High Stiffness Air Bearing Design and Experiments for CMP Machines

Philip Greene, Zongren Shang, Douglas Bittner
Dover Instrument Corporation
Flanders Road, Westboro, MA 01581, USA

INTRODUCTION

The Chemical Mechanical Polishing (CMP) process has been developed for planarizing the dielectric layers that insulate the levels of metal interconnect in integrated circuit (IC) chips. Most CMP machines are currently using ball bearings in the main spindles and table spindles. However, along with the enhancement of the required surface performances and the prediction of increased levels of interconnect [1], the stiffness, speed, and error motion of spindles have become the key properties for achieving advanced CMP process requirements. Air bearing spindles are ideal alternative products for CMP machines. Air bearing spindles possess much lower error motion than ball bearings. In applying air bearing spindles into CMP machines, one of the most important properties of air bearing spindles is stiffness. This paper will present the results of the studies.

Because the critical performance parameters of CMP spindles are the most important factors for satisfying advanced CMP requirements for speed, error motion, and stiffness, the bearing selection and design should be carefully considered. Air bearings possess many advantages over ball bearings and oil-lubricated bearings. For instance, oil bearings produce friction heat at higher speeds and the oil degrades at higher temperatures. These are fatal problems of oil-lubricated bearings in the CMP process. Although ball bearings have some advantages over air-lubricated bearings, the error motion of ball bearings is much larger than that of air bearings. In addition, the PV value of ball bearings limits their application at high speeds, such as in the CMP process. Air bearings possess low error motion and high-speed characteristics. The major existing problem of air bearings is lower stiffness relative to oil bearings.

In order to achieve high stiffness of air bearing spindles for CMP applications, studies have been conducted on air bearing spindles. The pressure distribution of the air film became the target to reduce the side-flow effect for increasing the stiffness when using orifice-type restriction. In addition to the theoretical studies, experimental studies have also been conducted on the air bearing spindle. In the tests, the stiffness of the air bearing reached 8.7×10^8 N/m (4.97 lbs./μin) in the axial direction, and 1.9×10^9 N/m (11 lbs./μin) in the radial direction. All of the testing data of stiffnesses reached the target values from theoretical simulation within 10%. The spindle was also velocity tested to 6,500 rpm (390 meter/sec surface speed at the rotor periphery).

CONSIDERATIONS OF AIR BEARING DESIGN

Because of the high stiffness requirements, the bearing design should be carefully considered. In order to satisfy the specifications, the following issues need to be addressed:

1. Pressure ratio
2. Line feed coefficient
3. Pneumatic hammer
4. Airflow rate

In an air bearing design, the pressure ratio (outlet pressure of orifice/inlet pressure) is an important parameter because of its effect on loading capacity and stiffness. Due to the machining error, the pressure ratio was given between 0.55 to 0.75. This variation will give ~20% change in stiffness, which satisfies the specification requirements. However, once the pressure ratio was determined, the side flow of air caused a decrease in stiffness when using orifices as the restrictors. According to *Precision Machine Design* [2], a difference in the line feed correction coefficient ($1/\lambda$), which is a measure of side flow, may result in a significant change in stiffness. In order to increase the line feed coefficient, narrow and shallow grooves were used on the thrust bearings, while long and shallow pockets were used on the journal. (Fig.1).

The most common issue, which remains a serious concern, is pneumatic hammer. In order to reduce the restriction of airflow through the grooves (which increase the line feed coefficient), the depth and width of the grooves was relatively large. Consequently, these large grooves may cause pneumatic hammer. In an effort to counteract this occurrence, the grooves were reduced to the minimum size. This resulted in a reduction in the line feed coefficient, which in turn reduced the risk of pneumatic hammer. The width and depth of the grooves was 0.0125mm (0.0005”).

The final concern was the airflow rate. There is no doubt that the air gap is the key parameter involved in the airflow rate. To satisfy the specification, the air gap could not be larger than 0.0127mm (0.0005”). In order to satisfy the stiffness requirements, the air gap had to be smaller than 0.0125mm, which also satisfied the air flow requirements.

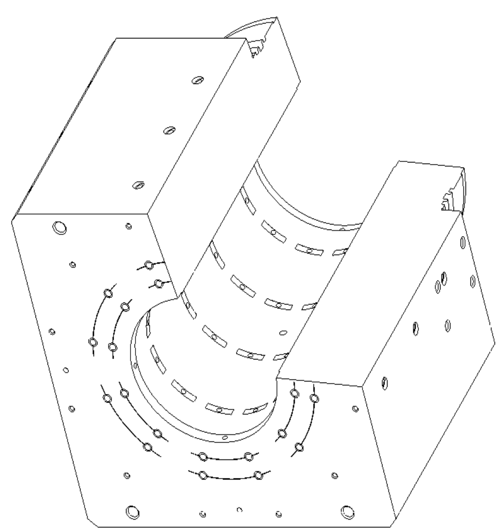

Fig. 1 Air Bearing Structure

EXPERIMENTAL RESULTS

The load versus displacement curves in both the radial and axial directions are illustrated in Figures 2 and 3. The displacements shown are of the bearing film only. The data was obtained by first measuring the combined displacements of the bearing film and the deformations of the bearing structure. The deformations of the structure alone were then measured. The differences between the two are the resultant displacements of the bearing film only.

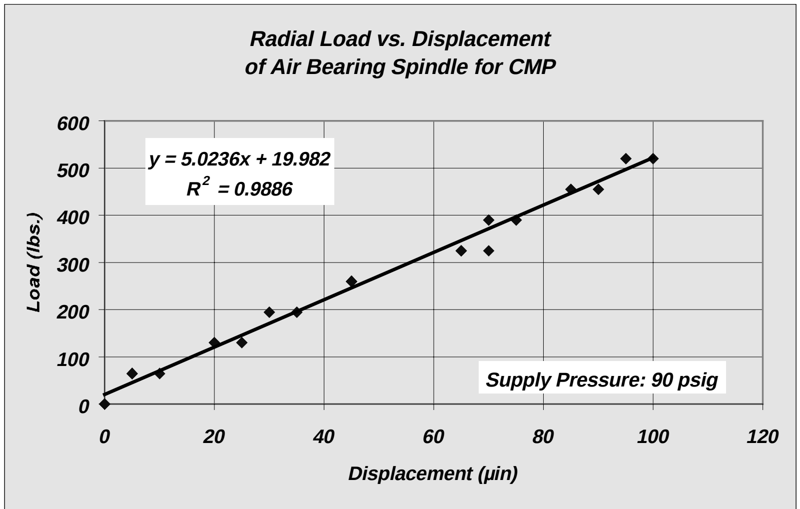

Fig. 2 Radial Load vs. Displacement

Capacitance probes measured the displacements and deformations, while an air cylinder applied the loads. The bearing supply pressure was operated at 90 psi.

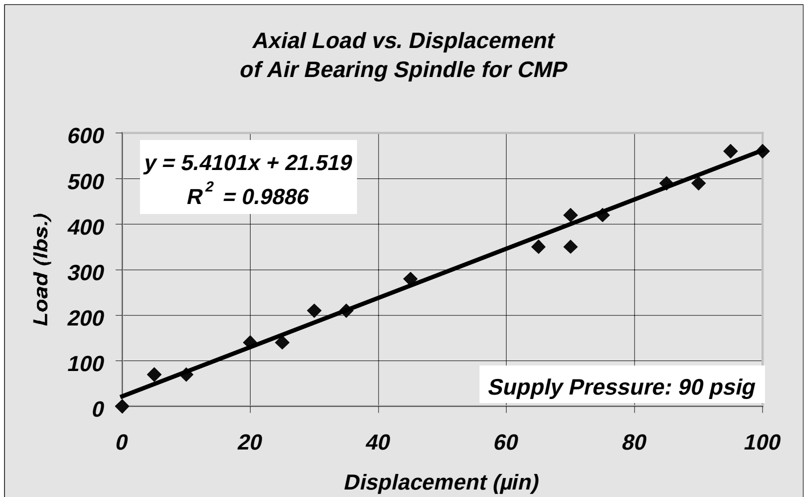

Fig. 3 Axial Load vs. Displacement

Below is a comparison of the theoretical and actual bearing stiffness and capacities:

	Calculated Stiffness	**Calculated Load Capacity**	**Tested Stiffness**	**Tested Load Capacity**
	Newton/meter	***Newton***	***Newton/meter***	***Newton***
Axial	8.89×10^8	2558	8.84×10^8	> 2180
Radial	1.03×10^9	2891	8.89×10^8	> 2180

CONCLUSIONS

1. The application of air bearings into the CMP process is feasible.
2. The stiffness requirements of most CMP machines (radial direction: 2.8×10^8 Newton/meter and axial direction: 6.13×10^8 Newton/meter) can be met by this design.
3. Both the pressure ratio and the line feed coefficient were reasonably designed. The pressure ratio was 0.57 and the line feed coefficient was 0.65.
4. The balance of the line feed coefficient and the pneumatic hammer should be considered when determining the depth and width of the grooves.
5. The error between theoretical calculation and experimental data is within 10%.

REFERENCES:

[1] *National Technology Roadmap for Semiconductors: Semiconductor Industry Association Report*, SEMATEC, Inc., Austin, TX, 1997.

[2] Slocum, Alexander.H., *Precision Machine Design*, Massachusetts Institute of Technology, Society of Manufacturing Engineers, Dearborn, MI, 1992.