SEP 3 1965

MASTER

Date Issued: September 1, 1965

Document Number Y-DA-921

Conf-650651-1

UNION CARBIDE CORPORATION
Nuclear Division

Y-12 PLANT

Contract W-7405-eng-26
With the US Atomic Energy Commission

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PRODUCTION-SIZE LATHE HYDROSTATIC COMPONENTS

P. J. Steger

A paper to be presented at the ASME Spring Lubrication
Symposium in New York City on June 7-9, 1965.

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Larry E. Williams

Authorizing Official

Date: *06/24/2005*

Oak Ridge, Tennessee
May 21, 1965

PRODUCTION-SIZE LATHE HYDROSTATIC COMPONENTS

The successful application of hydrostatic gas-lubricated bearings at DuPont, using a porous material for the bearing pad, indicated that the use of this type of bearing for production machine tool components is feasible. As a result, two lathe components were fabricated at the US Atomic Energy Commission's Oak Ridge Y-12 Plant as a part of a development program to improve the accuracy of machine tools. The Y-12 Plant is operated for the AEC by the Union Carbide Corporation-Nuclear Division.

The first of these components was a hydrostatic boring bar (Slide 1). This is a cross section of the boring bar. Although it is a precision slide, its construction is such that it does not require holding close tolerances on all components during fabrication and assembly. The only precision requirement is for flatness and, to a minor degree, parallelism of the pad surfaces and surfaces of the bar. Parallelism deviation is limited to the travel of the springs used to obtain pad loading against the bar. The tolerance specified on the fabrication drawings for components parts was ± 0.005 inch.

The bearing pads are of the same porous graphite material as used in the spindles described in Mr. Lewis' paper. After machining, they were attached to the shoes with epoxy and then lapped to a flatness of 10 microinches. The bar was lapped to a flatness of 20 microinches on each surface and parallel to 0.0005 inch for each pair of opposing sides. The shoes are supported on spherical seats and spring loaded against the bar with a force equal to that produced by the air film pressure at a film

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CROSS SECTION OF HYDROSTATIC AIR-BEARING BORING BAR

thickness of 25 - 50, microinches. The deflection rate of the bar is ten micro-inches per pound of load measured at the tool position with the bar extended 12 inches, and with an air supply pressure of 90 psig. At this supply pressure and a film thickness of 25 - 50 microinches, air consumption is only 1/2 scfm.

The accuracy of movement over a 12-inch span, as measured with an alignment interferometer and an electronic level, was:

Yaw - 20 microinches,
Pitch - 60 microinches,
Roll - 2 seconds.

This bar was designed for use in a dusty, abrasive atmosphere. After testing under these conditions without shielding or way covers for over 70,000 cycles, using a six-inch stroke and a 60-pound simulated tool force, the bearing pads and bar showed no significant deterioration. Slide 2 shows a view of the bar in its test environment.

Another outstanding feature of this boring bar is its ease of maintenance. If one of the pads should become plugged, or damaged, it can be quickly and easily replaced. You simply back off the pad-adjusting screw and slide the pad out, as shown in Slide 3.

Slide 4 shows the hydrostatic boring bar or slide assembly. It has an accuracy which compares with the best mechanical slides and yet has simple fabrication requirements, low operating costs due to small air flow requirements, negligible friction and wear characteristics, and simple maintenance if ever needed.

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BORING BAR IN DUSTY, ABRASIVE TEST ENVIRONMENT

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BORING BAR WITH PAD REMOVED

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BORING BAR READY TO MOUNT ON LATHE

The second hydrostatic air-bearing component built was a production-size spindle which is to be mounted on a 15 by 15-inch tape-controlled turning and boring machine. A cross section of the spindle design is shown in Slide 5. The spindle's journal bearings are seven inches in diameter and the thrust bearings are 12 inches in diameter. It has an overall shaft length of 32 inches.

Porous graphite material was also used for these bearing pads. The thrust bearings which are located at each end are preloaded with a flexure assembly located at the rear thrust which serves to take up shaft expansion. Since heat generation is low, by using a high deflection rate, shaft growth can be taken to the rear with little change in the thrust film thickness. Thus, the only forward growth which occurs is expansion of the front thrust runner and the increase in the front thrust film. The preload force is equal to that produced by the thrust film pressure at the operating film thickness.

The journal bearing assembly is made up of three shoes supported on spherical seats mounted on radial flexure units. They are positioned radially by means of differential screw assemblies.

Slide 6 shows a breakdown of the journal shoes. The shoe is machined to allow a proper air distribution to the porous pad. The design closely resembles that of the spindles designed by Mr. Lewis of DuPont who served as a consultant on the project.

The radial stiffness of the spindle measured at a point six inches in front of the face is 1.5 microinches per pound of load applied at this same point. Axial stiffness is

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CROSS SECTION OF HYDROSTATIC AIR-BEARING SPINDLE

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BREAKDOWN OF JOURNAL BEARING SHOE

0.05 microinch per pound. These measurements were made with a 0.0003-inch film thickness in both the thrust and journal bearing. Comparing these with the calculated values, we found that we could accurately predict spindle stiffness using a simple supported beam equation for calculating shaft deflections. Slide 7 shows the assembled spindle under test.

The accuracy of the spindle was measured using two capacitance-type proximity transducers, oriented 90 degrees out of phase, with their output displayed on an oscilloscope. The static radial runout measured was eight microinches, TIR, with a probable accuracy of $\pm 20\%$. The axial runout was five microinches. The dynamic runouts at 1400 rpm were:

Radial - 8 microinches,

Axial - 10 microinches.

Heat generation in the bearing is phenomenally low compared to standard ball-bearing spindles. Calculations and tests indicate that approximately 1/4 horsepower is required to rotate the spindle at 1500 rpm. Although spindle growth measurements have not been concluded, thermocouples installed in the carbon pads indicate insignificant change in temperature for runs of 2 - 3 hours at speeds to 1000 rpm.

Here, again, as with the slide, construction of the spindle is such that no close dimensional tolerances are needed for fabricating the components. Precision requirements are roundness, concentricity, and perpendicularity of the shaft.

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ASSEMBLED SPINDLE UNDER TEST

An outstanding feature of the porous graphite pads is that the loss of air film during rotation will not have the same devastating effect that would occur with a normal hydrostatic spindle, since graphite has fairly good lubricating qualities on its own. Loss of air has occurred during testing at 1500 rpm without obvious damages. No change in rotational accuracy of the spindle has been detected after air failure followed by sudden seizure of the shaft assembly.

Slide 8 shows the spindle ready for installation on a tape lathe. No machining has been done with the spindle to date.

Laboratory tests of these hydrostatic air-bearing systems have been very encouraging. Their outstanding characteristics have given impetus to our continuing efforts to significantly improve the precision and long-term reliability of machine tools.

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SPINDLE READY TO MOUNT ON LATHE