. (2).

Table 4 illustrates the results. The change in the electromagnetic force from the computation is 13 N. According to the recommended preload range depending on the number of rotations shown in Table 2, the allowable preload range is ±100 N when the median is set as the target preload. The change in the electromagnetic force due to the strain is within the allowable preload range. Therefore, the designed electromagnetic actuator will not have problems due to strain.

The electromagnetic field analysis of the actuator was carried out with the finite element method for design verification of the electromagnetic forces of the electromagnetic actuator. 2-D axisymmetric modeling was used for the analysis of the electromagnetic field of the electromagnetic actuator and the dimensions and input conditions are identical to the design conditions.

Fig. 4 illustrates the flow of the magnetic field lines of the electromagnetic actuator. The analysis revealed smooth flow of magnetic field lines and no magnetic saturation at the air gap. In addition, an analysis by the virtual work method revealed 2,564 N of electromagnetic force, which is in agreement with the design load capacity of 2,500 N. Therefore, the designed electromagnetic actuator fulfills the design specifications.

Fig. 5 illustrates the prototype of the variable preload device.

Fig. 6 illustrates the configuration of the experiment device. A load cell was installed at the end of the disc to directly measure the change in the actuating force with respect to control of the electric current. Dynamic and static forces are both important in actuating force.

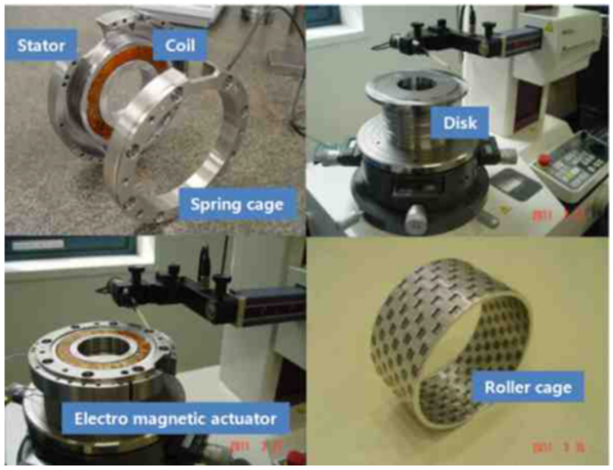

Fig. 5 Photograph of the manufactured components of the variable preload device

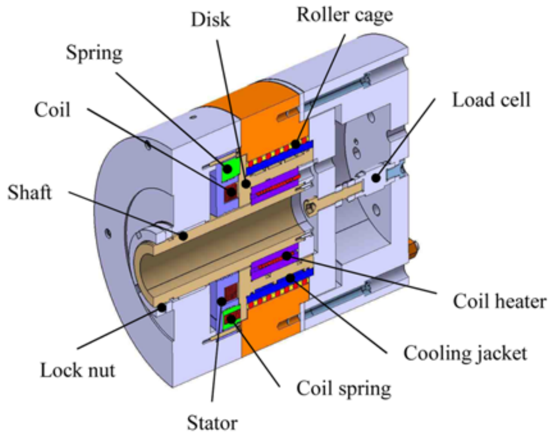

Fig. 6 Assembly drawing of the designed experiment device

Piezoelectric force sensor is suitable for measuring periodic dynamic forces but is not suitable for long term static forces due to their thermal drift characteristic. On the other hand, a strain gauge force sensor is suitable for measuring long term static forces. Therefore, a piezoelectric sensor and a strain gauge sensor were configured to be used alternatively.

In order to evaluate the performance of the variable preload device and determine the correlation between electric current and actuating force, reproducibility and hysteresis characteristics tests were conducted using the experiment device.

In the reproducibility tests, the electric current was increased by 0.5A from 0 to 3A to measure the load on the load cell. Fig. 7 illustrates the test results between control over the electric current and the actuating force.

The reproducibility was confirmed within 5N, and change in the air gap due to the actuating force within the experimental conditions was not observed.

Fig. 8 illustrates the hysteresis characteristic test results. The electric current was increased by 0.1A from 0 to 3A and then decreased back to 0A. A hysteresis loop with increase and decrease of the electric current was observed. A difference of approximately 100N in actuating force occurred, with an increase and decrease of electric current of 2A of electric current was applied. In an actual spindle system, the system

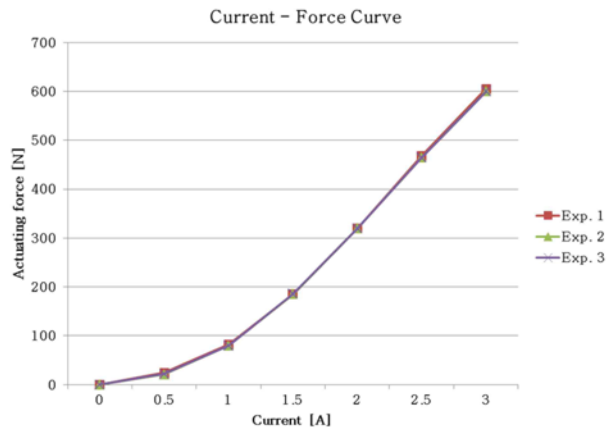

Fig. 7 Reproducibility characteristic test results

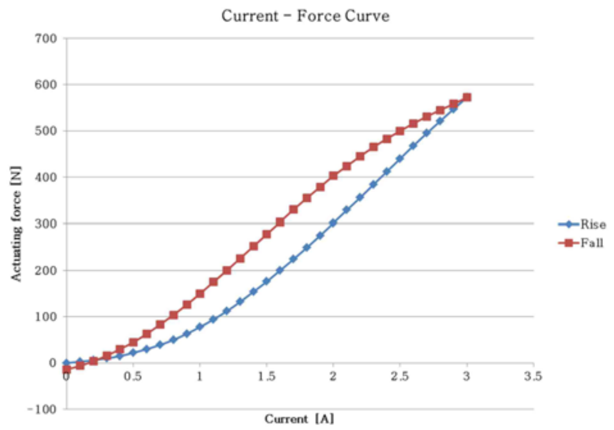

Fig. 8 Hysteresis characteristic test results

could experience the identical situation with increase and decrease of the rotational speed and an unappropriate preload could be applied to the bearing by the difference of the actuating force. Therefore, a measure has to be taken for the hysteresis loop.

At the design phase, the electromagnetic actuator could be designed with a material with high magnetic flux density to decrease the hysteresis characteristic. However, such a measure will not eliminate the hysteresis loop totally and will increase the production cost. At the operating phase, the electric current can be directed to the desired electric current from 0A to avoid a hysteresis loop. Thus, when 3A is the desired electric current, the electric current will be directed to 0A before the electric current is directed to 3A regardless of the previous electric current. In such a case, it will not be affected by a hysteresis loop but an unappropriate preload could be applied to the bearing when the electric current changes to 0A. A problem could, in particular, arise in bearings when the electromagnetic actuator has low responsiveness and the difference in the actuating force is large (when large electric current has to be applied). However, considering the responsiveness of the electromagnetic actuator and the applicable preload range, the method could be applied.

With the results above, the reproducibility of the actuating force of the proposed device is secured but a hysteresis loop occurred. With secure reproducibility of the actuating force, the hysteresis characteristic was determined to be avoidable with control methods. Therefore, the developed variable preload device will be applicable to an actual spindle system.

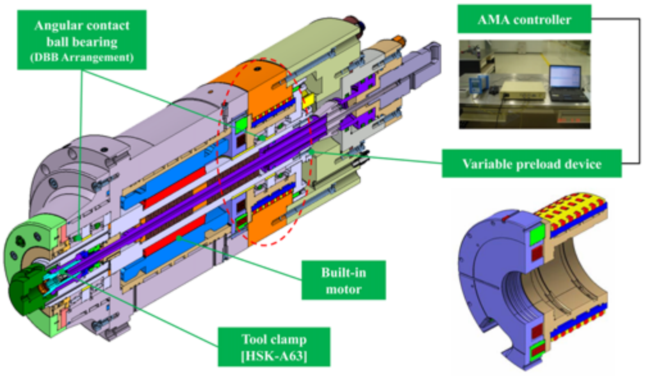

Fig. 9 Construction of the variable preload spindle

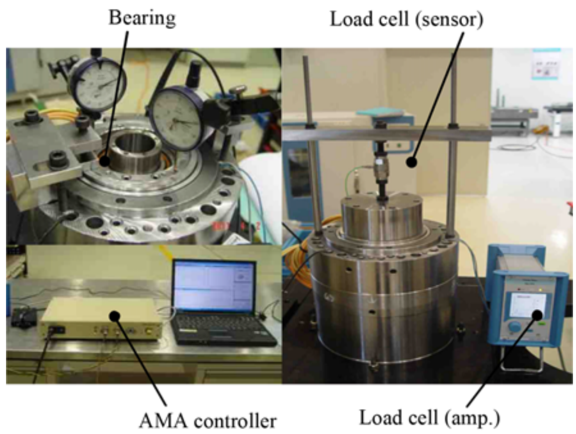

Fig. 10 Experimental set-up for a performance tests

2.4 Testing of the variable preload spindle

The developed variable preload device was applied to an actual spindle system. Fig. 9 illustrates the overall configuration of the variable preload spindle with the variable preload device.

The basic performance of the developed spindle was evaluated. The performance was evaluated in two major categories: an actuating force test, where the variable preload device was assembled with the spindle system, and a rigidity test to verify the effect of the variable preload spindle compared to the conventional constant pressure preload spindle.

Fig. 10 is a picture of the actuating force test setup.

The actuating force was measured, after verification of the air gap using a dial gauge and a fixture, as the electric current was varied by 0.5A from 0 to 4A.

Fig. 11 illustrates the experiment results. The experiment results were in agreement with the predicted value from the Maxwell stress tensor method with error of 15%. Considering the design margin of the developed variable preload device, application of the device to a spindle system is still plausible. In addition, repeatability, reproducibility, and other basic performances were acceptable.

Fig. 12 illustrates the experiment for the measurement of the rigidity. A hydraulic actuator, a load cell, and a displacement sensor were used to measure bending stiffness.

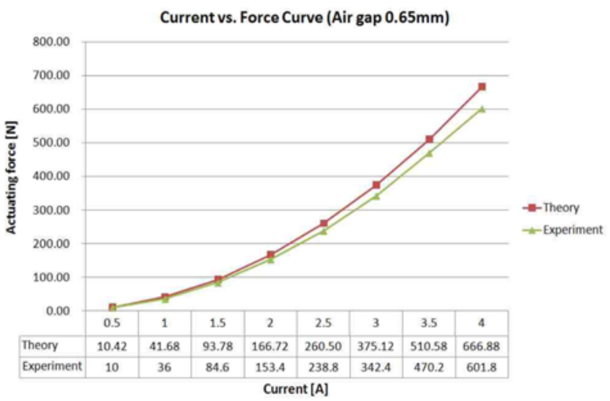

Fig. 11 Actuating force according to the electric current

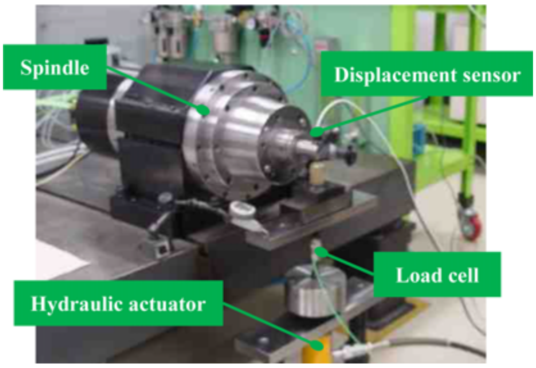

Fig. 12 Experimental set-up for bending stiffness test of the spindle

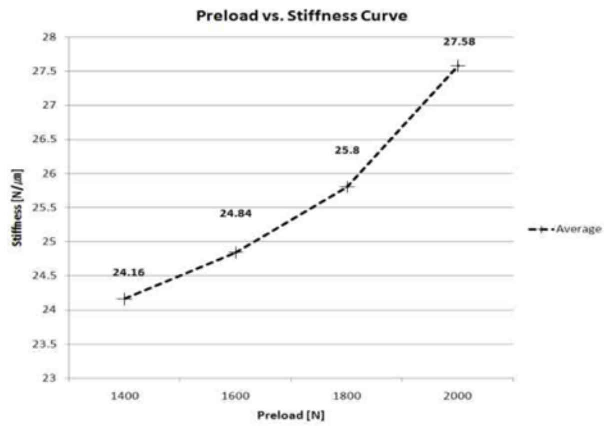

Fig. 13 Bending stiffness related to the preload applied to bearing

Fig. 13 illustrates the experiment results of the bending stiffness. From the experiment, a maximum 14% improvement in stiffness was observed with the varying preload. This result verified that the developed variable preload device varies preload and is capable of improving the stiffness of a spindle system.

3. Conclusions

This study has developed a variable preload device, with a novel structure, capable of varying the preload on the rolling bearing of a spindle. From the subsequent tests, we arrived at the following conclusions.

A new variable preload device composed of an electromagnetic actuator and a preload spring was proposed.

A prototype of the proposed preload device, with the proposed structure, was produced and the performance of the device was evaluated with an experiment device with a load cell. This paper also verified the actuating force and reproducibility of the proposed device with experiments and attempted to resolve the problem of a hysteresis loop by increasing and decreasing electric current with control.

The developed variable preload device was applied to an actual spindle system and a bending stiffness test was conducted. Smooth operation of the device and maximum improvement of 14% in stiffness were observed.

The proposed structure is expected to be more affordable to operate and more eco-friendly than the conventional hydraulic variable preload device. Quick and accurate control of preload will also be possible with the use of an electromagnetic actuator. However, miniaturization of the electromagnetic actuator, to install a preload device in a compact spindle, and heating problems are tasks that should be addressed in future studies. Environmental performance and durability tests of the variable preload device for commercialization will also be evaluated in the future.

ACKNOWLEDGEMENT

The authors would like to thank HYUNDAI WIA CORP. for supporting this work.

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