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LAPPING EQUIPMENT INSTRUMENTATION (U)

PDO 6984846, Topical Report

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Distribution August 1975

Prepared by J. L. Gray, D/812, under PDO 6984846

Instruments were developed for accurately measuring the mechanical power input to the surface of a spherical part being lapped to a superfinish. Slip rings were developed to remove the torque signal from the rotating shaft. Accurate gaging equipment for applied pneumatic pressure and lap machine shaft rotation rate was installed. Torque measurements accurate to ± 0.03 inch-ounce (± 0.212 mN·m), pressures to ± 0.01 psig (± 69 Pa), and rotation rates to ± 3 rpm (± 0.3 rad/sec) are continuously measured with the system.

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SUMMARY

Instruments were developed for measuring input lap torque (and input power), lap loading pressure, and lap rotation rate during single-part three-point lapping of spheres.

A torque transducer was designed, built, and tested for measuring the torque supplied to the sphere during lapping. The torque supplied by the driving motor was transmitted through a flexible beam to the lap that turned against the part surface. The free-end cantilever beam was gaged with semiconductor strain gages. The power supply, slip rings, and monitoring electronics were designed and built to exploit the advantages of the gages used. An analog computer module allowed the torque value to be multiplied by the rotation rate, giving the energy input to the spherical surface as an analog function of time.

The rotation rate was measured with a Servo-Tek linear dc generator, with an output which is a linear function of rotation rate, accurate to ± 3 rpm (± 0.3 rad/sec).

The lap loading pressure (the pneumatic gage pressure applied to the manifold supplying the lap pistons) was measured with a Kistler electronic balance gage accurate to ± 0.01 psig (± 69 Pa).

The levels of precision in measurement that were established appear to be within the limits required for process control.

DISCUSSION

SCOPE AND PURPOSE

A controlled semi-automated lapping process is needed for fabricating superfinished spheres of various sizes, metals, and alloys. Instrumentation was developed for measuring input lapping torque, lap pressure, and lap rotation rate during single part or three-point lapping of spheres. Accurate monitoring and control of lapping variables is necessary for achieving a controlled spherical lapping process to fabricate superfinished spheres.

PRIOR WORK

Previous work at Bendix dealt with an analysis of lapping equipment for polishing spheres. Surface defects which occurred during lapping of metallic spheres were investigated, and theories were advanced to explain their possible origins. However, no critical control parameters were identified, and no work was done to improve accuracy in monitoring and controlling lapping variables.

ACTIVITY

Three-point lapping is a technique used by Bendix to polish metal spheres to exceedingly fine surface finishes. The basic principles of operation of the three-point lap machine can be understood with reference to the schematic diagram in Figure 1.

Each lap shaft is driven by an independent speed-controlled motor and gear train. Radial force is applied along the lap axes by air pistons fed through the centers of the rotating shafts. In normal operation, the laps rotate in the directions indicated in the diagram. The left lap is rotated intermittently to give the sphere random motion. Normal lap speeds are 180 rpm (18.8 rad/sec) for both rear laps and 15 to 20 rpm (1.6 to 2.1 rad/sec) for the center lap. Normal lap loading pressures range between 5 and 15 psig (34.5 and 103.4 kPa). Major radial force changes are accomplished by changing the size of the pistons applying the force.

A recent study dealt with the effect of lap pressure and torque on surface finish. The present study describes development and implementation of instrumentation for continuous and accurate measurement of lap pressure, torque, and lap rotation rate during three-point lapping.

Figure 1. Schematic Diagram of a Three-Point Lapping Machine, Viewed From the Top

Torque Transducer System

Torque levels determined in previous three-point lapping work indicated the need for a sensitive torque transducer. The transducer developed employed a strain-gaged cantilever beam of heat-treated beryllium copper alloy number 172, Condition H, heated at 600°F (315°C) for 2 hours. The laboratory model of the lap head instrumentation for measuring torque applied to the ball during three-point lapping is shown in Figure 2. As drive plate A is rotated about the horizontal axis $h-h'$ by the motor drive, the drive dogs D in turn rotate plate B about the axis $h-h'$. The cylinder in which the lap head is mounted is free to rotate relative to B about axis $h-h'$ on a pair of thin precision instrument bearings. The beam is attached rigidly to plate B. As B rotates, the end of the beam engages the cylinder in slot S, causing the cylinder and lap head to turn with B. The torque that drives the lap shows up as beam deflection. The only error signals result from the drag in the ball bearing suspension system and the inertia of the lap carrier. The bearings were chosen to keep the drag friction error small, however.

Figure 2. Diagram of Lap Head System, Instrumented to Obtain Torque Measurements

Strain Gage and Wheatstone Bridge

Initial gaging of strain was accomplished with BLH SR-4 foil-type gages and associated electronics. The cumulative effects of slip ring noise, electronic noise, and electronic pickup demanded more signal than the SR-4 gages provided. To obtain the necessary signal level, semiconductor strain gages, type SPB2-35-500 (BLH Electronics, Inc.), with nominal resistance of 5000 ohms and nominal gage factor of +150, were used. The individual calibration charts supplied with the gages indicated an actual gage factor of +157 and an actual resistance of 4865 ohms.

Operating the gages at room temperature permitted the use of these semiconductor gages which are highly strain-sensitive, but moderately temperature-sensitive. To assure that no thermal effects could enter, a thermally self-compensating Wheatstone bridge arrangement was used (Figure 3). The two gages were

Figure 3. Wheatstone Bridge Arrangement

Figure 4. Gage Arrangement on the Free-Ended Cantilever Beam Used as the Torque Transducer

mounted on opposite sides of the same beam (Figure 4), and were, therefore, subjected to the same temperature. The effects of temperature cancel out because the incremental changes in resistance, caused by temperature changes, are the same in adjacent arms of the Wheatstone bridge. Because the thermal resistance shift is the same amount, and in the same direction in each arm, no bridge imbalance is produced.

The voltage E_0 expected from the typical Wheatstone bridge circuit is

$$E_0 = \frac{FE}{-4} (-\epsilon_1 + \epsilon_2 - \epsilon_3 + \epsilon_4),$$

where

ϵ_i = strain i ,

ϵ = zero for dummy gage or resistor,

E is the supply voltage, and

F is the gage factor.

For identical gages on two adjacent arms, and fixed resistances on the other two,

$$E_0 = \frac{FE}{-4} (-\epsilon_1 + \epsilon_2),$$

where

ϵ_2 = strain of gage $G_1 = -\epsilon_1$. Thus,

$$E_0 = \frac{FE}{-4} [-\epsilon_1 + (-\epsilon_1)] = \frac{\epsilon_1 F}{2} E, \text{ and}$$

$$E_0 = \frac{(\epsilon_1 E)}{(2)} F. \tag{1}$$

For this particular system, the supply voltage was 30 volts, and the gage factor F was 157. Therefore, from Equation 1 a range in microstrain from zero to 3000 corresponded to an output voltage range of zero to 7 volts. The measured torque output after 10X multiplication at the monitor input stage was approximately 500 millivolts per inch-ounce (71 V/N·m), indicating a real signal of 50 millivolts per inch-ounce (7.1 V/N·m).

Slip Ring Assembly

The two functions of the slip ring assembly were to transmit power to the strain gage circuit, rotating with the lap head, and to transmit Wheatstone bridge imbalance voltages, corresponding to applied torque levels, from the rotating head to the stationary monitor signal input terminals. Fabrication of the slip ring assembly (Appendix A) was dictated by lack of an end-of-shaft on which to mount a commercial unit.

The slip ring assembly consisted of two main components (Figure 2): the ring assembly, and the brush assembly. The ring assembly accepted input power from the brushes to drive the gage circuit and passed bridge imbalance voltages to the brushes leading to the monitor signal outputs. The brush assembly made the electrical contacts between the slip rings and the external circuitry. The slip rings were made of brass. Each ring was 0.300 inch thick (7.62 mm) and 0.150 inch wide (3.8 mm) (annulus). Six concentric rings were made in increments of 0.400 inch inside diameter (10.2 mm). They were rhodium-plated, wired, and molded into a lucite plate, using a specially designed mold.

The brush holder assembly, which positioned and held the high-silver-content carbon (Helwig 7025 grade) brushes in contact with the slip rings, was machined from Plexiglas. Set screws were used to tension the brush springs properly. The brush holder was held fixed with respect to the rotation of the drive shaft by a cable latching it to a stationary support. Two high quality instrument roller bearings seated in the brush holder plate with outer races flush supported it on the rotating shaft. The inside races, which rotated with the drive shaft, extended 0.062 inch (1.6 mm) beyond the brush holder surface, preventing contact between the brush plate and the ring plate when the two were pressed together against the brush spring pressure and locked in place with a lock ring.

Calibration of the Torque Transducer System

Figure 5 depicts the torque transmitting beam and the aluminum cylinder that is free to turn on precision bearings until engagement by the end of the strain-gaged beam. The torque transmitted is proportional to the beam strain as registered by the strain gages. The signals generated by the torque transducer were recorded as a function of the applied torque under both static and dynamic conditions. All torque signals were integrated over 1-second intervals with an HP-2041C digital integrating voltmeter (DVM). The applied torque was varied by varying the mass suspended on a known radius arm. The masses were loaded into a 25.07-gram bucket suspended by a thin nylon line wrapped around the cylinder (Figure 5). The electrical output voltage obtained with the integrating DVM was plotted versus the load mass in

Figure 5. Experimental Arrangement Used in Calibrating the Torque Measuring System

grams for the static case in which the drive shaft was not rotated. The results are illustrated in Figure 6.

To obtain the signal-versus-torque data for the dynamic case, in which the drive shaft was rotated at operating speed (Figure 6), the bucket was pulled up at a uniform velocity with its load of weights. The torque should theoretically have been exactly the same as in the static case because there were no accelerations involved. However, the two sets of calibration data diverge slowly with added mass in the bucket. The divergence is not likely to be associated with velocity changes (accelerations) because of the massive motor and gear box. Two possible sources of this divergence are roller bearing drag or oscillatory contributions to the integrated voltage value resulting from more pronounced oscillation of the beam with added mass. However, either curve serves as an acceptable comparison of signal to torque. Because the dynamic curve represents the actual lapping situation, it was chosen as the calibration curve for the torque.

Figure 6. Calibration Chart Obtained With Experimental Arrangement of Figure 5

Figure 7 presents dynamic and static calibration curves obtained in the manner just discussed, except that the suspension line was wound around the lap itself, rather than around the larger-diameter aluminum cylinder. The same divergence of the static and dynamic curves is evident, as well as the same degree of linearity of each of the curves.

The relationship between the calibrations illustrated in Figures 6 and 7 is

$$r_1 F_1 = r_2 F_2,$$

where

r_1 = aluminum cylinder, 0.707-inch radius (18 mm),

r_2 = lap radius of 0.264 inch (6.7 mm),

F_1 = bucket load, and

F_2 = load at r_2 that will produce the same strain in the beam as the load F_1 at radius r_1 ;

or

$$F_2 = \frac{(r_1)}{r_2} F_1,$$

$$F_2 = \frac{(0.707 \text{ inch})}{0.264 \text{ inch}} (F_1) = 2.68 F_1;$$

or, if $F_1 = 100 \text{ g}$, $F_2 = 267.8 \text{ g}$.

Using data from Figure 6, the output was 1.21 volts at $F_1 = 100 \text{ g}$. Using data from Figure 7, the output was determined at 267.8 g to be 1.24 volts. This voltage displacement between F_1 and F_2 of 0.03 volt is equivalent to 0.0304 inch-ounce (0.215 mN·m) on the scale of Figure 7.

Calibrations made on a fixed radius arm showed good repeatability. Calibrations made on radius arms of different sizes did not compare as well. Because the latter method provided the least accurate results, the torque from the voltage output was known within ± 0.03 inch-ounce ($\pm 0.212 \text{ mN}\cdot\text{m}$). Included in these calibrations were the noise contributions resulting from slip ring motion and simple motion effects on the transducer; but, from the calibration results, these effects were concluded to be small.

Figure 7. Calibration Chart Obtained With Line (Figure 5)
Wound Around Lap

An HP-7701B strip chart recorder was used to display the torque transducer output at zero applied torque and at various rotation rates. At zero rotation rate, there was no apparent width to the line other than the recording device pin contact width, indicating that no significant environmental electrical disturbances existed. When the output was displayed on an oscilloscope, approximately 22 millivolts of 60-cycle ac signal pickup noise were incurred at the exposed leads connecting the four contacts of each slip ring. The 22-millivolt signal did not appear on the chart recorder because the response of the recorder was not fast enough to follow the sinusoidal variation. Therefore, it printed the average value of the input which is zero. The upper limit of frequency for accurate recorder response was approximately 30 cps, falling off past that point to one-fifth amplitude presentation of 60 cps. That is, on a more sensitive scale, the chart recorder read 4 millivolts, while the oscilloscope read 22 millivolts at 60 cps.

The torque output was recorded for rotation rates of 45, 90, and 180 rpm (4.7, 9.4, and 18.8 rad/sec) when the lap was not in contact with a part and no torque load was applied. The torque head unit was clamped to the drive dogs to stabilize the unit as it rotated. In each case, there was a rotational contribution to the torque output at the lap rotation frequency. That contribution was the net result of all pickup noise, all slip ring noise, all gravity-initiated stresses in the beam, and all pivoting force fluctuations in the roller bearing-cylinder system. The amplitude of these periodic contributions diminished at the nominal rotation rate of 180 rpm. At 180 rpm the noise signal was well below the average signal levels expected, so that the signal-to-noise ratio was approximately 1.000 volt to 0.010 volt, or 100 to 1.

It should be noted that the signals under discussion all had a 10X multiplication within the monitor, and that oscilloscope signal tracing showed no detectable noise contribution from the monitor.

Scale Changing and Sensitivity Within Each Range

The full range of strain levels for which the gages were designed was employed on the working system. The range of measurable torque values [zero to 8.0 inch-ounces (zero to 56.5 mN·m)] for the working system can be halved; that is, the sensitivity can be doubled, merely by replacing the two passive resistors (R_1 and R_2 in Figure 3) in the Wheatstone bridge with active gages mounted in parallel to those already on the beam. The rule governing the bridge imbalance states that the bridge is unbalanced in proportion to the difference in the strains of gages located in adjacent arms and in proportion to the sum of the strains of gages located

in opposite arms. In this manner, the voltage imbalance in the bridge can be doubled for a given strain. The arrangement used would be as shown in Figure 8.

De-sensitizing the device (that is, spreading the scales) can be accomplished by simply placing another beam at 180 degrees from the gaged beam, or by changing the thickness of the existing beam. The second beam can be chosen in such a way that its thickness will control the total deflection of the gaged beam to whatever level is desired. It is necessary to calibrate each scale independently.

Transducers for Pressure and Rotation Rate

Pressure Gage

The Kistler Model 311D differential pressure gage, that replaced the conventional Bourdon tube gage normally used, was an electronic force-balancing transducer for which the manufacturer claimed repeatability of 0.02 percent of full scale [0.0002·15 psig = 0.003 psig (20.7 Pa)]. That is the result of a resolution claim of 0.01 millibar (1 Pa). The output from this transducer varied from 0.050 to 10 volts, depending on the pressure.

Rotation Rate

A tachometer generator, Servo-Tek Model SU-780D-1, designed for use with fast-response recording equipment and rated at 45 volts per 1000 rpm was used as a rotation rate transducer. A viton O-ring was used to drive a 1.000-inch-diameter (25.4 mm) pulley attached to the generator drive shaft. A pulley with the same diameter was fixed to the free end of the gear box shaft. The negative dc output voltage calibrated in rpm or rad/sec was displayed on the monitor digital display.

The output signal linearity at any speed from zero to 12000 rpm, claimed by the manufacturer, was better than 0.1 percent of the output at 3600 rpm. The uncertainty in rotation rate was

$$\text{Uncertainty} = \pm 0.001 \cdot 3600 \text{ rpm} \cdot \frac{45 \text{ volts}}{1000 \text{ rpm}} = \pm 0.162 \text{ volts} \\ = \pm 3.6 \text{ rpm}$$

The fluctuation of ± 3.6 rpm corresponded with the digital voltage variations observed in the measurement of 180 rpm. However, the fluctuations decreased with decreasing rpm, indicating the percent fluctuation remained relatively constant.

Figure 8A. Gage Mounting Arrangement to Double the Signal Output of the Cantilever Beam Transducer

Figure 8B. Wheatstone Bridge Circuit for Four Active Strain Gages

Monitor System

The torque, pressure, and rotation rate transducers were connected to the monitor and the strip charts as shown in Figure 9. The system was designed to accept three input signals and to provide analog outputs to recorders. All dc voltages fed into the monitor were negative, and in each case the positive side was grounded. This was done to provide the negative inputs required by the Intronic Model M311 precision analog multiplier-divider used to generate the analog output.

The monitor accepted input dc-voltage signals T_1 (torque), T_2 (gage pressure), and T_3 (rotation rate) from the transducers. T_1 ranged from zero to 180 millivolts, and T_2 and T_3 ranged from zero to 10 volts. Each input channel contained an electronic circuit which both matched the input impedance to the source and amplified the signal an appropriate amount to drive a recording device. Each plug-in input circuit card also provided calibration adjustment for its channel. The designs of the input cards which tailored and calibrated the transducer signals are given in Appendix C.

A Newport DPM No. 200A digital meter, with a range of ± 1.999 volts, was used to monitor the transducer signals at the output side of the input circuit card. The meter was connected to the appropriate card by a switching arrangement. The meter was connected automatically to the output of the analog module by this same switching arrangement when the analog function switch was closed.

The switches used to activate the meter to monitor the desired signal were Newark Electronics 29F2303 four-gaged DTDT lighted push-button switches with interlocks.

The X, Y, Z, and function columns were each comprised of a gang of four interlocked double-pole/double-throw switches. Pushing any one button in the set released the other three. The switch panel use is tabulated in Table C-1 of Appendix C.

With this system, the dc outputs from any two transducers were used to produce a continuous analog function. As long as the transducer outputs were linear with respect to the parameter changes they measured, the analog products or quotients remained valid. For example, energy input measure was the analog product of torque and rotation rate, or

$$E = T_1 \times T_3.$$

Figure 9. Block Diagram Measuring System for Torque, Rotation Rate, and Gage Pressure

Measured Torque Under Lapping Conditions

The torque transducer signals during lapping of a 0.62-inch-diameter (15.7 mm) copper sphere were displayed on the strip chart for various combinations of lap pressure and rotation rate in two modes of operation. In the first mode, the left rear lap was motionless and the others operated nominally at 180 rpm for right rear head and 18 rpm for center head. In the second mode, the left rear head rotated intermittently so that a normal run mode resulted. This second mode was considered the normal operating condition in which the left rear head rotated intermittently at 180 rpm (on 1.8 seconds, off 0.7 seconds), the center head at a constant 30 rpm, and the right rear head at a continuous 180 rpm. The part rotated about a horizontal axis perpendicular to the center shaft axis when the left rear head was on, and about the center head axis when the left rear head was off.

The torque transducer outputs for operation in the first mode at three different rotation rates and three different pressures showed periodicity varying in frequency and amplitude with rotation rate. At higher rotation rates, the amplitude of torque oscillation diminished. The increased pressures, which were applied to both rear heads by a common manifold, produced a proportional but nonlinear net displacement of the oscillating torque signal.

The torque transducer outputs for the second mode of operation (the normal mode) showed considerable torque fluctuations occurring between the *on* cycle and the *off* cycle. The torque decreased during the *on* cycle of the left rear head. In addition, the torque in the normal lapping mode varied considerably with applied pressure and as a function of the lap rotation rate.

Figure 10 shows an XY plot of torque versus pressure in the normal mode of operation. The torque increased with pressure in an approximately linear manner to a maximum value, where increased pressure only resulted in erratic torques. Curve C in Figure 10 represents the torque-versus-pressure trace obtained during the return to zero pressure. The apparent hysteresis effect might have been caused by exclusion of abrasive particles between the lap face and the part as the pressure was increased. Abrasive re-entry was permitted only in the torque-pressure region where the descent curve merged with the ascent curve. In precision lapping operations, it may be inadvisable to lap at torque-pressure points beyond the intersection of the ascent and descent curves.

The shape and character of the torque signal changed as the part was displaced from the ideal center defined by the intersection of the three lap axes. That change in signal shape was used to

Figure 10. Lapping Torque Versus Applied Pressure

aid in centering the part in the lap machine. Tests showed the method to be at least as accurate as visual alignment.

ACCOMPLISHMENTS

A transducer for measuring torque exerted on the sphere by the lap head during three-point lapping was designed, built, and tested. Torques in the range of zero to 8 inch-ounces (56.6 mN·m) were measured to within ± 0.03 inch-ounce (± 0.212 mN·m) with the torque transducer.

An electronic monitoring system was designed and built to record the torque, pressure, and rotation rate during the three-point lapping. An analog function module was included which allowed multiplication or division of one or more of the input signals.

FUTURE WORK

Simplification of the torque system with the object of eliminating the use of ball bearings will be investigated. The use of a pneumatic controller and actuator for varying the air pressure to control the input torque at a desired value during the three-point lapping will be investigated.

Appendix A

FABRICATION DETAILS OF THE SLIP RING
ASSEMBLY

Appendix A

FABRICATION DETAILS OF THE SLIP RING ASSEMBLY

FABRICATION OF THE SLIP RINGS

Brass plate was machined into concentric annuli (Figure A-1), all 0.1875 inch thick (4.8 mm) having annular width of 0.150 inch (3.8 mm) and inside radii incrementing by 0.200 inch (5.08 mm) from the smallest ring size of 0.750 inch (19.1 mm) inside radius. A brass bushing with an inside diameter of 1.001 inches (25.4 mm), an outside diameter of 1.123 inches (28.5 mm), and thickness of 0.500 inch (12.7 mm) was made to seat on the drive shaft to hold the ring plate.

Two holes 0.031 inch in diameter (0.8 mm) were drilled on one side of each ring to provide wiring sockets into which the lead wires, two per ring, were soldered, using appropriate lengths of 22-gage color-coded stranded wire.

MOLDING THE RING PLATE AND RHODIUM-PLATING THE RINGS

A mold (Figure A-2) was made to hold the brass rings in their proper relative positions during their encapsulation on three sides by a transparent epoxy. The six brass rings were fitted loosely into the six corresponding grooves in the aluminum mold which was sprayed with a mold releasing agent just prior to the potting operation. The brass bushing was inserted in the center to act as a centering control to locate the axes of the rings precisely on the axis of the rotating shaft.

A slotted aluminum ring 1 inch high (25.4 mm) was clamped around the outside of the aluminum mold with a threaded hose clamp which had a large outside diameter. Mold release was applied to the inside surface. The wires soldered to the rings were flattened below the surface level defined by the top of the aluminum clamp ring. The wires were then fed out through the 0.030-inch-wide (0.8 mm) slot and taped together.

Epon 828 epoxy was then mixed with U curing agent, and the assembly cavity was filled. The epoxy was allowed to cure at room temperature for 24 hours before the assembly was lifted from the mold.

After the ring assembly was thoroughly cured, standard metallurgical polishing procedures were used to lap all the exposed brass rings to form a uniplanar configuration. Each ring

Figure A-2. Brush Holder Plate

protruded 0.020 inch (0.5 mm) above the epoxy surface. Total material removal in this process was less than 0.002 inch (50 μ m) as they were very nearly uniplanar on removal from the mold.

The final operation was to generate a uniform matte background of very shallow scratches on the exposed brass surfaces, using 600-grit paper lubricated with water. This was followed by plating the exposed brass with 0.002 inch of rhodium.

The finished ring plate is shown in cross section in Figure A-1.

FABRICATION OF THE BRUSH HOLDER ASSEMBLY

The brush holder was fabricated from a lucite plate to the specifications in Figure A-2. Study of precision slip ring systems available in the open market indicated the best units used two brushes per rhodium-plated slip ring. To assure that the system would operate safely, four brushes were used.

The holes drilled in the plate (Figure A-2) were fabricated to allow a 0.125-inch-diameter (3.2 mm) brush to slip without catching so that the brush spring could control the position of the brush. The brass hexagonal head bolt was threaded into the back of the plate to serve as an electrical contact and as a brush spring compression adjustment. The mound of solder on the end of the hexagonal head allowed easy electrical connection between the four brushes contacting each ring, and to the external circuitry. The springs on the brushes governed the brush pressures once the hexagonal head screws were adjusted, and the gap between the plates was fixed. The brushes chosen were manufactured of high silver content carbon. They were 0.375 inch long (9.53 mm) (Helwig No. 7025 grade) with attached leads, springs, and caps.

The brush holder shown in Figure A-2 was held fixed, with respect to the angular rotation of the shaft, by a cable latching it to a support on the equipment base. To support it on the rotating shaft, two high-quality instrument roller bearings were employed. They were double-shielded, extra-thin section instrument bearings, MPB No. 3TAR17-21U, with a 0.875-inch inside diameter (22.2 mm), and a 1.3125-inch outside diameter (33.3 mm). The bearings were seated in the brush holder plate so that the outside races were flush with the outside surfaces of the plate when pressed in place. The inside races, which rotated with the drive shaft, extended 0.062 inch (1.6 mm) beyond these surfaces, permitting the lock ring to be firmly pressed home on the inside race without the plates contacting each other.

Appendix B

STRAIN GAGE INSTALLATION

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STRAIN GAGE INSTALLATION

The gages were mounted on opposite sides of the beam (Figure B-1), using standard strain-gage mounting techniques and Epon 828 epoxy with Type A hardener. Epon 828 was selected to provide an acceptable degree of rigidity in the installation while remaining flexible enough to refrain from fracturing within the range of strains anticipated.

The gages were mounted to the ground beam surfaces by applying liberal amounts of adhesive to both of the mating surfaces. Because the adhesive sets in approximately 4 hours, there was ample time to work out any entrapped air bubbles, and to align the gages properly with scribe lines to within 0.010 inch (0.25 mm). The positioned gages were then held in place under compression between RTV sheets. The resulting mounted gages showed no traces of delaminations or entrapped air bubbles, and there were no electrical ground paths through the backing.

Figure B-1. Placement of the Strain Gages on the Cantilever Beam

Appendix C

MONITOR SYSTEM CIRCUITRY

Appendix C

MONITOR SYSTEM CIRCUITRY

The circuits for conditioning the torque, pressure, and rotation rate signals are shown in Figure C-1. The selector circuitry for readout of torque, pressure, rotation rate, and products such as torque times rotation rate (mechanical power) is diagrammed in Figure C-2. Figure C-3 and Table C-1 give the arrangement of the monitor pushbutton switches and show the different variables or combinations of variables which can be read digitally by depressing the buttons.

Figure C-1. Diagram of Transducer Signal Conditioning Circuits

NOTE - All inputs (T1, T2, T3) must be negative.
 - Any operation involving T3, units are actual.
 - Any operation not involving T3, units are 100 times actual.

Figure C-2. Torque, Rotation Rate, and Pressure Function Selector Circuit Diagram

Figure C-3. Arrangement of Pushbuttons on the Monitor Which Allows Selection of Signal to be Observed

Table C-1. Switch Function for Monitor System

Displayed on Meter	Function Switch Pushed In	Column With Button Pushed In	Button Pushed In
T ₁ (torque)	Direct Read (Meter Read)	X	T ₁ Read,
T ₂ (pressure)	Direct Read (Meter Read)	Y	T ₂ Read,
T ₃ (rotation rate)	Direct Read (Meter Read)	Z	T ₃ Read,
XY	XY	X	T ₁ or T ₂ or T ₃
		Y	T ₁ or T ₂ or T ₃
Z/X	Z/X	Z	T ₁ or T ₂ or T ₃
		X	T ₁ or T ₂ or T ₃

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