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DESIGN, DEVELOPMENT AND EVALUATION
OF A PRECISION AIR BEARING ROTARY TABLE
WITH
LARGE DIAMETER THROUGH-HOLE

BY

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B.S. (CALIFORNIA STATE POLYTECHNIC UNIVERSITY) 1980
SUBMITTED IN PARTIAL SATISFACTION OF THE REQUIREMENTS FOR
THE DEGREE OF MASTER OF SCIENCE IN ENGINEERING
IN THE GRADUATE DIVISION OF THE
UNIVERSITY OF CALIFORNIA, DAVIS

MASTER

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ACKNOWLEDGEMENTS

The author would like to thank the Lawrence Livermore National Laboratory Machine Tool Development group in the Materials Fabrication Division, and Bradley L. Fix and Daniel C. Thompson in particular, for the opportunity to be responsible for the design, fabrication and testing of the Certification of Process Gage Rotary Table. Thanks and appreciation are also extended to Stan Edson and Stan Locke for their design support and to Jose Vargas, Jay Jacobs and Charlie Cass for their unfailing patience in fabricating and assembling the Rotary Table. The author would like to extend his appreciation to Ken Blaedel and Professor Harry Brandt for their technical expertise throughout the project as well as for their guidance in drafting this document. And finally, special thanks are due to Vivian Shinmoto and Suzette Scott for their patience and skill in assisting with the typing of this thesis.

Work performed under the auspices of the U.S. Department of Energy by the

Lawrence Livermore National Laboratory under contract W-7405-Eng-48

Abstract

A large diameter precision air bearing rotary table with a 16.0 inch diameter through-hole was designed, fabricated and tested in the course of this research. The rotary table will be used in conjunction with a specialized, computer controlled precision inspection machine being designed for the Department of Energy's (DOE) Nuclear Weapons Complex (NWC). The design process included a complete engineering analysis to predict the final performance of the rotary table, and to ensure that the rotary table meets the required accuracy of 4.0 microinches of total radial (3.5 microinches average radial) and 4.0 microinches total axial (3.5 microinches average axial) errors. The engineering analysis included structural deformation, thermal sensitivity and dynamic analyses using finite element methods in some cases, as well as other analytic solutions. Comparisons are made between predicted and tested values, which are listed in the rotary table error budget. The rotary table performed as predicted with measured axial and radial stiffnesses of 1.1E06 lbf/inch and 2.9E06 lbf/inch, respectively, as well as average radial, axial and tilt errors of 2.5 microinches, 1.5 microinches, and less than 0.05 arcseconds, respectively.

Introduction

During the design phase of a precision machine or component

one must consider possible sources which could introduce error motion into the system. In the analysis of error sources it is common to attempt to isolate each of the sources, and then to categorize them as either repeatable or non-repeatable errors¹. Repeatable errors have the potential of being minimized using compensation techniques, whereas non-repeatable errors must be dealt with directly to either remove the sources of non-repeatability or reduce them to acceptable levels. Blaedel² refers to the latter approach as error avoidance. The strategy that has been applied to this research was to use error avoidance in the design phase to eliminate as many non-repeatable errors as possible by application of sound design principles³⁻¹¹ then to map any repeatable errors and use software compensation to meet the rotary table accuracy requirements published in the Certification of Process (COP) Gage Error Budget¹².

The rotary table (Figure 1) has several areas that required research and innovative solutions to meet specific accuracy requirements, which are presented in Table 1 and will be discussed by major sub-headings in this study. Specifically, these areas are the Mechanical Design, Thermal Sensitivity, Dynamic Performance, and Position Feedback.

Background

The DOE Precision Flexible Manufacturing System (PFMS) project is an NWC effort to automate the manufacturing and certification of

Figure 1. COP Gage Air Bearing Rotary Table

TABLE 1 - COP GAGE ROTARY TABLE ERROR BUDGET

ROTARY TABLE GEOMETRY ERRORS

	PREDICTED	TESTED
Radial Average Error	3.5 microinches	2.5 microinches
Axial Average Error	3.5 microinches	1.5 microinches
Tilt Average Error	0.08 arcsecond	< 0.05 arc-second

ROTARY TABLE THERMAL ERRORS

	PREDICTED	TESTED
Thermal Effects - Drive Motor	3.0 microinches	< 1.0 microinch

precision hemispherical shell components. One of the key features of an automated flexible manufacturing system is to be able to "certify the process" such that eventually, the inspection of every part would be unnecessary. The COP measuring machine is an essential piece of equipment in certifying the manufacturing process by accurately measuring hemispherical shells produced by a PFMS cell¹³. While the primary motivation for the COP gage is to diagnose the hemishell manufacturing process, part certification may be a reasonable extension of this machine. Therefore, as many general inspection requirements as possible (stiffness, repeatability, part access, thermal stability, etc.) have been considered in the COP gage design.

Figure 2 shows the configuration of the COP gage. A granite base supports the y-slide and lower z-ram as well as the welded INVAR bridge structure, which in turn supports the upper z-ram. The y-slide supports the rotary table and provides motion in the horizontal y-direction. The measurement probes are supported from the upper and lower z-rams which move in the vertical z-direction as the part, which is mounted on the rotary table, traverses in the y-direction and rotates.

The design of the COP gage has been carried out using a modern precision engineering approach. The requirements for high accuracy (30 microinch maximum allowable contour error), relatively high speed traversing (25 inches per minute) and high data acquisition rate (2000 samples per second) have resulted in challenges in both mechanical and control system design. One of the major

CERTIFICATION OF PROCESS GAGE

Figure 2. CDP Gage

Physical Characteristics

Slide Travels: Upper Z = 10.00 in.
 Lower Z = 18.00 in.
 Y = 24.00 in.

Overall Dimensions: Height = 123.50 (10'3")
 Width = 78.00 (6'6")
 Length = 132.00 (11'0")

subassemblies essential to the design of the machine is the rotary table package.

The COP gage rotary table design is shown in Figure 1. The rotary table consists of a stator that is supported by the y-slide and a rotor that is supported on aerostatic bearings by the stator. A position encoder ring is mounted to the top of the rotor and the encoder read-head system is mounted to the top of the stator. A direct-drive D.C. motor is mounted to the bottom of the rotary table, with the rotating components connected to the rotor and the stationary components mounted to the stator.

Precision air bearing rotary tables are available from several commercial vendors^{14,15}. What makes the COP gage rotary table design unique are the large outer diameter of 32.0 inches, a through-hole diameter of 16.0 inches and a height limitation of 8.5 inches. The large through-hole is necessary to allow access of the measurement probes to both the inner and outer part contours in a single part set-up, thus eliminating non-repeatable set-up errors. The rotary table project also provided the opportunity to use analysis as a design tool to predict the performance of the rotary table during the design phase, which may not always be an option in industry depending on the time and resources available.

One of the key areas of study during this research that falls under the sub-heading of Mechanical Design was the design of the journal and thrust air bearings. The bearings were designed to meet the stiffness and load capacity specifications in the requirements document (Appendix 1). The bearings were also designed to

minimize air flow to reduce the effects of Joule-Thomson cooling. Material issues were addressed to ensure that the rotary table bearings could be economically manufactured, maintained, and repaired as necessary.

The next key area mentioned in the introduction is that of thermal sensitivity. The rotary table will be subjected to thermal loading from several sources which could introduce errors. This research investigates the effects of rotary table errors resulting from the heat from the drive motor and Joule-Thomson cooling. Methods of minimizing the effects of thermally induced errors were recommended and implemented where necessary.

The third key area addressed in this research is that of the rotary table dynamic performance. It has been determined that the first mechanical resonant mode of the gage structure must be 80 Hertz or above to meet the inspection time, data acquisition rate and servo performance goals. This research includes dynamic analysis performed during the design phase to address this issue.

The last key area discussed is the rotary table position feedback. The overall size and configuration of the rotary table components demand a position feedback system that uses new encoding technology to economically meet the resolution and accuracy requirements¹⁶⁻¹⁸. The number of encoder systems available in this size and geometry, and with the required positioning accuracy and repeatability is virtually non-existent. A commercial vendor¹⁷ worked with the project to modify one of their prototype products to provide the encoder system that was implemented on the rotary

table.

Mechanical Design

The design of a precision piece of machinery requires the integration of many interdependent design factors such as bearing design, thermal effects, allowable and predicted errors, material selection, fabrication techniques, and final assembly procedures. The success of the final product is dependent on how these factors are allowed to interact. The decisions and trade-offs between these design factors for the rotary table are discussed in the following section.

Journal, Thrust, and Preload Bearings

Before the basic issues such as accuracy, size, material, manufacturability, and thermal constraints could be considered for the COP rotary table design, the more basic issue of the type of bearings (hydrostatic vs. aerostatic) used for the journal and thrust bearings needed to be addressed. The choice between oil or gas lubrication typically depends upon the type of application for which the bearings are intended¹⁹. In this case, the requirement for a bearing that does not contaminate the inspection environment, along with the need for moderate stiffness, load capacity and high accuracy, pointed to aerostatic bearings as the design of choice.

The next decision was to select the particular type of air

bearings to be used (step type vs. orifice), based on the performance requirements of the rotary table (Appendix 1) as well as on the cost and manufacturability issues associated with each type of air bearing design. Orifice bearings (Figure 3) generally offer higher performance characteristics such as stiffness and load capacity and are more efficient in terms of required area, but are more costly to manufacture due to their more precise tolerances and the size and number of orifices. Step-type air bearings (Figure 4) are easier to manufacture but also tend to provide more moderate stiffnesses and load capacities. Consequently, step bearings generally require more area to support a load.

Journal Bearing

The moderate stiffness requirements of $5.0\text{E}05$ lbs/inch radial and $67.0\text{E}06$ lb-inch/radian tilt for the rotary table as defined per the ANSI/ASME B89.3.4M-1985 specification²⁰ lead the designer to a step-type design for the journal bearing. Coupled with the stiffness requirements is the geometry of the rotor and stator such that there is sufficient area for a step-type bearing design and also the fact that step bearings would be less expensive to manufacture in this case. Hence, the journal bearing of the rotary table was designed as a step bearing of the type referred to by Donaldson²¹ as a non-uniform clearance bearing where the generation of a restoring force depends upon a built-in variation in the radial clearance. Donaldson's method starts by deriving the

Figure 3. Typical Orifice Air Bearing Design

Figure 4. Typical Step Air Bearing Design

Reynolds equation for an incompressible Newtonian fluid, then reducing the equation to provide an approximate solution. The theory is based on laminar viscous flow without turbulence which implies Reynolds numbers below 100 in the film and preferably smaller. Two methods are available to solve the Reynolds equation that pertain to this design. The first, called the "small eccentricity ratio solution", gives exact values for the initial slope of the force-deflection curve including circumferential flow but does not address the non-linear behavior of the bearing at higher eccentricities. The second, the "short bearing solution", calculates the forces over the full range of displacement but is an approximation because circumferential flow is ignored. The approximation used to design the rotary table journal bearing was the method where the small eccentricity ratio solution is calculated and is then used to ratio the short bearing solution, providing an improved solution over using either method by itself. Because of the relatively moderate stiffness requirement of 5.0×10^5 lbs/inch coupled with a desired flow rate on the order of 2 SCFM or less the bearing was designed with a relatively small clearance of 300 microinches. The journal bearing design includes 0.25 inch wide lands, a step height of 240 microinches, and a gap of 300 microinches. The calculated flow rate is 1.1 SCFM at a working pressure of 30 psig. The final bearing dimensions are shown in Figure 5. The final design shows the step bearing detail machined into a bronze sleeve, which is shrunk into the stator housing. The bronze sleeve is for crash resistance and is discussed below. The

Figure 5. Rotary Table Journal and Thrust Bearing Designs

pressurized air is fed from a manifold behind the bronze sleeve, through eight 1/16 inch diameter holes and into a 0.125 inch square feed groove in the rotor side of the bronze sleeve.

Axial Bearing

The moderate axial stiffness requirement for the rotary table of $1.0\text{E}06$ lbs/inch combined with the working load capacity requirement of 200 lbs also point to the choice of a step-type bearing for the thrust application. Again, the geometry of the rotary table provides a sufficient bearing area for the relatively less expensive step-type bearing design. A problem encountered here was that the workpieces that are to be inspected on the rotary table range from 20 to 200 lbs, and if an air bearing were designed for some nominal load capacity between these two values, then the bearing performance parameters such as gap and stiffness would vary significantly as the part weight ranged from the minimum to the maximum values. One solution to this problem would be to design the bearing to be extremely stiff, on the order of $1.0\text{E}07$ lbs/inch. The solution chosen in light of the rotary table geometry, however, was to design another air bearing to preload the rotor against the thrust bearing to about 80 percent of the nominal preload required by the thrust bearing design. The part to be inspected then supplies the additional nominal required preload and the rotor is thus captured between the preload and thrust bearing. The exercise here was to design the preload bearing in a manner in which it

supplies the required preload to the thrust bearing, but is non-influencing to the axial stiffness of the thrust bearing. In this case the preload bearing was designed to have an axial stiffness of $5.0E04$ which is $1/20$ of the axial stiffness of the thrust bearing. This allows the thrust bearing to operate to within about ± 10 percent of the calculated performance values, depending on the part weight. Both the preload and thrust bearings were designed according to research by Roblee²² using his hybrid model of a step bearing which not only retains the lumped parameter representation of the perimeter region and recess of the step-type bearing, but also includes an approximate squeeze-film model for the interior region of the bearing. For the class of narrow perimeter-region bearings, Roblee states that it is reasonable to approximate the gas flow over the perimeter as purely longitudinal, with the perimeter region modeled as an equivalent rectangular strip with a length and width which are combined with the ideal gas law to formulate an equation for the mass flowrate through the perimeter. The mass flowrate of the interior is equated to the mass flowrate of the perimeter for each half of the step bearing separated by the feed groove, which then allows the calculation of the internal pressures for each half of the bearing since the pressures in the feed groove and outside the lands are known. Once the internal pressures are determined, the load capacity is calculated for each region of the bearing and summed to arrive at the total working load capacity for the step bearing. The flowrate is calculated next by substituting the pressures back into the equation for mass

flowrate, recognizing that the total flowrate is the sum of the mass flowrates out of each side (inside and outside diameters) of the step bearing. To calculate the bearing stiffness, the gap was varied a small amount and the change in the total load capacity due to that small change in gap was computed. The total stiffness was then calculated as the change in load capacity divided by the change in gap. The performance parameters calculated for the thrust bearing included 0.15 inch wide lands, a step height of 300 microinches, a gap of 350 microinches, and a flowrate of about 1.7 SCFM at a working pressure of 30 psig. Similarly the preload bearing was designed with 0.15 inch wide lands, a step height of 2000 microinches, a gap of 400 microinches and a flowrate of about 4.0 SCFM at a working pressure of 30 psig. In an effort to maximize the moment stiffness of the rotary table both the preload and thrust bearings were designed with a 0.125 inch square air feed groove which is segmented into eight sections, with one pressurized air feed per segment.

It should also be noted that all three air bearings on the rotary table were designed for a relatively low working pressure of 30 psig since the first COP Gage could end up at any one of three or four sites within the DOE Complex, some of which may not have high pressure air readily available. Additionally it was desired to keep the total air consumption for the rotary table air bearings to about 6 or 7 SCFM in an effort to minimize Joule-Thomson cooling effects which are discussed below. The total calculated air flow for the final journal, thrust and preload bearing designs is 6.8

SCFM. The thrust and preload bearing designs are presented in Figures 5 and 6, and the complete thrust and preload bearing design calculations are presented in Appendix 2.

Joule-Thomson Effects

The total air consumption of the journal, preload and thrust air bearings on the rotary table is worthy of consideration because of possible Joule-Thomson cooling effects. The Joule-Thomson effect in air is a process where air changes in temperature when it is throttled²³. The effect on a design such as the rotary table where the air bearings are symmetric and two are opposed would be to affect the time it takes for the rotary table to reach thermal equilibrium from start-up. If the bearings were not symmetric, then localized cooling of some areas and not of others would result in possible time-varying errors in roundness and flatness. Hence, symmetry was stressed in all three air bearing designs discussed above. The Joule-Thomson effect due to all three rotary table air bearing designs was calculated in the manner suggested by Blaedel and Donaldson²⁴ to be 0.71 Watts, which is low enough not to be of concern. The calculations were performed by deriving an expression from a temperature-entropy diagram for the refrigeration that a real gas can sustain as it undergoes an adiabatic expansion. It should also be noted that the 0.71 Watts of Joule-Thomson cooling is substantially greater than the 0.0151 Watts of heat calculated due to the shearing of the air film in the thrust and journal

Figure 6. Rotary Table Preload Thrust Bearing Design

bearings. This means that when the rotary table is operating at the maximum circular velocity of 10 rpm the net effect is still almost 0.7 Watts of refrigeration.

Rotary Table Geometry Errors and Predictions

The accuracy requirement for the overall COP gage as stated previously is 30 microinches of total allowable contour error. This number is the total of many component error factors, one of which is the total error of the rotary table. The rotary table error budget, shown in Table 1, contains the allowable predicted average errors for the rotary table design package, separated into the radial, axial and tilt components. The actual measured values are also presented in Table 1. Radial motion is defined according to the ANSI/ASME B89.3.4M-1985 Axes of Rotation Standard²⁰ as error motion in a direction normal to the Z reference axis and at a specified axial location. Likewise, axial motion is defined as error motion collinear with the Z reference axis, and tilt motion is defined as error motion in an angular direction relative to the Z reference axis. All three motions are illustrated in Figure 7. Each total error can be broken down further into the two components of average error and asynchronous error, which are best illustrated by the error plots shown in Figure 8. The total Error Motion Polar Plot is defined as the complete error motion polar plot as recorded. The Average Error Motion Polar Plot is the mean contour of the total error motion polar plot averaged over the number of

Figure 7. Definitions of Error Motions

(a) Total Error Motion

(b) Average Error Motion

(c) Asynchronous Error Motion

Figure 8. Polar Plots of Error Motion and its Components

revolutions and shown as the thick line in Figure 8b. The Asynchronous Error Motion Polar Plot is the deviation of the total error motion polar plot from the average error motion polar plot.

Allowable Errors

Any errors of the rotary table, such as the radial, axial and tilt total errors, appear to be errors in the part being measured. Typically these types of errors are not detected and hence cannot be corrected for. The machine could be mapped to remove repeatable errors but this is a very time consuming process and would hinder the use of the rotary table as a stand-alone precision package. Another solution would be to implement a real time error correction system which employs interactive hardware such as capacitance gages, but this would add to the cost of the machine and decrease the reliability. The solution chosen in this case was to design the rotary table to an accuracy that could be achieved economically but does not contribute significantly to the COP error budget. This required accuracy is reflected in the predicted average radial, axial and tilt errors shown in Table 1.

Predicted Errors

In evaluating the impact of the predicted 3.5 microinch average radial and axial errors and the 0.1 arc-second average tilt error on the manufacturing of the rotary table, it needs to be

recognized that some averaging effect due to the air bearings comes into play. Typically, it has been found^{8,9,10,11} that the average error of an air bearing with this type of geometry is about 1/10 of the measured roundness of the component parts. This means that in order to achieve 4 microinches of total allowable axial error the rotor and stator each need to be machined to about 20 microinches each of total runout which seems achievable in light of the many commercial products available^{14,15}, though in smaller sizes, that are accurate to 40 microinches.

Air bearing rotary tables generally exhibit much less asynchronous error than average error. Average errors, or how well one can make something round, are a function of the quality of the manufacturing process. Asynchronous errors, or errors that vary from one revolution to another, are due to transient effects such as dirt or thermal variations and are typically of low order in air bearing tables because of the beneficial averaging effects. This is the basis for predicting and measuring only the average errors for the rotary table as presented in Table 1.

Measured Errors

The tested values presented in Table 1 were measured and displayed using the polar plot method shown in Figure 8.

Rotor and Stator Material Selection and Design

Machinability and Crash Resistance

To select the material for the rotor, stator and preload ring, the machinability, crash resistance/recovery and robustness of the rotary table package need to be considered because they impact one another. The first choice of material for parts that are as large, heavy and precise as the rotor and stator is steel. However the final machining tolerances on all three parts demand more precise machining than conventional turning and milling machines can offer and in fact are more typical of precision grinding or diamond turning processes. The first approach is to consider diamond-turnable materials (aluminum, bronze, nickel, plastics, etc) as candidates for the rotor, stator and preload ring. When other requirements are factored in such as cost, stability and strength, the only real possible candidate remaining out of the available diamond-turnable materials is aluminum. Aluminum presents a solution to the final machining problem but other characteristics are less desirable when comparing aluminum to steel. Steel is stiffer which has an impact on structural deflection under loads. Steel is also desirable from a thermal point of view with a coefficient of thermal expansion about half that of aluminum. Conversely, steel is not diamond-turnable. Thus the problem becomes a trade-off between being able to diamond-turn the aluminum but perhaps not being able to live with the errors that the weight and

thermal perturbations induce, or finding a way to machine or grind the steel to the tolerances that are needed to provide the required accuracy. It was decided to explore ways of machining the steel before accepting aluminum as the material by default.

The first step in the process was to try to find available machines that were accurate enough to fabricate the parts. The first choice at LLNL was to examine the possible use of one of the many precision NC "T-Base" lathes. These machines generally have accuracies in the range of 5 to 10 microinches for parts up to about 16.0 inches. The problems encountered upon investigating these machines further were the limited slide travels and spindle load capacities of about 500 lbs. It should be noted that the dimensions of the rough machine blank for the stator are on the order of 33 inch diameter and 6 inch thickness. The blank weighs about 600 lbs. LLNL does not have dedicated precision grinders with the capacities to handle the size and weight of the stator, and commercial companies with grinding or turning machines capable of handling the stator were either backlogged for months or were cost-prohibitive because of the one-only nature of the rotary table parts. The next step was to examine the possibility of single point turning the steel on a diamond-turning machine to achieve the required accuracy. Diamond-Turning Machine 3 (DTM-3) at LLNL was considered as a logical choice. DTM-3 is an oil-showered horizontal spindle diamond-turning machine with proven accuracies of 1 to 5 microinches⁴. The LODTM³ at LLNL would have been an even better choice due to the vertical spindle configuration of the machine,

but was eliminated because of the steel contamination problem in a machining environment dedicated primarily to the fabrication of precision optics.

A cutting test was designed to analyze whether or not stainless steel could be single point turned on DTM-3. A number of tools and tool inserts were obtained, including carbide T-16, various grades of CBN (BN100, BN300), polycrystalline diamond, chinese cobalt, and ceramic (WG300). A 17-4PH stainless steel plate was rough-machined to simulate the approximate locations of the thrust bearing detail including the segmented feed grooves on the stator as well as the kinematic mounting pads on the rotor. The plate was then hardened to 40-44 Rockwell C and mounted to a faceplate. The faceplate supporting the test plate was mounted onto DTM-3 and was machined using the various tools and tool inserts. Most of the tool cutting edges broke down when they encountered the interrupted cut over the pads, except for the cobalt tool which handled the pads as well as the air bearing detail simulating the thrust bearing in the stator. By running at about 88 sfm and by taking about an 0.001 inch depth of cut, the cobalt tool cut the pads and the bearing detail flat to less than 20 microinches. Some tool wear was observed during the cut so compensation was implemented since the tool wear seemed predictable. It appeared at this point that a possible solution to the rotor, stator and preload ring machining problem was to single-point turn the parts on DTM-3.

The problem of crash resistance / crash recovery was

considered next in the selection of materials. It should be noted that there are two requirements for crash resistance in the rotary table design. One is between the rotor and stator thrust surfaces and the other is between the rotor and stator journal surfaces. The maximum speed of the rotary table is 10 rpm under normal operating circumstances and while this is not fast by air bearing spindle operating standards, damage could occur if the air were shut off to the table or if the rotor crashed into the stator for any other reason while rotating. Hence in the development of a piece of precision equipment such as this, it is desirable to account for such eventualities by building a course of recovery into the design. This can be done in several ways. If the parts were made out of aluminum, the rotor and stator could be hard anodized and teflon impregnated, providing a hard, lubricated surface if the rotor should crash into the stator. If the parts were made out of steel then there could be many courses of crash protection. One would be to bond some type of material such as plastic or teflon to one or both of the surfaces and then try to machine the air bearing detail into that material. This was eliminated due to the difficulty of machining plastic to microinch-type dimensions and flatnesses. Also materials such as plastic do not enjoy dimensional stability after machining. Another possibility would be to make one of the rotary table components out of a material such as porous graphite, which has been done successfully in some air bearing applications²⁵. This approach allows the possibility of supplying the pressurized air directly through the graphite, eliminating the

need for feed holes and grooves. However, the size of the parts involved in this design as well as the inspection-type environmental requirement of the rotary table eliminated this design, since graphite is considered relatively dirty compared to other materials. Various types of platings on steel were also considered for this design, such as plastic, teflon, nickel and teflon-impregnated nickel. These materials have the added attraction of being diamond-turnable. However, the teflon and plastic did not adhere to the steel when this was attempted at LLNL and were thus eliminated. There has been little success at LLNL for similar applications of plating nickel onto steel and then diamond-turning the nickel. Even if the nickel is plated successfully onto the steel, delamination can occur during the diamond-turning or during actual operation of the parts involved. Because of the success in single-point turning of the stainless steel test piece on DTM-3, it was decided to eliminate aluminum and to make the rotor, stator and preload ring out of stainless steel 17-4PH in the "A" condition, hardened to about 40-44 Rockwell with a bronze insert designed into the journal bearing diameter to provide crash resistance. The 17-4PH grade of stainless steel was selected due to the fact that it is inherently corrosion resistant, very machinable, and has proven to be stable over time. The "A" condition designates that the forgings have been solution annealed by heating to 1900 degrees F for half an hour then cooled in air or oil in order to facilitate the machining process. The inclusion of the bronze insert would allow diamond-turning of the journal

bearing detail into this insert and also provides crash resistance between the stainless steel rotor and the bronze journal surface. The bronze insert is shrunk into the stainless steel stator so that if damage does occur, then the bronze either can be re-machined or else a new bronze insert can be shrunk into place and machined to dimension. In the unlikely event that the journal surface of the rotor also sustains damage, the journal surface could be re-machined and the new dimensions of the bearing could be adjusted by machining the thickness of the bronze insert in order to maintain the original gap between the rotor and stator.

This now leaves the problem of crash protection between the rotor and stator at the location of the thrust bearing. The amount of damage that would occur between these two surfaces is a function of the speed of rotation of the rotor and the weight of the workpiece on the table when the rotor hits the stator. The damage could range from none or minimal to serious. If serious damage did occur, then the solution would be to re-machine the thrust surfaces of the rotor and/or stator to eliminate the damage, and subsequently adjust the flying height of the thrust and preload bearings back to their nominal dimensions by machining the height of the preload ring. If the damage to the rotor were extreme, then a new rotor could be fabricated although the odds of this happening are low. The choice of hardened stainless steel offers a higher degree of corrosion resistance and provides a more robust solution to the design problem when one considers the machinability problems and the hazards imposed to the rotor and stator during the assembly

(and disassembly) process.

Manufacturing Plan

The next step in the design process is to determine the method of manufacturing the rotary table parts and to outline the manufacturing steps involved from start to finish. Originally, part of the charter of the COP Gage project was to involve an industrial partner which was selected from a number of candidate precision machine tool manufacturers, in the fabrication of all of the COP gage components including the rotary table. This was mandated by DOE in an effort to facilitate the manufacture of possible future machines duplicating the COP Gage. In light of this mandate it was decided to specify the rough and finish manufacturing steps involved in fabricating the rotor and stator, recognizing that the vender could choose to deviate slightly from the specified manufacturing procedure as long as the end product met the tolerances specified on the rotor and stator design drawings. As the design evolved, however, it became clear to the industrial partner that he did not have the capability to finish-machine parts as large as the rotor and especially the stator in house. This led to the scenario alluded to earlier where the rotor and stator parts would be rough-machined outside of LLNL and then finish-machined at LLNL on DTM-3. After this process was well under way it became clear to the industrial partner that he did not even have the capability to rough machine the parts in house due to other

manufacturing commitments and when he sent bids outside, schedule and cost issues further complicated the matter. It was decided at this point to rough machine the rotor and stator at LLNL using the conventional machining facilities within the Materials Fabrication Division, and so the manufacturing plan was adapted accordingly.

The manufacturing plans for both the rotor and stator are divided into rough machining and final machining processes, separated by the heat treatment where the parts were hardened to 44-48 Rockwell C. The high hardness was desirable since grinding could still be implemented as a back-up position and also since the hardness of the stainless steel of 40-44 Rockwell C used in the cutting test did not seem to hinder the single point turning of the material. Both plans are similar in methodology, starting out by rough machining one face, then turning the part over and machining the other, then turning the part back over again and machining the first face to ensure that the faces are flat and parallel to each other, always using the minimum amount of chucking pressure on the inner and outer diameters of the parts. As with any precision machining of steel parts, it was desired to perform an intermediate stress relief of the parts to normalize any machining-induced stresses along the way. However, heating of 17-4PH in the "A" condition runs the risk of starting the irreversible precipitation hardening process. On the other hand it is possible to perform a cryogenic stress relief where the steel is cooled to cryogenic temperature and held there a certain length of time, which for the parts the size and weights of the rotor and stator is about 15-18

hours. This procedure was performed by placing the rotor and stator on a level surface inside a large cryogenic container, then filling the container with liquid nitrogen and maintaining the level of liquid nitrogen throughout the process. The rotor and stator were held at liquid nitrogen temperature for about 24 hours. After the cryogenic process was complete, the rotor and stator were re-mounted to the machine and the rough machining process continued by finish-machining all non-critical surfaces to their final dimension and finishing all critical surfaces to within 0.010 inches of final dimension (see rough machining drawings, Figures 9 and 10). All details that required milling such as the pads and air feed grooves were also machined into the parts at this time. Critical surfaces are those such as the pad, journal, thrust and preload surfaces for the rotor, and the preload mount surface, bottom mounting pad surfaces, and the thrust and journal bearing surfaces for the stator. These are the surfaces that were to be finish machined on DTM-3. The preferred machining process selected for finish machining the parts based on the cutting test was a single point process using a cobalt machine tool on DTM-3, with a grinding process being the back-up choice. The manufacturing plans are presented in detail in Appendix 3.

1. ALL SHAPES ARE IN SHORT
2. SHAPES AND POLYMERIZATION PER
ANS 114.54-1972
3. SURFACE TREATMENT PER AMS 2481-1972
4. ABBREVIATIONS PER AMS 11.1-1972
5. BREAK SHAPES CODES 008-020.
6. 1/2" OR STR.
7. HEAT TREAT TO Rc 40/44
8. FOR FINISH MACHINE DIMENSIONS
SEE DRAWING ANGLE-10728

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WILL OTHERS SPOIL?

1. ALL DIMENSIONS ARE IN INCHES
2. SUGGESTED AND TOLERANCING PER ANSI Y14.5M-1993
3. SURFACE FINISH PER ASME B46.1-1978
4. APPROPRIATIONS PER ANSI Y14.1-1977.
5. BREAK SHOWN DIMS. 0.08" O.S.D.
6. 250° OR BTH.
7. HEAT TREAT TO RC 40-44

[illegible]

WILSON

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100

Figure 10. Table, Air Bearing Rotary, Stator Assembly Drawings

100

Figure 10b. Table, Air Bearing Rotary, Stator Assembly Drawings

Actual Manufacturing Procedure

Rough Machining

As mentioned above, all of the critical dimensions were to be finished to within 0.010 inches of final dimension during the rough machining stage. During the heat treatment of the rotor, however, it was observed that the inner and outer diameters of the rotor shrunk on the order of 0.005-0.006 inches. This was taken into account on the stator by taking the ratio of the shrinkage of the rotor to its diameter, then compensating on the inner and outer diameters of the stator by this ratio. This resulted in leaving less material on the inner diameter and more material on the outer diameter by the amount of 0.008 inches on average. This allowed for all stator dimensions being nominally 0.010 inches oversize after rough machining and heat treatment.

Another deviation occurred during the heat treatment / hardening procedure. Originally it was decided to harden the parts to 44-48 Rockwell C to aid the machining of the material. It was discovered later that some risk is involved in trying to harden 17-4PH to 48 Rockwell C, in the form of possible cracking of the material. This risk is not incurred during the hardening of 17-4PH to 40-44 Rockwell C. Hence in light of the small relative difference in hardness it was decided to harden the rotary table parts to 40-44 Rockwell C.

Final Machining

The final machining took place according to the plan as described above, starting with the rotor. The rotor was mounted to the DTM-3 faceplate, always on three pads and was machined using the Cobalt tool obtained from China. An LVDT was used while the part was still on the machine to get an approximate check on runout and flatness across the face, as well as the runout of the journal surface. However, circular flatness cannot be determined solely by inspecting the part on the machine using an LVDT. When the part appeared to be close to 20 microinches flatness across the face as well as about 30 microinches flatness on the journal, the part was removed from DTM-3 and was transported to the LODTM facility for inspection. The inspection revealed that the face had a circular flatness of about 100 microinches as opposed to the desired 20 microinches, and that a 120 microinch taper was machined into the journal surface. It was surmised that the taper was due to tool wear since there had been evidence of tool wear during the machining process and that the circular flatness problem was due to one of the three pads on the mounting surface not being flat, either due to dirt or to damage incurred during the flip-flopping of the part.

In an effort to solve these problems the pads on the rotor were lapped flat and the tool wear was characterized in order to compensate for the tool wear in the machining program. The rotor was re-mounted on DTM-3 and the journal and both face surfaces were

re-cut. Extreme effort was also made to dampen tool vibration during the cutting process by using copious amounts of duct seal around and between the rotor and the DTM-3 faceplate. An additional attempt was made to decrease part vibration by mounting a plastic cylinder-like oil flow damper to the rotor on the end away from the faceplate in order to enhance the flow of oil in-between the rotor and plastic cylinder. The tool wear problem observed on the previous machining attempt was more extreme than was observed during the cutting test and it appeared that the rotor had localized hard spots or bands where the hardness was different from the surrounding material. The best success cutting the journal surface occurred when the cut was performed with the tool moving from each end of the journal towards the center, making multiple passes at the same depth of cut.

The part was again removed from DTM-3 and was inspected, this time using a surface plate and an LVDT mounted on an air bearing spindle. The journal was found to be flat within 20-25 microinches, but the face was only flat to about 60 microinches. It appeared that the behavior of the rotor during the actual machining was not like that of the material during the cutting test, for reasons that were not understood. At this point the decision was made to revert to the back-up position of trying to grind the parts on DTM-3.

The grinder used for this application was one that incorporates an air bearing spindle and integral drive motor with a six-inch grinding wheel. The grinder was easily adapted to the tool post on DTM-3 and the rotor was again mounted on the three

pads to the faceplate of DTM-3. The pads, faces and journal surface were all ground by allowing the wheel to contact the surface, then allowing the wheel to spark out for multiple passes. Once again, an LVDT was used to check the face and journal surfaces with the part mounted on the machine. Before the part was removed for inspection, the LVDT indicated that the face and journal surfaces were flat to less than 20 microinches and that the journal was round to within 30 microinches.

The part was removed and inspected as before using the surface plate, LVDT and air bearing spindle. The inspection process yielded a non-roundness of 80 microinches on the journal which did not correlate well to the non-roundness of the journal of 30 microinches as checked on the machine. Now it was clear that something had changed from the time the rotor was bolted down on three pads with the axis of rotation in the horizontal position on the machine to the time the rotor was supported with the axis of rotation in the vertical position in the free state on a surface plate. It was suspected that the part could be sagging under its own weight as it was bolted to the face plate in the horizontal position. The part was then machined round and flat under these conditions and was found to be round and flat when inspected on the machine. But the machining was performed in a stressed state and went out of flat when the part was released and removed from the machine. Static analysis of the rotor determined that one could induce deformation into the part by bolting the rotor to the faceplate with the faceplate in the vertical position.

The final results were achieved by designing a cast iron faceplate that was removable in a repeatable fashion using a camlock adapter from DTM-3 for the purposes of mounting the rotor and stator to the faceplate in the horizontal position (axis of rotation vertical). The repeatability of about 100 microinches was required to minimize positional errors that could be introduced when the faceplate was removed to flip-flop or inspect the part being machined. The faceplate was then bolted to DTM-3 and the parts were ground to their final dimensions. The mounting pads on the bottom of the stator were lapped flat before the stator was mounted to the faceplate, then the journal bearing detail in the bronze insert mounted in the inner diameter of the stator was diamond turned before the stator thrust surface was ground, all in the same set-up.

Assembly Plan

The assembly of the motor and tachometer to the rotor and stator as well as the assembly of the rotor, stator and preload ring, already no easy task due to the large mass of and small clearances between the parts, is further complicated by the presence of the rare-earth neodymium magnets in the motor and tachometer stators. These magnets create an extremely strong magnetic field which can make positioning the steel components to any precise degree extremely difficult, even parts as massive as the rotor and stator. Because of this a series of fixtures was

designed that allows one to both grasp the parts and jack them into position, since the motor and tachometer windings attach to the rotor and the stators attach to the rotary table stator. These fixtures also allow for jacking the rotor down into the stator once the motor parts are attached. This procedure would also come into play any time that the rotor needs to be removed from the stator, such as for maintenance or repair.

Thermal Sensitivity Analysis

Drive Motor Thermal Loading

The rotary table will be subjected to thermal loading from the drive motor during normal operation. An analysis was necessary to determine the axial growth of the rotor due to this heat input since this would translate into a direct axial error. The analysis was done using the MAZE²⁶ mesh generator and TOFAZ2D²⁷ which is a two-dimensional heat transfer finite element code. The rotor was modelled as an axisymmetric steel ring with the heat input from the motor modelled along the bottom surface (see Figure 11). The following assumptions were made during the course of the analysis;

- 1) The ring convected heat from three surfaces.
- 2) The heat flux due to the drive motor was 1 Watt (0.0350 Btu/hr-in*2), based upon previous analysis and experimental data.

Figure 11. MAZE Finite Element Rotor Mesh

- 3) The heat flux was applied for two hours. This represents an azimuthal sweep operation of the machine which was assumed to be the worst possible real-life case, and in fact was later downscaled to 1 hour.
- 4) Ambient temperature was assumed to be 70.0 degrees F.

The thermally induced deformation was calculated as shown in Figure 11 by holding a node in the model fixed at the approximate location of the rotary table thrust bearing flange (which is node 60 in Figure 11) and then examining the growth in both directions from that point. Figure 12 shows a plot of temperature vs. time for nodes 51, 91 and 96. Based on this time history the growth in the "plus" axial direction (up) due to motor thermal effects is 6.2 microinches, and the growth in the "minus" direction (down) is 12.2 microinches. The deformation down does not influence part measurement accuracy since the axial position of the rotary table is determined primarily by the thrust and not the preload bearings and the part being measured is supported from the top surface of the rotor. The growth in the radial direction due to the motor thermal effects is about 6.9 microinches but because this deformation is symmetric about the rotary table axis it does not directly influence part measurement accuracy. Based on the simulation the predicted error due to growth of the rotary table in the axial direction is 3.1 microinches after one hour since the 6.2 microinches was due to a heat flux after two hours.

The results are conservative for two reasons. First, the model

Figure 12. TOPAZ2D Plot of Rotor Temperature vs. Time

neither includes nor accounts for the complete thermal mass of the rotary table (i.e. the bearing flange was not included in the model). Inclusion of the complete thermal mass would result in a longer time constant. And second, the thermal loading case considered in this model is not typical of a standard measurement cycle, as mentioned above. For the purposes of predicting the axial error due to the heat from the drive motor a realistic thermal loading would be a heat flux of less than 1.0 Watts over a period of 1 hour, in which case the axial growth will be less than 3.0 microinches. The actual value was measured to be less than 1.0 microinch as shown in Table 1.

A cooling jacket was designed into the motor / tachometer assembly as a conservative back-up for any unforeseen operational circumstances, because it was relatively simple to do and there was space in which to do it.

Environmental Temperature Effects

The environment of the COP gage will be temperature controlled to ± 0.1 degree F, peak to valley. The environmental temperature effects for the rotary table are those effects that are caused by the rotor thermal inertia responding to the maximum allowable ± 0.1 degree F temperature change. The thermal response of the rotor calculated in the previous finite element analysis based on a 1 Watt heat flux was used to predict a total axial error of 0.3 microinches due to a ± 0.1 degree F temperature change. The free

convection coefficient of air of 0.0069 Btu/hour-in*2 -degree F and the surface area of the rotor were used to calculate a scaling factor of 0.1 for the curves in Figure 12. The response of the rotor from Figure 12 after an hour, 0.2 degree F, was then scaled and multiplied by the thickness of the rotor and the coefficient of thermal expansion for steel to produce the result discussed above.

Dynamic Performance

The requirement for the first resonant mode of the COP gage of 80 Hertz or above as discussed previously is driven by the inspection time and data acquisition rate requirements as well as by the servo performance goals. The rotor and stator were modelled as a spring-mass system using the previously computed stiffness of 1.09E06 lbs/inch of the combined thrust and preload bearings along with the 180 lb mass of the rotor. The first resonant mode of the rotary table is calculated to occur at about 344 Hertz²⁸.

Position Feedback System

Rotary Encoder

The overall size and configuration of the rotary table components require the consideration of a somewhat different approach to the position feedback problem to meet the resolution and accuracy requirements in an economical fashion. The required

encoder resolution and accuracy requirements of 0.001 degree for one complete revolution, and 0.003 degree peak-to-valley, respectively, are fairly mild when one considers the encoder technology available today¹⁶⁻¹⁸. What provides the challenge in this case is finding an encoder that meets the requirements and is compatible with the geometry of the rotary table, specifically the large diameter rotor with the through-hole.

The first approach to the encoder problem considered driving an encoder directly off of the rotor or off the direct drive motor. The problem encountered here is that the only accessible area of the rotary table is the top surface, and this is where the parts and fixturing will be located. Also, most of these schemes involve various types of rotary-type encoders driven through some type of ratio device which could contribute positioning errors due to backlash when a change in the direction of the rotor is encountered, as well as inaccuracy from the imprecision of the drive mechanism.

The next encoder design considered was the annular glass scale type of encoder^{16,18} where a grating is machined into a glass annulus which in turn would be mounted to the top of the rotary table. The position of the scale relative to the ring is determined by read heads which are positioned on the stator. This would be satisfactory for the rotary table from an accuracy and positioning point of view but becomes less desirable when one considers the possibility of dragging and dropping 200 lb parts on top of the glass scale.

Another type of encoder similar in principle to the annular glass scale encoder is an encoder made for rotary encoder applications using reflective technology, where the grating of a known number of lines is machined into a reflective surface, in this case the top or side of the rotor. A read head then uses reflected light to read and interpolate the position of the grating. This product was desirable since it did not incorporate any glass or other fragile material that would be mounted to the top of the rotor, but it was more expensive than some of the other options.

The approach finally selected was an offshoot of a linear encoder system¹⁷ where a grating is fabricated into a self adhering flexible plastic/foil encoder tape for use with one readhead. For linear systems the tape and readhead are available to much higher resolutions than are required for the COP gage rotary table application - the system being used has a resolution of 40 microinches, which is adequate because the rotary table only requires about 120 microinches. The manufacturer¹⁷ is currently developing a encoder system for rotary applications based on this technology that is not yet on the market where the same tape is stretched around a disc and joined so that the gap and alignment of the tape at the junction does not affect the performance of the readhead. Although this system is not yet commercially available the vendor agreed to sell LLNL the tape and a readhead that has the capability to read a once-per-revolution fiducial mark, and also to provide documentation for the installation of the tape. The

installation of the tape on the side of the encoder ring was the responsibility of LLNL.

The tape was cut to a length nominally 0.005 inches larger than necessary to fit the encoder ring diameter. The ends were then laser welded together while observing the optical fringe pattern in the vicinity of the joint. Once the ends were joined together, the tape was stretched over the ring and lowered into position on the 0.003-inch locating step previously machined into the side of the encoder ring. A final check of the tape end alignment was performed using the encoder readhead after the tape was installed onto the ring.

Motor / Tachometer Requirements and Selection

The axis drive and velocity feedback requirements for the rotary table are for a direct drive D.C. torque motor and tachometer. The motor and tachometer were custom built²⁹ to provide a peak torque of 68 lb-ft, an operating range of 4-10 rpm, and a torque ripple of less than or equal to 5 percent. The motor and tachometer are geometrically identical to each other consisting of a stator ring which houses the magnets and a rotor ring which houses the windings, with the dimensions modified to reflect a 20.0 inch outer diameter, 17.5 inch inner diameter and a 1.30 inch height. The motor and tachometer stack into an assembly and bolt to the bottom of the rotary table with the magnet rings bolting to the stator and the winding rings bolting to the rotor. The assembly

details are contained in the rotary table assembly plan which will be formulated as a separate document.

Performance Testing and Results

Static Stiffness

The static radial and axial stiffnesses of the rotary table were measured by applying a 10 lb. force to the rotor in the appropriate direction and measuring the displacement of the rotor using an air bearing LVDT. The rotary table was resting on an inspection-quality granite surface plate. The displacement due to a 10 lb. force centered in the axial direction was measured to be 9 microinches at the point of application and at 30 psig journal and thrust bearing operating pressures resulting in an axial stiffness of $1.1\text{E}06$ lbf/inch. The radial displacement due to a 10 lb. force applied in the plane of the rotary table top surface was measured to be 34 microinches/lbf at 30 psig. This results in a radial stiffness of $2.9\text{E}05$ lbf/inch. The predicted static axial and radial stiffnesses were $1.0\text{E}06$ lbf/inch and $5.0\text{E}05$ lbf/inch, respectively.

The measured radial stiffness is about 40 percent below the predicted value and while this discrepancy is of little consequence to the performance of the rotary table a partial explanation of the lower than predicted value can be provided. During the assembly stage it was observed that the total radial gap between the rotor

and stator was about 800 microinches instead of the nominal design gap of 600 microinches. Also during the assembly stage, it became necessary to shim some of the preload ring/ air supply manifold components upwards about 0.060 inches which in turn raised the rotor about the same amount relative to the stator. This in turn reduced the height of one of the journal bearing lands by about 30 percent. When the journal bearing performance parameters are recalculated taking these modifications into account the predicted stiffness turns out to be about $3.5E05$ lbf/inch which is a 17 percent difference from the tested value. It should also be pointed out that increasing the operating pressure of the journal bearing increases the radial stiffness, by design. Operating the journal bearing at 45 psig raised the measured radial stiffness to $3.5E05$ lbf/inch.

Radial Error Motion

The radial error motion of the rotary table was measured in accordance with the ANSI/ASME B89.3.4M-1985 Axis of Rotation specification²⁴. A 1.0" diameter steel ball with a known error of 0.5 microinches was mounted to the top surface of the rotor and an air bearing LVDT was used to horizontally indicate the ball. The rotary table was again resting on an inspection-quality granite surface plate. The information from the LVDT was used to drive a chart recorder for one revolution of the rotary table. The information from the inspection chart was then used to create a

polar plot of the data from the one revolution, which is shown in Figure 13. The scaled average radial error from the plot shown is 2.5 microinches as determined by the prescribed "MRS" method²⁴. The predicted value for the average radial error from Table 1 was 3.5 microinches.

Axial Error Motion

The rotary table average axial error was measured using the same set-up as was used to measure the radial error in the previous case except that the LVDT was used to indicate the ball in the vertical direction. The resulting polar plot is shown in Figure 14. This error plot yields a scaled average axial error of 1.25 microinches, again based on the "MRS" methodology. The predicted value for average axial error from Table 1 was 3.5 microinches.

Tilt Error Motion

The same type of set-up that was used to measure the rotary table average radial error was used to measure the tilt error except that this time the ball was elevated 8.0 inches above the surface of the rotary table, using a riser at the table center. The ball was again indicated in the horizontal direction and the resulting error motion plot is shown in Figure 15. By again using the "MRS" method prescribed in the ANSI/ASME B89.3.4M-1985

Data from "R.T. Average Radial Error"

■ Column 2

Figure 13. Rotary Table Average Radial Error Plot

Data from "R. T. Average Axial Error"

■ Column 2

Figure 14. Rotary Table Average Axial Error Plot

Data from "Average Radial Error-Ball High"

■ Column 2

Figure 15. Rotary Table Average Radial Error Plot -
Ball at 8.0" Elevation

specification²⁴ and comparing this plot to the plot in Figure 13 the tilt error was determined to be less than 0.05 arcseconds. The predicted value from Table 1 was 0.08 arcseconds.

Drive Motor Thermal Effects

The objective of this test was to determine the deformation of the rotary table as a consequence of the maximum credible heat load from the drive motor. The rotational symmetry of the rotary table leads to minimal shift of the axis of rotation in the radial and tilt directions. The axial error motion of the rotary table is sensitive to the motor heat showing up as thermal growth of the rotor.

The set-up for this test involved setting up a 9.0 inch tall 1.031" diameter steel rod inside the through-hole of the rotary table, about 0.25 inches from the inside edge of the rotor and resting on the granite surface plate. A gauging stand was then placed inside the through-hole of the rotary table such that it overhung the steel rod and the top surface of the rotor. The gauging stand was also supported off the top surface of the granite. This gauging stand was used to support two capacitance gauges, one which gauged the three pads on the top surface of the rotor once per revolution and one which gauged the steel rod. The rotary table was then started from rest and allowed to accelerate to 10 rpm, where it was operated for the duration of the test which was one hour. The purpose of gauging the steel rod was to be able

to separate out the growth of the fixturing from the growth of the rotary table if the fixturing were to move around during the test. It turned out that the growth of the fixturing averaged less than 1 microinch during the one hour test period, and in fact the growth of the rotor for one hour at 10 rpm was below the reliable resolution of the test hardware, which was 1 microinch. The predicted value from Table 1 for the thermal growth of the rotary table after 1 hour of operation at 10 rpm was 3.0 microinches. The calculation that gave rise to this prediction was conservative, because the 2-D finite element model did not include the thermal mass of the bearing flange, nor did it account for the forced convection around the rotor due to the high pressure air exhausted by the thrust and journal bearings. Also, at the time the analysis was done during the design stage, the motor winding support can was in intimate contact with the lower mounting surface of the rotor. This was later revised to a three-point mounting scheme at this surface to eliminate any structural influence the motor support can could exert upon the rotor. The three-point mounting also reduced the contact area for motor heat conduction into the rotor.

Conclusions

The rotary table exhibited high performance as evidenced, for example, by the radial and axial errors measured at the table top of 2.5 microinches and 1.5 microinches, respectively. The conclusion drawn from reviewing the results is that the engineering

design methods employed in this project were successfully used to meet the objective of the design and manufacture of a rotary table to the required specifications while allowing predictions of the rotary table final performance during the design stage. The success of this methodology is borne out by the fact that the rotary table performs better than predicted in all of the performance criteria except for radial stiffness. The success of this methodology depends on having access to such engineering tools as an error budget to formulate the performance requirements, analytical calculation and modelling of the air bearings, analytical calculation of the rotary table dynamic characteristics, and thermal analysis by finite element method. This approach is quite distinct from traditional empirical methods typically applied to precision machine components.

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APPENDIX 1

COP Gage Y-Axis Rotary Table Requirements

C.O.P. GAGE

Y AXIS ROTARY TABLE DESIGN REQUIREMENTS

MAXIMUM AXIAL LOAD CAPACITY	1500 lbs.
WORKING LOAD CAPACITY	200 lbs.
MAXIMUM PART SIZE	18.0 inch outer diameter
THROUGH-HOLE DIAMETER	16.0 inches
AXIAL STIFFNESS	1,000,000 lbs./in.
RADIAL STIFFNESS	500,000 lbs./in.
ANGULAR STIFFNESS	67,000,000 lb.in.per radian
MAXIMUM RADIAL ERROR MOTION	4.0E-06 in.
MAXIMUM AXIAL ERROR MOTION	4.0E-06 in.
MAXIMUM TILT ERROR MOTION	0.1 arc-second.
MAXIMUM R.P.M.	10.0
POSITION FEEDBACK	Encoder
ENCODER RESOLUTION	0.001 deg./360 deg.
ENCODER ACCURACY	0.003 degree p-v
VELOCITY FEEDBACK	direct drive tach.
TEMPERATURE CONTROL	+/- 0.1 degree F, environment - closed cooling of drive motor (if required).
MAXIMUM OPERATIONAL AIR SUPPLY PRESSURE	80 psig
AXIS ORIENTATION	vertical
AXIS DRIVE	direct drive D.C. torque motor.

APPENDIX 2

Trust Bearing Calculations

The thrust and preload bearing calculations were performed in accordance with the following nomenclature:

A_i	The effective area of the bearing interior minus the area of the air feed groove.
A_a	The effective area of the bearing perimeter.
A_g	The cross-sectional area of the air feed groove.
B_{pi}	The average circumference of the bearing interior.
B_{pa}	The average circumference of the bearing perimeter.
C_g	Flow coefficient for a groove of square cross-section.
g_c	Acceleration of gravity.
H_1	Depth of step measured from the surface of the stator.
H_2	Clearance of bearing to the bearing lands, measured from the surface of the stator.
K	Bearing stiffness.
L^1	Width of one-half of the bearing interior, minus the width of the air-feed groove.
L_2	Width of the bearing land.
L_g	Average circumference of one air feed groove segment.
m_a	Mass flowrate through the perimeter of the bearing.
m_i	Mass flowrate through the interior of the bearing.
P_a	Absolute atmospheric pressure.
P_g	Gage pressure
P_i	Absolute pressure at the bearing interior.
P_a	Absolute pressure at the bearing land.
P_r	Alternate symbol for pressure at bearing interior.
P_s	Absolute supply pressure.
Q_a	Volumetric flowrate at the bearing land.
R	Gas constant in the perfect gas law.
T	Absolute temperature.

W_p	Bearing load capacity.
ρ	Gas density.
ν	Kinematic viscosity.

Reference UCRL-53643 Design of Externally Pressurized Gas Bearings for Dynamic Applications.

Roblee, J. W., p. 47 - 55, 105 - 107.

Thrust Bearing Calculations:

$$H_1 = 650 \mu\text{in} \quad L_1 = 0.5375 \text{ in}$$

$$H_2 = 350 \mu\text{in} \quad L_2 = 0.15 \text{ in}$$

$$P_s = 44.7 \text{ psig}$$

P_s , P_{a1} , and P_{a2} are known values. We need to equate

$$\dot{m}_{i1} \equiv \dot{m}_{a1}$$

and

$$\dot{m}_{i2} \equiv \dot{m}_{a2}$$

to calculate P_{i1} and P_{i2} using equation 4.3. To do this we need to calculate B_{pi1} , B_{pa1} , B_{pi2} and B_{pa2} (which are average circumferences) for use in equation 4.3. We also need to calculate A_{i1} , A_{a1} , A_{i2} and A_{a2} for use in equation 4.4.

So,

$$Bpi_1 = 2\pi \left(\frac{11.6 \text{ in} + 11.0625 \text{ in}}{2} \right) = 71.196 \text{ in}$$

Similarly,

$$Bpa_1 = 73.356 \text{ in}, Bpi_2 = 67.034 \text{ in}, Bpa_2 = 64.874 \text{ in}$$

and,

$$Ai_1 = \pi(11.6 \text{ in}^2 - 11.0625 \text{ in}^2) = 38.27 \text{ in}^2$$

Similarly,

$$Aa_1 = 11.00 \text{ in}^2, Ai_2 = 36.03 \text{ in}^2, Aa_2 = 9.73 \text{ in}^2$$

Now,

$$\dot{m}i_1 \equiv \dot{m}a_1,$$

therefore,

$$\frac{H_1^3}{24 \mu RT} \left(\frac{Bpi_1}{L_1} \right) (Ps^2 - Pi_1^2) = \frac{H_2^3}{24 \mu RT} \left(\frac{Bpa_1}{L_2} \right) (Pi_1^2 - Pa_1^2)$$

Solving for Pi_1 ,

$$Pi_1 = \left[\frac{\left(\frac{H_1^3 Bpi_1 Ps^2}{L_1} + \frac{H_2^3 Bpa_1 Pa_1^2}{L_2} \right)}{\left(\frac{Bpi_1 H_1^3}{L_1} + \frac{Bpa_1 H_2^3}{L_2} \right)} \right]$$

plugging in values,

$$Pi_1 = 36.695 \text{ psia.}$$

Similarly, setting $\dot{m}i_2 \equiv \dot{m}a_2$ and solving for Pi_2 ,

$$Pi_2 = 37.043 \text{ psia.}$$

Next, equation 4.4 is used to calculate the load capacity Wp . Note that the actual "area" term is substituted for the quantity " $BpLp$ ", in an effort to be more exact. Also, note that there are four components to the load capacity Wp in this case - one for each land and one for each half of the pocket.

The equation 4.4 terms are modified per the above graph for this particular design.

Outer Land:

$$Wp_{OL} = Bp \int_0^{Lp} (P(x) - Pa) dx = \frac{BpLp}{3} \left\{ \frac{2Pr^2 - Pr Pa - Pa^2}{Pr + Pa} \right\}$$

$$Wp_{OL} = \frac{Aa_1}{3} \left\{ \frac{2 P_{i_1}^2 - P_{i_1} P_{a_1} - P_{a_1}^2}{P_{i_1} + P_{a_1}} \right\} = 138.23 \text{ lb}$$

Outer Half Pocket:

$$Wp_{OP} = \frac{A i_1}{3} \left\{ \frac{2 P_s^2 - P_s P_{i_1} - P_{i_1}^2}{P_{i_1} + P_{a_1}} \right\} + \{ P_{i_1} - P_{a_1} \} A i_1$$

$$Wp_{OP} = 999.95 \text{ lbs}$$

Inner Land:

$$Wp_{IL} = \frac{Aa_2}{3} \left\{ \frac{2Pi_2^2 - Pi_2Pa_2 - Pa_2^2}{Pi_2 + Pa_2} \right\}$$

$$Wp_{IL} = 124.34 \text{ lbs}$$

Inner Half Pocket:

$$Wp_{IP} = \frac{Ai_2}{3} \left\{ \frac{2Ps^2 - PsPi_2 - Pi_2^2}{Pi_2 + Pa_2} \right\} + \{Pi_2 - Pa_2\}Ai_2$$

$$Wp_{IP} = 947.27 \text{ lbs}$$

$$\text{Total load capacity} = \Sigma Wp_{OP} + Wp_{OL} + Wp_{IP} + Wp_{IL}$$

$$Wp_{TOTAL} = 2209.79 \text{ lbs}$$

Flowrate Calculation:

Since $mi_1 \equiv ma_1 \equiv mi_2 \equiv ma_2$, we only need to calculate the flowrate at the inner and outer diameters or at the inner and outer pocket diameters.

Back to equation 4.3:

$$mi_1 = \frac{H_1^3}{24 \mu RT} \left(\frac{Bpi_1}{L_1} \right) (Ps^2 - Pi_1^2) = \frac{H_2^3}{24 \mu RT} \left(\frac{Bpa_1}{L_2} \right) (Pi_1^2 - Pa_1^2)$$

$$\dot{Qa} = \frac{\dot{m}}{\rho_*} \text{ and } \rho_* = \frac{Pa}{RT}$$

So,

$$\dot{Qa}_1 = \frac{H_2^3}{24 \mu Pa} \left(\frac{Bpa_1}{L_2} \right) (Pi_1^2 - Pa^2)$$

$$\dot{Qa}_1 = 25.64 \text{ in}^3/\text{sec} = 0.89 \text{ CFM}$$

Similarly,

$$\dot{Q}a_2 = 23.19 \text{ in}^3/\text{sec} = 0.81 \text{ CFM}$$

$$\text{Total } \dot{Q} = 1.7 \text{ CFM}$$

Calculation of Stiffness

To calculate stiffness, the scheme will be to vary H by a small amount, then calculate the change in Wp due to this change.

Stiffness is calculated as:

$$K = \left(\frac{\Delta Wp}{\Delta H} \right)$$

So,

Let $\Delta H = 0.000003 \text{ in}$. Then $H_1 = 653 \mu\text{in}$, $H_2 = 353 \mu\text{in}$.

Recalculation of Pi_1 and Pi_2 as done previously yields:

$$\begin{aligned} Pi_1 &= 36.628 \text{ psi} \\ Pi_2 &= 36.978 \text{ psi} \end{aligned}$$

Now, recalculating the load capacity Wp yields:

$$\begin{aligned} Wp_{OL} &= 137.78 \text{ lbs} \\ Wp_{OP} &= 998.75 \text{ lbs} \\ Wp_{IL} &= 123.96 \text{ lbs} \\ Wp_{IP} &= 946.17 \text{ lbs} \\ \Sigma Wp_{TOTAL} &= 2206.66 \text{ lbs} \end{aligned}$$

$$K = \frac{\Delta Wp}{\Delta H} = \left(\frac{2209.79 \text{ lbs} - 2206.66 \text{ lbs}}{0.000003 \text{ in}} \right)$$

$$K = 1.043 \text{ E } 06 \text{ lbs/in}$$

Calculation of Actual Load Capacity:

Since the quantity Wp is actually the preload required on the bearing to operate at the calculated parameters, it is necessary to calculate the real load capacity of the bearing. To do this, it is assumed that the gap is reduced by 1/2, so that $H_1 = 475 \mu\text{in}$, $H_2 = 175 \mu\text{in}$. We then need to calculate the new Wp , then the actual load capacity is Wp_{old} minus Wp_{new} .

Then, the new values are calculated as follows:

$$\begin{array}{ll}
 P_{i_1} & = 41.477 \text{ psia} \\
 P_{i_2} & = 41.650 \text{ psia} \\
 W P_{OL} & = 170.67 \text{ lbs} \\
 W P_{OP} & = 1087.20 \text{ lbs} \\
 W P_{IL} & = 152.01 \text{ lbs} \\
 W P_{IP} & = 1026.60 \text{ lbs} \\
 W P_{TOTAL} & = 2436.48 \text{ lbs}
 \end{array}$$

So, to collapse the bearing to 1/2 the flying height once the bearing is preloaded, the required load capacity is:

$$2436.48 \text{ lbs} - 2209.79 \text{ lbs} = 226.7 \text{ lbs}$$

Main Feed Groove Design:

The pressurized air lines to the rotary table are designed as 1/8" diameter lines. Hence, assume that the thrust bearing feed groove is 1/8 in. square in cross-section. Also, assume 8 feeds into the groove, and that the groove is segmented in-between feeds. The reason for the segmented groove is to increase the angular stiffness of the bearing. The pressure drop at the end of each segment is calculated using equation 6.7, as follows:

$$\Delta P_g = \frac{\pi \mu L_g}{N^2 C_g A_g^2} \left\{ \frac{RT \dot{m}_0}{P r_0} \right\}$$

Where,

$$N = 8, C_g = 0.88, A_g = (0.125 \text{ in})^2 = 0.01562 \text{ in}^2$$

$$L_g = \frac{2 \pi (11.0)}{8} = 8.640 \text{ in}, P r_0 = P_s = 44.7 \text{ psi}$$

$$R = 2.47 E 05 \text{ in}^2 / \text{sec}^2, \mu = 2.62 E - 09 \text{ lb sec} / \text{in}^2$$

$$g_c = 386.4 \frac{\text{sec} \text{ lbm}}{\text{lb f sec}^2}, \dot{m}_0 = \dot{m}_{i_1} = \frac{\dot{Q}_{i_1} P_{i_1}}{RT}$$

So,

$$\dot{m}_{i_1} \times g_c = 0.00278 \text{ lbm/sec}$$

and $\Delta P_g = 1.09 E - 04 \text{ psi}$ which is well below the $\Delta P_g \leq 0.01 P r_0$ criteria.

Preload Bearing Calculations:

$$\begin{aligned}
 H_1 &= 2400 \mu\text{in} & Bpi_1 &= \frac{2\pi(11.60 + 11.25)}{2} = 71.785 \\
 H_2 &= 400 \mu\text{in} & Bpa_1 &= 73.356 \text{ in} \\
 P_s &= 44.7 \text{ psia} & Bpi_2 &= 68.801 \text{ in} \\
 L_1 &= 0.350 \text{ in} & Bpa_2 &= 67.230 \text{ in} \\
 L_2 &= 0.150 \text{ in} \\
 Ai_1 &= \pi(11.60\text{in}^2 - 11.25\text{in}^2) = 25.125 \text{ in}^2 \\
 Aa_1 &= 11.003 \text{ in}^2 \\
 Ai_2 &= 24.080 \text{ in}^2 \\
 Aa_2 &= 10.084 \text{ in}^2
 \end{aligned}$$

Then, in the same manner as calculated for the thrust bearing,

$$\begin{aligned}
 Pi_1 &= 44.4818 \text{ psi} & \Sigma Wp &= 1837.636 \text{ lbs} \\
 Pi_2 &= 44.4913 \text{ psi}
 \end{aligned}$$

This is the actual load that will be supplied to the thrust bearing as preload. Assuming 400 to 450 lbs for the rotor, part and fixturing weight,

$$400 \text{ lbs} + 1838 \text{ lbs} = 2238 \text{ lbs}$$

The required preload for the thrust bearing was 2210 lbs.

continuing,

$$\dot{Q}_{TOTAL} = 3.972 \text{ SCFM}$$

$$STIFFNESS \quad K = \frac{\Delta W_p}{\Delta H} = 50,240 \text{ lbs/in}$$

Main Feed Groove Design:

Assume 8 feeds / 8 segments, for the same reasons as the thrust bearing. Also assume a 0.125 in square cross-section.

Then, $\Delta P_g = 3.12E - 04 \text{ psi}$, which meets the $\Delta P_g \leq 0.01 P_{r_o}$ criteria.

APPENDIX 3

Rotor and Stator Manufacturing Plans

COP GAGE ROTARY TABLE MANUFACTURING PLAN - ROTOR

NOTES:

Plate specifications to the vendor will specify a steel plate of 17-4PH stainless steel, 6.0 inches thick, 26.0 inches O.D., with a nominal 12.0 inch diameter hole cut thru the center of the plate, concentric to the O.D.

Rough and final machining will occur at LLNL. The part will be heat treated to 44 ROCKWELL C after rough machining. Finish machining will occur at LLNL with possible participation by RTH manufacturing personnel.

In the chucking procedures referred to throughout this plan, use only enough chucking pressure to keep the part from moving in the chuck during the machining process - DO NOT USE EXCESS CHUCKING PRESSURE AS THIS WILL DISTORT THE PART.

THE FINAL MACHINING WILL BE DONE AT LLNL ON DTM-3 AND WILL BE A SINGLE-POINT MACHINING PROCESS OR A GRINDING PROCESS, OR SOME COMBINATION OF THE TWO, DEPENDING ON THE SUCCESS OF THE INITIAL SINGLE-POINT MACHINING PROCESS.

- step 1 Mount the 17-4PH SSTL plate to the machine by chucking the plate on the O.D. Turn the I.D. to 0.1 inches on a side of the finished size using a carbide tool.
- step 2 Re-mount the part to the machine by chucking the plate on the I.D. with the top surface up. Turn the O.D. and all other accessible surfaces on the outside of the part to 0.1 inches of the finished dimensions.
- step 3 Flip-flop the part, again chucking on the I.D. and rough machine the remaining surfaces and outer diameters to within 0.1 inches of the finished dimensions.
- step 4 Stress-relieve the part using a cryogenic process where the part is cooled to approx. -320 degrees F for approx. 20 hours. NOTE: This will be done in house.
- step 5 Re-mount the part to the machine by chucking the plate on the O.D. with the top surface up and finish-machine the part I.D. to the final dimensions using a new carbide tool.
- step 6 Re-mount the part to the machine by chucking on the

I.D. with the top surface up and semi-finish machine all accessible outside surfaces and outer diameters to 0.01 inches of the finished dimensions.

- step 7 Finish machine the encoder ring mounting surface relief, preload bearing surface relief and thrust bearing surface relief to their final dimensions.
- step 8 Reverse the chucking orientation and semi-finish machine the remaining surfaces and outer diameters to within 0.01 inches of the final dimensions.
- step 9 Mount the semi-finished rotor to the table of a milling machine, top surface up, and mill the material in between the locating pads on the top surface of the rotor. Also mill the material between the locating pads on the encoder ring mounting surface.
- step 10 Re-mount the rotor upside down to the table of the milling machine and mill the material in-between the locating pads on the bottom surface. Remove the rotor from the milling machine.
- step 11 Locate, drill and tap the three fixturing holes on both the top and bottom surface of the rotor, as well as the encoder ring mounting surface.
- step 12 Heat treat the part to 44 ROCKWELL C using a vacuum furnace, with the part resting on a flat plate.

Note:

At this point, the rotor is ready for final machining.

- step 13 Mount faceplate and three brass pads to the rotor of the grinding machine (TBD) and turn the pads flat to the machine axis of rotation using a diamond tool.
- step 14 Cut the dust groove detail into the brass pads using a diamond tool and the same part set-up from the previous step.
- step 15 Change diamond tools and take a de-burring cut on the brass pad dust grooves using the same part set-up as before.

NOTE: Steps 16 and 17 pertain to a grinding process only. The word "machine" pertains to which ever method is selected for the final machining.

- step 16 Remove the carbide tool and tool post from the machine and install the air bearing grinding head, using the adapter plate.

- step 17 Dress the two surfaces of the grinding wheel parallel and perpendicular to the surface of the DTM-3 faceplate with the grinding head orientated at a 45 degree angle with respect to the faceplate.
- step 18 Mount the bottom surface of the semi-finished rotor to the faceplate on the three brass pads and bolt in place (rotor is right-side up on the faceplate).
- step 19 Machine the top surface locating pads.
- step 20 Flip-flop the rotor on the brass pads so that the rotor is mounted on the top surface locating pads (upside-down).
- step 21 Machine the bottom surface locating pads, the journal bearing surface and the thrust bearing surface to spec.
- step 22 Flip-flop the rotor on the brass pads so that the rotor is mounted on the bottom surface locating pads (right-side up).
- step 23 Machine the thrust/preload surface outer diameter, preload bearing surface, encoder ring mounting pads, and encoder ring mounting surface outer diameter to spec.
- step 24 Remove rotor from DTM-3 and inspect all surfaces.
- step 25 Re-mount part to faceplate and on to the machine and re-machine as necessary.
- step 26 Re-inspect rotor and record finished part information for use in the finish-machining of the stator.

COP GAGE ROTARY TABLE MANUFACTURING PLAN - STATOR

NOTES:

Plate specifications to the vendor will specify a steel plate of 17-4PH stainless steel, 6.0 inches thick, 33.0 inches O.D., with a nominal 16.0 inch diameter hole cut thru the center of the plate, concentric to the O.D.

Rough machining will occur at RTH with LLNL acting as consultants in the process. The part will be heat treated to 40-44 ROCKWELL C after rough machining. Finish machining will occur at LLNL with participation by RTH manufacturing personnel.

In the chucking procedures referred to throughout this plan, use only enough chucking pressure to keep the part from moving in the chuck during the machining process - DO NOT USE EXCESS CHUCKING PRESSURE AS THIS WILL DISTORT THE PART.

THE FINAL MACHINING WILL BE DONE AT LLNL ON DTM-3 AND WILL BE A SINGLE POINT MACHINING PROCESS OR A GRINDING PROCESS, OR SOME COMBINATION OF THE TWO, DEPENDING ON THE SUCCESS OF THE INITIAL SINGLE-POINT MACHINING PROCESS.

FOR THE STATOR, THE FOLLOWING SURFACES ARE REFERRED TO AS "CRITICAL SURFACES" IN THIS PLAN. ALL OTHER SURFACES ARE "NON-CRITICAL" :

1. Preload ring mount surface.
2. Thrust Bearing surface.
3. Journal Bearing surface - bronze.
4. 6 pad surfaces on bottom of stator.

- step 1 Using a 4 jaw chuck and a flip-flopping process similar to that used in manufacturing the rotor, rough-machine all I.D.'s, O.D.'s and steps to 0.100 inches of final dimension using a turning process.
- step 2 Stress-relieve the part using a cryogenic LN process where the part is cooled to approx. -320 degrees F for approx. 30 hours. NOTE: This will be done in house.
- step 3 Using a milling process, machine the 6 mounting surface pads, drill the lightening holes and the holes in the six mounting pads, and drill and tap the four 0.5" diameter fixturing holes.

- step 4 Using the six counterbored holes in the mounting pads for fixturing purposes, finish-machine all non-critical surfaces to their final dimensions (bore I.D., O.D., top flat O.D., 3 relief areas).
- step 5 Rough-machine the critical surfaces (thrust bearing surface, journal I.D., pads, preload ring mount surface) to 0.030 inches of final dimension.
- step 6 Drill all air supply / exhaust holes, drill and tap preload ring mounting holes, mill in the segmented air supply grooves and the journal air supply groove.
- step 7 Heat treat the part to 40-44 ROCKWELL C using a vacuum furnace, with the part resting on a flat plate.

Note:

At this point, rough-machining of the stator is complete, and the part is now ready for final grinding or single-point turning on DTM-3.

- step 8 Using the same machine tool or grinder set-up that was used on the rotor, mount the stator onto the faceplate of DTM-3 using the six mounting pads on the bottom surface of the stator.
- step 9 Turn on the pressurized air supply to the journal bearing, in order to keep dirt and dust from plugging the journal bearing feed holes.
- step 10 Machine the final step bearing detail into the face of the stator. Also machine the mounting surface for the preload bearing ring to the required height and flatness.
- step 11 Replace the carbide tool or grinding wheel with a diamond tool and machine the bronze journal surface to the final step bearing dimensions. Record the final diameter of the journal based on the machining of a pin and the machine coordinates of the final pass.
- step 12 Use an LVDT mounted in the tool holder to make charts of the journal roundness and face flatness.
- step 13 Remove the stator from the machine for inspection.

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