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AIR-BEARING SLIDES FOR PRECISION
X-Y TURNING MACHINES

W. H. Rasnick

UNION CARBIDE CORPORATION
NUCLEAR DIVISION
OAK RIDGE Y-12 PLANT

operated for the **ATOMIC ENERGY COMMISSION** *under* **U. S. GOVERNMENT** **Contract W-7405 eng 26**

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ABSTRACT

Air-bearing slides have been installed on precision X-Y turning machines. Performance tests indicate that the slide bearings met design and experience predictions. An evaluation based on the results of tool-path accuracy tests, machined test parts, and abnormal operating experience show improved performance and maintainability.

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SUMMARY

Air bearings were designed, fabricated, and installed on the tool slide of a precision turning machine and the tool and work slides of a similar machine at the Oak Ridge Y-12 Plant. (a) Optical-flat checks showed an improved bearing performance, especially in regard to short-wavelength errors, when compared with roller bearing slides. Stiffness and load-capacity values were below that of the roller bearings; however, performance of the slides was completely satisfactory in machining parts. During normal operations, the initial tool slide survived a high impact loading ("wreck") without measurable damage. Conclusions reached are that air-bearing slides are feasible for precision machines where load and tool-force requirements are compatible.

(a) Operated by the Union Carbide Corporation's Nuclear Division for the U S Atomic Energy Commission.

INTRODUCTION

Precision X-Y turning machines require slideways that provide straight, accurate, frictionless carriage movements. These machines now use roller bearings for the slides. The desired straightness of travel is difficult to obtain, particularly with respect to the short-wavelength variations in this straightness. Air-bearing slides should improve this condition significantly.

In addition to the advantages of negligible friction, zero wear, and a straighter travel, the air bearing can withstand a greater shock loading than a roller bearing without damaging the ways. Three factors provide this advantage: (1) a larger bearing area, (2) the presence of a fluid squeeze film, and (3) the low compressive strength of the graphite pad compared to the cast iron ways.

This project was undertaken to determine the feasibility of using air-bearing slides with respect to both maintenance and performance.

AIR-BEARING SLIDES

DESIGN

Concept

For the first application of air-bearing slides, the tool slide of a modified 18-inch by 26-inch turning and boring machine was selected. This machine has a basic way configuration of a flat way and a female vee. Also, this type of machine meets the criterion for noncaptive or floating air-bearing slides in that the load on the bearings due to the tool slide's weight is an order of magnitude greater than the tool forces. For example, the tool slide weighs approximately 1,500 pounds while finishing-cut tool forces are 5 - 10 pounds. Graphite bearing pads with steel backing epoxied to the slide were chosen for the initial feasibility study (Figure 1). Although an adjustable self-aligning bearing pad would have more advantages from an installation, maintenance, and performance standpoint, it was decided not to incorporate this feature into the design as it would have entailed machining the slide; which, in turn, would have prevented further use of roller bearings if the air bearings proved to be inadequate.

Figure 1. SLIDE MODIFICATION TO A TURNING AND BORING MACHINE.

After the initial studies were completed, a self-aligning adjustable feature was included in the conversion of both tool and work slides of a similar machine (Figure 2). The final design configuration, as seen in Figure 3, consisted of two flat pads on adjustable spherical seats and two vee pads on fixed spherical seats. Graphite, which has been used extensively at Y-12 for gas-bearing applications, was chosen as a bearing pad material for several reasons: (1) it provides protection against way damage because of its lower compressive strength than the way material, (2) it does not gall on occasional rubbing contact as does some other gas-bearing materials, (3) it can easily be lapped to custom fit the ways against which it will run, and (4) it can be impregnated^(b) to provide maximum resistance to self-excited vibrations to which heavily loaded, noncaptive gas bearings are sometimes extremely

(b) A surface treatment for modifying the flow through a graphite gas bearing is described by Rasnick and Shell in Document Y-SC-7, *Impregnation of Porous Graphite for Flow Modification*; June 7, 1971.

susceptible. In fact, having the ability to cope with these vibrations was a necessary condition before such an application could be attempted.

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Figure 2. MODIFIED TURNING AND BORING MACHINE WITH AIR-BEARING SLIDE WAY PADS SUPPORTED ON SPHERICAL SEATS.

Figure 3. CROSS SECTION OF A MACHINE SLIDE.

Procedures

Design procedures were fairly straightforward. The two pads on each way were located so as to provide an even distribution of the load between them and as far apart as possible to provide a stiffer system in regard to rotation of the slides from an external moment. Pad area was calculated so that an average film pressure of 40 psi would be seen by the pads. Design calculations for the tool-slide pads are given in Appendix A.

In designing the pads and spherical seat supports for the work slide, care had to be taken so that excessive bending of the pad was avoided. A nominal design limit of 50 microinches deflection from the center to the end of the pad was set. Design procedures for the work-slide V pads are given in Appendix B.

FABRICATION AND INSTALLATION

Procedures

Fabrication of the pads and supports, seen in Figure 4, were for the most part routine, although the male and female spherical seats were lapped together to minimize pad distortion. On the initial installation, the only machining of the tool slide necessary was relocating the bolt holes for the lead screw nut. Considerable machining, however, was required on the slides for installing the spherical seat supports. Lapping of the flat pads was done previous to the installation, while the V pads were lapped to the V ways during the installation work. After lapping, the pads were impregnated to run on about a 200-microinch film.

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Figure 4. AIR-BEARING PADS FOR THE MACHINE WORK SLIDE SHOWING SEATS AND SOCKETS.

To load the slides with the spherical seat supports, the pads were placed on the ways, vacuum was applied to them, and the slides were gradually lowered down onto them with a hydraulic load balancer. Due to some distortion it was necessary to lap the work-slide pads with the full load of approximately 5,000 pounds. This work was accomplished by reducing the supply air pressure until the slide was difficult to move. Lapping compound in perchloroethylene was used. After this lapping was completed, the slide was removed, the pads cleaned, and the slide replaced. Very little of this type of lapping was required on the tool-slide pads, as might be expected by the lighter loads. After final lapping and cleaning, connections were installed for the air supply to the pads, and pressure switches were installed on the bearing side of the pressure regulators in order to shut the machine off in the event the air pressure dropped below that necessary to float the bearings. Figure 5 shows the bearing supply manifold with regulators and pressure switches; two of the installed tool-slide bearings can be seen in Figure 6.

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Figure 5. BEARING SUPPLY MANIFOLD.

Precautions

In any gas-bearing application, three precautions are necessary: (1) the supply air must be clean and reasonably dry, (2) the bearings must be protected from oil, grease, dust, and chips, and (3) the metal surfaces against which the bearings run must not rust or corrode. The first of these conditions was met by the use of 10 and 0.1 micron filters connected in series and attached to the plant instrument air supply having a moisture content of 18 - 30 ppm. The second condition was met by installing rubber shields on the tool-slide covers. The work-slide covers seemed adequate and were not altered. Also, the joints where the slide covers fastened to the slide were sealed with a gasket and silicone rubber. Three approaches are available for coping with the third condition. The first approach is to use a noncorrodible or nonrusting way material, but this method is not possible since this was a retrofit-type installation. Secondly, a noncorrosive plating approximately 20 - 30 microinches thick could be applied to the ways. In fact, a technique has been developed to do this. The third method is the one which was used as a matter of expediency and it remains to be seen whether or not it will suffice over long periods. This method consists of using an extremely light coat of rust repellent that is just heavy enough to prevent rust but thin enough to prevent the bearings from becoming clogged. Experience over almost a year of operation on the initial installation has indicated that it may be adequate if the air conditioning system maintains an approximate 50 percent relative humidity.

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Figure 6. INSTALLED TOOL-SLIDE BEARINGS.

Cost

Total cost of converting the turning machine to air-bearing slides on both axes was approximately \$19,000. This estimate excludes the cost of materials and development labor. This cost was arrived at by adding 656 manhours for fabricating the bearing components and machining the slides to accept the pad supports, 8 manhours for assembling the components, and 348 manhours for installation and testing. It might be noted that the cost per machine would probably drop if a number of machines were to be converted. Estimates are that the cost per machine could be reduced to \$14,700 if 10 machines were converted.

In building a new machine, money could probably be saved by the fact that half of the precision metal surfaces would be eliminated.

TESTING AND EVALUATION

Stiffness and Load Capacity

Stiffness and load-capacity tests were run on the air-bearing slides of both machines. On the machine with rigidly fastened pads made with tooling arranged for machining 30-inch-diameter parts, compliance tests indicated that the bearings contributed approximately 7 microinches of vertical tool deflection per pound of vertical tool force at a moment arm of 30 inches from the center of the V and flat bearings. In the horizontal

direction (perpendicular to the tool-slide motion), a tool deflection due to the bearings was measured at 3 microinches per pound of horizontal tool force in the same direction at a height of 20 inches from the bearings (Figure 7). With the spherical seat design, bearing stiffness was increased somewhat, though not drastically. The vertical tool deflection due to the film was 4 microinches per pound, while the horizontal deflection was 3 microinches per pound. Figure 7 indicates how the forces were applied.

Figure 7. COMPLIANCE TESTS ON THE MACHINE TOOL SLIDE.

It might be noted that although the film stiffness was increased, the structural stiffness was somewhat decreased by the spherical-seat design. In the case of the vertical tool deflection, the total deflections turned out to be essentially the same for each design. However, from the standpoint of the ease of fabrication, installation, and stability, the spherical-seat design is advantageous. As expected, the work-slide bearings were stiffer both in the structure and gas film than the tool-slide bearings, due, mainly, to the higher mass loading. Horizontal deflection measured on the slide from a horizontal force applied over the front pads was 1.67 microinches per pound due to the bearing. Total deflection was 2.33 microinches per pound.

Load capacities of each slide were tested by adding weights where actual forces would likely be applied until the slide motion began to drag. Vertical-tool-force load capacities were 80 and 125 pounds for the rigid and spherical seat designs, respectively. Work-slide part and fixture weight limits were found to be approximately 600 pounds.

One objective of any development effort of this nature is to determine whether or not the equations used for design calculations are sufficient models of the actual case. To determine this, the measured bearing film stiffness was compared to the calculated stiffness. In this case, the agreement was fairly good. A 100-pound load on top of the tool slide near the center of gravity of the tool slide caused a film deflection of approximately 13 microinches, compared to the calculated value of 15 microinches.

Accuracy

Tests were run to determine the straightness of the slides over both long and short wavelengths. Collimation checks on all slides showed yaw and pitch errors within one second over the entire travel of the slides. Optical-flat tests indicate that the air-bearing slides have significantly improved the smoothness of travel of both slides.

Figure 8 presents strip-chart records of the optical flat checks on the machine prior to disassembly to install the air bearings; Figure 9 gives the strip-chart records for the same checks after air-bearing installation. The only deviation from a smooth curve is a cyclic deviation caused by the lead screw. In order to minimize this error, "flex bolts" (special bolts with a reduced cross-sectional area) were used to couple the lead screw nut to the slide. This installation is indicated by the drawing of Figure 10. Figure 11 shows glass bar checks on the tool slide before and after adding the "flex bolts", while Figure 12 shows a trace on an optical flat along a 45-degree angle before and after installing the air bearings. Air bearings appear to have reduced significantly the lost motion or backlash on this particular machine. Ball sweep checks, which are used to determine the contour path accuracy of the machine, are seen in Figure 13. These tests emphasize the improvement that was gained by the gas bearings and "flex bolts".

Performance

Performance of the air-bearing slides was evaluated in terms of stability and part dimensions. To test stability, a (40-mil-deep) dry facing cut was taken on a 30-inch-diameter aluminum faceplate rotating at 125 rpm. No chatter or instability was observed. Performance of both machines equipped with air-bearing slides in machining parts has been entirely satisfactory.

In terms of maintenance, the air bearings have performed well. During the time the machine was operated with the original air-bearing slide, an accident occurred in which the work slide rammed the tool slide, causing damage in portions of the work slide ways in the form of a ripple. Fortunately, the ways were damaged only at one end of the travel distance and an extensive scraping job was unnecessary. Examination of the tool slide and ways showed no visible damage to either. Subsequent collimation checks confirmed this. Of considerable interest is the fact that due to the manner of the "wreck", the tool-slide bearings would have sustained a much higher impact load than the work-slide bearings. It can be safely speculated that damage to the tool slide and ways could have been extensive were it not for the air bearings.

CONCLUSIONS AND RECOMMENDATIONS

It can be stated that air bearings are feasible for precision X-Y turning machines and provide certain advantages as well as disadvantages over roller-bearing slides.

Advantages

1. Less friction (possibly reducing the servo error).
2. No wear on accurate ways.

(a) X Slide Horizontal

(b) Y Slide Horizontal

Figure 8. STRAIGHTNESS OF SLIDE TRAVEL WITH ROLLER BEARINGS. (Machine 310; Slide - 5"/mm, Chart - 1 mm/sec)

Figure 9. STRAIGHTNESS OF SLIDE TRAVEL WITH AIR BEARINGS. (Machine 310)

Figure 10. "FLEX-BOLT" MOUNTING ARRANGEMENT.

(a) Before "Flex-Bolts" Were Installed

(b) After "Flex-Bolts" Were Installed

Figure 11. LATERAL TOOL SLIDE CYCLIC ERROR.

(a) Before Air-Bearing Installation

(b) After Air-Bearing Installation

Figure 12. FORTY-FIVE-DEGREE TRACE ON AN OPTICAL FLAT.

(a) Before Installing the Air Bearings

(b) After Installing the Air Bearings

Figure 13. OUTER-CONTOUR BALL SWEEP.

3. Elimination of any short-wavelength error due to rollers.
4. Less susceptibility to damage due to impact or shock loading.

Disadvantages

1. Less stiffness. (Although apparently adequate for normal precision machining operations, errors can be caused by operators leaning on the slides, hoses, or cables. Also, the slide can be more easily influenced by eccentricity errors in lead screws.)
2. Require special care to insure that the ways are kept clean, dry, and free from corrosion.
3. Somewhat lower load capacity.
4. Less flexibility in locating the load. (On a tool slide, the boring-bar holder could not be moved from one end of the slide to the other as could be done with a roller-bearing slide.)

Many factors must be considered before a recommendation can be made to convert any given machine to air-bearing slides. However, all these factors must be tied to the nature and purpose of the machine. The tighter the tolerance the machine must hold, the more advantageous air bearings become in that, if properly applied, they can provide a smaller contribution of slide-way error to total machine error than any other bearing system. This factor becomes perhaps more significant in choosing a design for new machines with more exacting tolerances.

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APPENDIX A

SLIDE-BEARING DESIGN

Area

Total slide weight = 1,580 pounds, including the weight to balance the boring bar. Choosing a 40-psig film pressure for a noncaptive system with four pads:

$$A = \frac{wt}{P_f}, \text{ or}$$

$$A = \frac{1,580}{40} = 39.5 \text{ in}^2 \text{ projected upward.}$$

Projected area of each pad:

$$A = \frac{39.5}{4} = 9.9 \text{ in}^2.$$

Stiffness

Stiffness, S , can be approximated by:

$$\text{Area} \times \frac{P_s - P_f}{h}.$$

Choosing an "h" of 100 microns:

$$S = 9.9 \frac{80 - 40}{0.0001} = 9.9 (400,000) = 3,960,000 \text{ lbs/in.}$$

$$\text{Compliance} = \frac{1}{3,960,000} = 0.25 \text{ microinch lb/pad.}$$

Flow

By equating the slot-flow equation for a recessed hydrostatic bearing with the Darcy's law equation, design permeability and flow per unit area can be calculated.

The slot-flow equation is:

$$Q = \frac{bh^3 (P_f^2 - P_a^2)}{24MlP_f},$$

where:

b represents the perimeter of the bearing in inches,

h the film thickness in inches,

P_f the film pressure in psia,

P_s the supply pressure in psia,

M the viscosity in Reyns,

l the equivalent slot length in inches,^(c)

P_a the ambient pressure in psia,

K the permeability in millidarcies, and

A the area in in².

The Darcys' law equation is:

$$Q = \frac{KA (P_s^2 - P_f^2) \times 1.53 \times 10^{-12}}{2MTP_f}$$

Choosing h to be 200 microinches and equating these two equations and solving for Q/A yields:

$$Q/A = 0.008 \text{ scfm/in}^2.$$

This number was the approximate value used for impregnation.

(c) Steger, P. J.; *Air-Bearing Spindle for Machine Tools - Spherical Design*, Y-1581, p 17; Union Carbide Corporation-Nuclear Division, Oak Ridge Y-12 Plant, Oak Ridge, Tennessee; September 29, 1967.

APPENDIX B

DESIGN PROCEDURES

Pad Design

Bearing area design procedures are listed in Appendix A.

Typical Pad Support Design (Work Slide)

Bearing area was calculated to carry 1,700 pounds per work slide V-way pad at a 40 - 50 psi film pressure. Calculated length was approximately 11.75 inches.

Calculation of the Moment of Inertia

Maximum bending deflection was set at 50 microinches at the end of the pad. It was decided to use the bending equation for a uniformly loaded cantilever beam since it most closely correlated with the actual situation:

$$y_{\max} = \frac{Wl^3}{8EI}$$

$l = 4 \frac{3}{8}$ inches from the edge of the ball seat to the end of the pad.

$$50 \text{ microinches} = \frac{850 \text{ lbs } (4 \frac{3}{8})^3}{30 \times 10^6 I}$$

Solving for I yields:

$$I = 5.9 \text{ in}^4$$

Using this value and working with the geometrical limitations of the V way, a configuration, as depicted in Figure B-1, was obtained. Final dimensions were calculated by increasing the vein forcing web height to obtain the desired moment of inertia which was calculated by the aid of a computer program. The results are listed in Table B-1.

Figure B-1. FINAL V-PAD SUPPORT DESIGN CONFIGURATION.

Table B-1

**FINAL CROSS-SECTIONAL DIMENSIONS
OBTAINED FOR V-PAD SUPPORT**

H = 2.125 inch**L = 1.75 inch****B = 4.125 inch.**

Note: This configuration yields a moment of inertia of 6.26
in⁴ which is higher than required.

Similar calculations were performed on the flat-way pad. However, the V pad is most critical because of its length.