22
11-12-74
copy NT IS

BDX-613-879 (Rev.)

TRANSDUCER FOR GAGING
PRESSURE-SENSITIVE PARTS

PDO 6984674, Final Report

W. G. Eby, Project Leader

Project Team:
C. W. Berry
J. C. Bolinger

November 1974

MASTER

Prepared for the
United States Atomic Energy Commission
Under Contract Number AT(29-1)-613 USAEC

**Kansas City
Division**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Printed in the United States of America

Available From the National Technical Information Service, U.S. Department of Commerce, 5285 Port Royal Road, Springfield, Virginia 22151.

Price:	Microfiche	\$2.25
	1- to 50-Page Paper Copy	\$4.00
	51- to 150-Page Paper Copy	\$5.45
	151- to 325-Page Paper Copy	\$7.60

TRANSDUCER FOR GAGING
PRESSURE-SENSITIVE PARTS

November 1974

Project Leader:
W. G. Eby
Department 142

Project Team:
C. W. Berry
J. C. Bolinger

PDO 6984674
Final Report

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Technical Communications

**Kansas City
Division**

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

TRANSDUCER FOR GAGING PRESSURE-SENSITIVE PARTS

BDX-613-879 (Rev.)
November 1974

Prepared by W. G. Eby, D/142, under PDO 6984674

A low-contact-force, linear, variable-displacement transformer has been developed. This air-actuated transducer is wholly pneumatic and is capable of developing a contact force ranging from 0.7 to 1.5 grams regardless of its attitude. Because the outside diameter of the case is 0.5 inch (12.7 mm), the unit is readily adaptable to existing fixtures.

THE BENDIX CORPORATION

KANSAS CITY DIVISION

KANSAS CITY, MISSOURI

A prime contractor for the
Atomic Energy Commission
Contract Number AT(29-1)-613 USAEC

CONTENTS

Section	Page
SUMMARY	6
DISCUSSION	8
SCOPE AND PURPOSE	8
ACTIVITY	8
<u>Gas Bearing Design</u>	12
<u>Porous Graphite Test Bearing</u>	15
<u>Orifice-Compensated Bearing Evaluation</u>	17
<u>Porous Graphite Bearing Re-Evaluation</u>	18
<u>Counterbalance Design Evaluation</u>	21
<u>Gage Head Design-Concept Evaluation</u>	22
<u>Prototype Transducer</u>	24
<u>Functional Testing</u>	30
ACCOMPLISHMENTS	34
FUTURE WORK	35
REFERENCES	37
BIBLIOGRAPHY	38
APPENDIX. TRANSducer DESIGN WITH SIDE-BY-SIDE PISTONS	39
DISTRIBUTION	45

ILLUSTRATIONS

Figure		Page
1	Concept of Mechanically Linked, Counterbalanced, Air-Actuated Transducer.	9
2	Concept of Pneumatic Counterbalanced Transducer	11
3	Expected Relationships Between Pressure Drop Across and Flow Rate Through Specimens of Porous Material as Functions of Length and Cross-Sectional Area	13
4	Measured Air Flow Rate Through Graphite Specimens.	14
5	Pressure Drop Versus Flow Rate for Three Specimens of CS Graphite	16
6	Section of Orifice-Compensated Test Bearing	18
7	Setup for Re-Evaluation of Porous Graphite Bearing.	19
8	Section of Counterbalance Engineering Evaluation Model.	21
9	Section of Gage Head Engineering Evaluation Model.	23
10	Counterbalance Damping Method	24
11	Section of Pneumatic Counterbalance Transducer (P-76159).	25
12	Pneumatic Counterbalance Transducer Prototype (Polaroid)	26
13	Test Setup for Determining Contact Force Versus Angular Position (P-76101)	32
14	Disassembled Prototype in Angle Test Fixture (P-76106).	34
A-1	Transducer With Side-by-Side Pistons (Battelle Figure 17)	41
A-2	Detail of Transducer With Side-by-Side Pistons (Battelle Figure 17)	42

TABLES

Number		Page
1	Moment Capability of Porous Graphite Bearing. . . .	20
2	Pressure Differentials of Counterbalance.	22
3	Prototype Functional Test	27
4	Pressure Differentials Necessary to Obtain 1-Gram Contact Force of Prototype.	28
5	Pocket Pressures of Prototype	28
6	Prototype Contact Force After Seating End Cap.	29
7	Prototype Contact Force After Sealing End Cap.	29
8	Results From Determination of Optimum Design Parameters	30
9	Statistical Analysis of Connected System.	33
10	Statistical Analysis of System With 18-Inch Separation of Sections	35

SUMMARY

The increased use of low-density materials in the manufacture of parts has seriously challenged the measurement capabilities of the available dimensional measurement systems. Expanded polystyrene and silicone foam rubber are examples of materials that cannot be accurately inspected by available contact or noncontact gaging equipment. A study of the relationship between tip-force and tip-radius in the gaging of polystyrene disclosed that a contact force greater than two grams could produce deformation depths exceeding the part tolerance.

The objective of this project was to design and build a low-contact-force displacement transducer which would maintain its gaging force regardless of its orientation. To achieve this objective, a transducer concept was developed using: (1) air bearings to minimize frictional drag on the sensing shaft; (2) controlled air pressure on the sensing shaft to produce the desired contact force; and (3) a counterbalance system to maintain a uniform contact force.

An evaluation of this concept indicated that a uniform contact force could not be maintained. Because of case-size limitations, the friction developed by the mechanical counterbalance linkage would be greater than the desired contact force. The study indicated that the desired contact force could be obtained by increasing the diameter of the case. A pneumatic concept was then developed using a case-diameter of 0.5 inch (12.7 mm). This concept utilizes a pressure drop across the counterbalance cylinder to support the gravity loads along the axis of the cylinder when the orientation of the probe is changed.

The most important facet of this concept is the unique method employed for detecting and neutralizing gravitational forces. A two-element air bearing was designed to detect any axial displacement of the counterbalance cylinder. When a displacement is detected, the air bearing instantly develops pressure differentials across the ends of the cylinder to maintain the cylinder in a centered position. These differential pressures are tapped and are transmitted to the gage-head section. The design of the gage-head sensing shaft includes a piston upon which the differential pressures are directed. Since the average area of the piston equals the area of either end of the counterbalance cylinder, the action of the shaft is similar to the action of the counterbalance cylinder. A one-gram contact force was obtained from the differential pressure by increasing the area of the piston face.

Prior to the construction of a prototype transducer, the design concept was tested by the use of a model that permitted making

changes in details. The counterbalance cylinder was found to require the addition of damper pistons to obtain the desired sensitivity. With these incorporated, the unit functioned with a contact force of less than two grams.

After the successful testing of the model, the design of the prototype transducer was finalized and the unit was built. Size limitations were established for a case diameter of 0.5 inch (12.7 mm) and a length of 4 inches (101.6 mm). The functional limits were set for a tip pressure of 0.5 to 2 grams obtainable in any position of the unit and with the counterbalance and the gage-head section separated by a distance of 18 inches (0.457 m). Tests of the prototype found it capable of meeting these limits with an air-supply pressure of 3.5 psi (24.1 kPa).

Tests also were performed to determine performance limitations of the unit under simulated shop conditions. These tests evaluated the dynamic response characteristics and the effects of vibration. The vibration test found the unit to be insensitive to vibration up to 50,000 μG (0.0013 m/s^2) at an amplitude of 250 microinches (0.0064 mm). At that point, the unit was found to have a linear displacement of 5 microinches (0.0001 mm). The frequency response of the unit was established at 9.1 Hz.

The use of shop air had been expected to contaminate the air bearing used to support the sensing shaft and the counterbalance cylinder; however, continued functioning on shop air did not produce any reduction in operational efficiency.

The dimensional relationship of the outside diameters of the sensing shaft to the inner diameters of the case is critical to the successful functioning of the unit. Part tolerances and concentricity relationships of approximately 0.0001 inch (0.0025 mm) are required; deviations from these values produce an adverse effect upon the contact force range.

Changes in the air-supply pressure will not only change the contact force but will increase the force range. An increase in the operating pressure had been expected to impart a rotary action to the sensing shaft; however, operating the unit with a supply pressure of 15 psi (103.2 kPa) did not produce the expected action, even when the unit was rotated through a 360° vertical plane.

The transducer developed through this project makes possible the obtaining of accurate dimensional data from parts constructed of pressure-sensitive materials. Further improvement upon the design is expected to produce gage heads having a contact force in the 0.1-gram range. This type of development will assure the availability of dimensional data under many circumstances.

DISCUSSION

SCOPE AND PURPOSE

The increased use of low-density materials in the manufacture of parts has seriously challenged the measurement capabilities of the available linear contact inspection equipment. Materials such as expanded polystyrene and silicone foam rubber cannot be accurately inspected by available contact or noncontact equipment. Commercially available equipment either displaces the part or produces material deformations that exceed the part tolerances.

A prior study found that a contact force greater than two grams should not be used to inspect expanded polystyrene parts.¹ Several methods of obtaining a contact force of two grams or less were considered, and a conceptual design was completed. This project was initiated to develop the concept and to construct a prototype.

ACTIVITY

To obtain a high degree of expertise in the design of air bearings, a contract was negotiated with Battelle Memorial Institute to evaluate and to change the proposed transducer design, as necessary. Upon completion of this task, Battelle was authorized to construct the prototype.

The construction of a device similar to that shown in Figure 1 was planned, and the following functional and design goals were established.

- The transducer would be actuated by air.
- A contact force ranging from 0.5 to 2 grams for any gage-head orientation angle would be developed and maintained.
- The unit would have a 0.625-inch (15.9 mm) diameter and would be approximately 3 inches (7.6 cm) long.
- A standard transformer and amplifier would be utilized for sensing displacement.
- Porous graphite material would be used for the air bearing to contain the counterbalance cylinder and the sensing shaft.
- An opposed mass would compensate for the gravitational forces acting on the sensing shaft during orientation-angle changes.

Figure 1. Concept of Mechanically Linked, Counterbalanced, Air-Actuated Transducer

- A minimum-friction pivot and linkage system would provide the linkage between the compensator cylinder and the sensing shaft.
- The facing-end areas of the cylinder and the shaft would vary enough that pocket air pressure would develop continuous shaft extension and contact force.

Battelle's engineering evaluation of the concept disclosed several difficulties that would require correction before the device could function as desired. The following were the main areas of concern.

- The linkage-bearing contact pressures would be approximately 400 psi (2.8 MPa) and would result in excessive contact forces.
- With an assumed coefficient of friction of 0.1, no contact force would be developed under certain loading conditions.
- Large linkage angles would produce excessive forces on the pivot pins, thus resulting in a tipping movement of the shafts with respect to their air bearings.

Calculations indicated that if the weight and the size of some of the components were reduced significantly, and if the coefficient of friction could be reduced, the system might be made to work--although probably not at the desired level of force. An alternative would be to increase the case diameter to permit redesign of the linkage system.

The transmission of the counterbalance force pneumatically, rather than mechanically, was another approach suggested by the Battelle engineer. As shown in Figure 2, the wholly pneumatic system is composed of two sections: the compensator unit and the gage head. The compensator section is basically a pressure-differential generator. As gravity acts upon the compensator cylinder, a pressure drop develops across the shaft to support the gravity loads along the axis of the shaft when the system orientation is changed. The pressure differential developed in pressure pockets P_1 and P_2 maintain the compensator shaft in a centered position.

The compensator pressure pockets are coupled to similar pockets in the gage-head unit. When the compensator shaft and the gage-head shaft are identical in weight and have equal-pressure cross-sectional areas, the gage-head sensing shaft will duplicate the action of the compensator shaft. To obtain the desired contact force, the cross-sectional pressure area of the gage-head shaft of pressure pocket P_1 was increased to produce a theoretical 1-gram contact force for a pressure of 1 psi (6.9 kPa) in the gage-head pockets.

Figure 2. Concept of Pneumatic Counterbalanced Transducer

Based on the above information, Battelle was authorized to proceed with the development of the concept and the construction of a prototype to function within the following limitations.

- The contact force would be between 0.5 and 2 grams.
- The developed contact force would be within the above range for any orientation angle of the transducer.

- The design would include a device to eliminate axial rotation of the sensing shaft.
- The transducer would function within the contact-force range when the sections were separated by a distance of 18 inches (45.7 cm).
- The case would be 0.5 inch (12.7 mm) in diameter and 4 inches (10.2 cm) long.
- The gage head would be designed for disassembly to permit replacement of the LVDT core and field.

Gas Bearing Design

Battelle's initial task was to design an air bearing to contain the counterbalance shaft and the sensing shaft. Porous graphite materials were selected for use because of their low coefficient of friction and the ease of construction. With limited data available for use in designing a bearing, the empirical study approach was used to determine the pressure-flow characteristics through the restrictor material. The data obtained was intended to generate a family of curves similar to those shown in Figure 3. Once the study curves were established, a cross-sectional area and a flow length could be selected for a bearing design which would fall on or near the theoretical pressure-flow curves. Assuming a nominal pressure drop across the restrictor, the curve could be linearized about that pressure drop and the flow rate through a restrictor operating at the nominal pressure drop could be established. Using this information, the bearing clearance dimension and the flow length could be designed to obtain the same nominal flow rate and a reasonable gas-film stiffness.

Tests conducted using the designated material produced data with much variability between like specimens. The detailed examination of the material found that it had a grain structure resembling grains of rice. New specimens having the grain structure parallel and perpendicular to the flow length were made with various diameters and with machining capabilities for varying the flow length (thickness).

Figure 4 is representative of the data obtained from the new specimens. The flow rate was expected to be a linear function of the specimen cross-sectional area with an increase in flow length producing a proportional increase in flow rate. Results of this nature will produce curves appearing as straight lines through the zero point. Because the curves do not appear as expected, the existence of some sensitivity to the cross-sectional area which is apparently related to a nonuniformity in the properties of the material is implied. In addition, the scatter in

Figure 3. Expected Relationships Between Pressure Drop Across and Flow Rate Through Specimens of Porous Material as Functions of Length and Cross-Sectional Area

data points for specimen-lengths of 0.0625 inch (0.16 cm) indicates a variability in the flow characteristics of the specimens. This variability is greater than that which can be tolerated in the design of a gas bearing.

Figure 4. Measured Air Flow Rate Through Graphitar Specimens

A new material was obtained and specimens 0.75 inch (19 mm) in diameter were made with the thickness varying from 0.107 inch (27 mm). Flow rate data from these specimens are presented in Figure 5. A significant difference is shown in the flow rates obtained through two specimens of essentially the same length. The 0.052-inch (1.32 mm) specimen again indicates an apparent nonuniformity in the material.

Based on the data obtained from the tests, the conclusion was reached that the differences in the flow rates of the specimens are probably due to nonuniformities of the materials.

Porous Graphite Test Bearing

Although material nonuniformities appeared to be present in the graphite materials, calculations were made using the characteristics of the Graphitar material and the Union Carbide Grade CS material to determine the resulting bearing parameters. Two papers were used as guides for the equations of design and evaluation.^{2 3}

Based on the measured flow characteristics of the Graphitar specimens and the geometry of the Bendix design, calculations were made to determine the design characteristics of a porous-wall bearing using a supply pressure of 45 psi (310.3 kPa). The calculated bearing clearance, 67 microinches (0.0017 mm), was considered impractical for this gas bearing application.

Calculations were also made to determine the design parameters of a gas bearing using the Grade CS material and a supply pressure of 75 psi (517.1 kPa). The calculated radial clearance for the bearing geometry, 485 microinches (0.0123 mm), was considered practical; therefore, in spite of material nonuniformities, a test bearing was designed and built.

The porous-graphite test bearing was operated at the designed supply pressure of 75 psi (517.1 kPa) with a radial clearance of approximately 0.0005 inch (0.0127 mm). The bearing exhibited severe pneumatic hammer which rendered it essentially inoperative. By slowly reducing the supply pressure to approximately 5 psi (34.5 kPa), a nearly stable condition was obtained. At this supply pressure, the shaft rotated freely and exhibited an axial oscillation.

The paper by Robinson and Sterry⁴ was consulted for a discussion on the stability of porous-wall gas bearings. It stated that under certain conditions, an increase in the bearing clearance or a reduction in the supply pressure produced a stable condition. Because a reduction in the supply pressure did eliminate the pneumatic hammering, additional shafts were made to determine the effect of increasing the radial clearance.

Figure 5. Pressure Drop Versus Flow Rate for Three Specimens of CS Graphite

Tests using the various shafts indicated that increases in radial clearance produced a stable bearing. This stability, however, was restricted to the elimination of the air-hammer phenomenon; the bearing still exhibited both an axial oscillation and a rotation of the shaft, neither of which seemed to be controllable. A row of rather large openings or nonuniform pores were observed in the bore of the graphite sleeve, and the assumption was made that this condition probably contributed to the instability of the shaft. Although the test bearing operated properly in terms of floating the shaft in a frictionless manner, the existence of axial and rotational instability severely limited its use for the intended linear transducer application.

Sterry and Robinson had tried to correlate the existence of pneumatic hammer with a critical axial Reynolds number in the bearing clearance. For the original design clearance, the axial Reynolds number was calculated to be 957 for a dimensionless length parameter of 0.875. These values did not fall in the region of instability defined by Sterry and Robinson and no explanation for the unstable behavior was developed.

Orifice-Compensated Bearing Evaluation

In an effort to obtain an air bearing that would limit the rotary action, the decision was made to evaluate an orifice-compensated bearing. Calculations were made and a design was developed to utilize a supply pressure of 40 psi (275.8 kPa), a radial clearance of 0.001 inch (0.025 mm), and an orifice diameter of 0.011 inch (0.28 mm). This design concept, shown in Figure 6, was intended to produce the stability that apparently was not available through the use of porous graphite material.

Pneumatic hammer was not encountered when the bearing was operated; however, rotational driving of the test shaft occurred at a magnitude greater than that encountered with the porous graphite bearing, and a reduction in the supply pressure did not materially affect the problem. The rotational drive was attributed to a possible physical irregularity which might produce different circumferential flow rates around the shaft in opposite directions at a particular location. For example, this effect could be caused by orifice holes which were off-center or slanted relative to the bearing bore, thus producing a jet-type action.

A second bearing was constructed and the greatest possible care was taken in drilling the orifice feed holes and machining the bearing bore to assure the proper orientation of the feed holes. The evaluation of this second orifice bearing produced results similar to those obtained from the first bearing--that is, a strong tendency to rotate the shaft was present.

The evaluation of the Bendix concept included the determination of the gas bearing performance parameters, a portion of which was the determination of the supply pressure to support the estimated radial load applied to the sensing shaft. The maximum load was estimated to be 29 grams, including the sensing shaft weight and the linkage-geometry force. This approximate load divided by the projected bearing area of 0.125 in.² (0.018 mm²) gave an effective bearing pressure of 0.51 psi (3.5 kPa). Allowing for a load coefficient of 0.1, a supply pressure of 5.1 psi (35.2 kPa) would be required to support the estimated radial load.⁵ Because the applied moment would also load the bearing, a higher supply pressure would be required.

Because the difficulty in analytically estimating the moment capabilities of a porous gas bearing, the experimental approach was used to determine if a bearing could support the loads that would be imposed on a working probe. Using the setup shown in Figure 7, tests of a porous graphite bearing provided the results shown in Table 1.

Figure 7. Setup for Re-Evaluation of Porous Graphite Bearing

Table 1. Moment Capability of Porous Graphite Bearing

Supply Pressure (psi) (kPa)	Moment Capability	
	C* = 0.0010 in. (0.0254 mm) (g·in.) (N·cm)	C* = 0.00125 in. (0.0318 mm) (g·in.) (N·cm)
5 (34.5)	11.9 (0.296)	
10 (68.9)	17.5 (0.436)	15.25 (0.380)
15 (103.4)	19.4 (0.483)	18.4 (0.458)
20 (137.9)	20.2 (0.503)	19.4 (0.483)
*Radial clearance		

The moment capability of the gas bearings was determined by measuring the applied moment at a given supply pressure at which a sticking of the shaft in the bearing bore occurred. Based upon this data and an estimate of the applied loads, a supply pressure of 10 psi (68.9 kPa) would be required for a porous gas bearing having a reasonable clearance.

A comparison of the test data from the first graphite bearing with the design equations indicated that the reduction in supply pressure and the increase in bearing clearance were conducive to increased bearing stability. An evaluation of the bearing design for lower supply pressures and bearing clearances therefore was deemed necessary.

Using the estimates of the load and movement requirements for the bearing tests and for tests performed to determine the actual load and movement capabilities of the test bearing, a design supply pressure of 10 psi (68.9 kPa) was chosen. A radial clearance of 0.001 inch (0.025 mm) was chosen, which resulted in an axial Reynolds number for the bearing gap of 81. The dimensionless length parameter, L , was computed to be 0.885. (Based on the paper by Sterry and Robinson,⁶ this should produce a stable bearing configuration.)

In tests performed on the bearing, the shaft exhibited none of the rotary or axial motions found in previous tests.

Counterbalance Design Evaluation

After the development of the air bearing, work was started to test the functional capabilities of the counterbalance system. An engineering evaluation model was designed and built (Figure 8). The application of a 10-psi (68.9 kPa) supply pressure produced an unstable axial motion of the shaft. The engineering design calculations had indicated that a small change in the inlet flow to either recess P_1 or P_2 would produce a rapid change in pocket pressure.⁷

Figure 8. Section of Counterbalance Engineering Evaluation Model

A paper by H. H. Richardson⁸ indicates that a gas bearing will be unstable if the pool volume is greater than one-third the clearance volume. Since the design had a large pool volume and essentially a clearance volume of zero, the end housings were bored larger to permit the entrance of a longer shaft into

the pocket to reduce the sensitivity of the unit to pocket-pressure changes. The outlet flow was then between the shaft and the end-bore diameters (capillary tube-type restriction) instead of across the end of the shaft.

This change in design eliminated the axial instability but produced unequal pressure differentials, depending upon the position of the unit. The results obtained are shown in Table 2.

Table 2. Pressure Differentials of Counterbalance

Axis Orientation	$P_1 - P_2$ (psi) (kPa)
Horizontal	0 (0)
Vertical, P_1 Up	+0.200 (1.4)
Vertical, P_2 Down	-0.250 (-1.7)

Measurements made of the end-housing bores disclosed an inside-diameter variation of 0.00025 inch (0.00635 mm); also, the center line of the bore was not coincident with the air-bearing axis. A new pair of end housings was therefore carefully machined and assembled.

The functional test of the revised system indicated the calculated differential pressure of 0.225 psi (1.55 kPa) in either vertical position.

Gage Head Design-Concept Evaluation

With a functional counterbalance system perfected, a gage head model, shown in Figure 9, was built according to the design analysis previously reported in detail.⁹ The functional test of the united gage head and counterbalance disclosed that the available pressure differential would not support the weight of the sensing shaft in either vertical position; also, the shaft would not consistently extend itself in the horizontal position.

The difficulty in obtaining proper operation was diagnosed as insufficient air pressure in the probe pressure pockets. The apparent cause was determined to be a series-orifice-effect which caused a pressure drop between the two assemblies.¹⁰

The analysis indicated that supply pressures exceeding 40 psi (275.6 kPa) were necessary to develop a nominal pressure of 1 psi (0.869 kPa) in the probe pockets. The solution to this

Figure 9. Section of Gage Head Engineering Evaluation Model

problem was to remove some of the restriction in the flow between the counterbalance and gage head probe pockets. This was accomplished by increasing the diameter of the counterbalance outlet orifices to 0.060 inch (1.524 mm) and the diameter of the inlet orifices to 0.030 inch (0.762 mm). Gage head inlet orifices were left unchanged pending a functional evaluation of the changes made.

The test indicated partial success with evidence of some sensing-shaft support in the tip-down position; no shaft support was

evident in the tip-up position. A vibration of the sensing shaft was observed and it was found to be sensitive to the orientation of the counterbalance shaft. The vibration was most severe when the counterbalance unit was in a vertical position. Support of the sensing shaft was evident when the unit was positioned with the tip down since the tip-force was less than the mass weight of the shaft.

The test also indicated that the changes in the inlet and outlet flow restrictors for the counterbalance unit had produced an axial instability of the shaft. The solution, shown in Figure 10, involved adding damping to the counterbalance shaft.

Figure 10. Counterbalance Damping Method

A new set of counterbalance end-housings and a new shaft were made. The functional evaluation of the total system showed the unit to be completely successful. The pressure differential across the counterbalance shaft for both vertical orientations was 0.35 psi (2.41 kPa), which checked with the cylinder mass of 7.84 grams. The sensing shaft tip-force was found to be approximately 1 gram in the tip-up position and 0.9 gram in the tip-down position. These values were obtained using a 3.5-psi (24.1 kPa) system supply pressure.

Prototype Transducer

With the successful functioning of the engineering evaluation model, the design of the prototype was completed and construction

was started. Figure 11 is an isometric sectional view of the prototype, together with a view showing the two sections separated. Figure 12 shows the two sections united. The final system has only one air-supply port instead of the two shown in Figure 11.

Figure 11. Section of Pneumatic Counterbalance Transducer

The functional test of the completed prototype unit found the transducer incapable of functioning as designed. Battelle reported the unit produced a contact force of 1.1 grams when positioned in the horizontal plane. In a tip-up position, the contact force went to zero at 13.5 degrees above the horizontal. A tip-down position produced a contact force of 6.5 grams, and 8.75 grams contact force was obtained with a zero supply pressure. From these measurements, Battelle concluded that the pressure drop between the two units was too high to even approximate the design assumption that the pressure acting on the sensing shaft piston was equal to that acting on the counterbalance shaft. The following changes therefore were recommended.

- Add external air lines to achieve a lower pressure drop.
- Reduce the clearance between the sensing-shaft piston and the gage head inside diameter to provide a greater restriction of air at that location.

Figure 12. Pneumatic Counterbalance Transducer Prototype

- Reduce the weight of the sensing shaft to permit support at the lower differential pressures.
- Change the differential area on the sensing-shaft piston to achieve the desired contact force.
- Provide throttling on one or both of the gage-head exhaust ports.

Based on the required system changes, the conclusion was reached that changes in the design would not produce a unit that would function as desired. To provide the required functional capability and still utilize the basic engineering concept, Battelle submitted a third proposal. The information concerning this concept is contained in the Appendix.

When Bendix was informed of the Battelle test data, the shipment of the prototype to this facility was requested. With the unit in-house, a series of tests was designed to determine the reason for the inoperative condition.

The first test was made to determine the differential capabilities of the counterbalance system. The test data found the system produced a differential pressure of 0.367 psi (2.5 kPa) when the unit was in a vertical position (tip-up or tip-down). A zero

differential was found when the unit was in the horizontal position. These values were obtained using a system pressure of 5 psi (34.5 kPa). As previously reported, Battelle had found the differential to be 0.35 psi (2.4 kPa) at a system pressure of 3.5 psi (24.1 kPa). This variance was expected with the increase of 1.5 psi (10.3 kPa) in the system air pressure.

The functioning of the complete unit was tested and produced the results shown in Table 3. These values tended to confirm the Battelle opinion that a pressure drop existed within the differential pressure air lines. Because these results did not agree with the functional results from the engineering evaluation model, the next studies were established to determine the dimensional accuracy and relationship of the sensing shaft and the gage-head case internal diameters, to determine the gage-head pocket pressures, and to determine the gage-head pressure differentials needed to obtain a 1-gram tip contact force for the three unit orientations.

Table 3. Prototype Functional Test

Axis Orientation	Tip Contact Force
Tip Up	Probe not extended
Horizontal	Probe not extended
Tip Down	7.2 grams

The dimensional study found that the clearance between the sensing shaft piston and the gage-head case was 0.0005 inch (0.0125 mm) greater than specified by the design. The weight of the sensing shaft was also found to be 1.5 grams heavier than the calculated weight of the counterbalance shaft; the actual difference was 1.24 grams. These conditions alone would have been sufficient cause for the operational failure.

The next test was made to determine the pressure differential necessary to obtain a 1-gram contact force in the horizontal and the two vertical positions. The pressure needed to support the 9.3-gram sensing shaft was calculated to be 0.41 psi (2.8 kPa). The results of the test are shown in Table 4.

The pressures for the tip-up and horizontal positioning of the gage head indicated that the weight and dimensional errors were not the primary reasons for the unit's inoperative condition; rather, the test indicated the suspected pressure drop was associated with the transmittal air lines for the P_1 pressure

Table 4. Pressure Differentials Necessary to Obtain 1-Gram Contact Force of Prototype

Axis Orientation	Pressure Differential (psi) (kPa)
Tip Up	0.81 (5.6)
Horizontal	0.10 (0.7)
Tip Down	0.52 (3.6)

pocket. To confirm this contention, a special gage-head sensing shaft was built that permitted tapping the gage-head pocket pressures. If the conclusion were valid, the application of a known supply pressure should produce an obvious pressure differential within the pressure pockets. The results obtained with a supply pressure of 1.4 psi (9.6 kPa) when the unit was in the horizontal position are shown in Table 5.

Table 5. Pocket Pressures of Prototype

Pressure Pocket	Pressure (psi) (kPa)
P ₁	0.70 (4.8)
P ₂	0.74 (5.1)

This last test indicated the presence of a pressure drop that was not necessarily associated with one air line; the evidence suggested the possibility of a clogged or restricted air line. Because the possibility existed that the calculations pertaining to the flow characteristics within the gage-head air lines might be in error, new calculations were performed. Again, indications were that the flow should be laminar and that the expected pressure drop should be approximately 0.016 psi (110.3 Pa). These figures indicated that the gage-head air lines should be checked for the presence of foreign material. X-ray negatives made of the gage-head case did not show the presence of obstructions; however, the negatives did show that the gage-head end cap was not seated by approximately 0.010 inch (0.254 mm). After the end cap was properly seated, the prototype was again tested for contact force. The results are shown in Table 6. These values, even though not up to the design level of 0.36 psi (2.4 kPa),

indicated for the first time that the system failure could be due to the manufacturing process rather than to a faulty design. This was substantiated by the continued indication of a pneumatic short during a leak test.

Table 6. Prototype Contact Force After Seating End Cap

Axis Orientation	P ₁ - P ₂ (psi) (Pa)
Tip Up	0.12 (827.4)
Horizontal	0.08 (551.6)
Tip Down	0.14 (965.3)

A review of the X-ray negative was made using a magnifier. An indication was found that the end cap had edge chamfers on surfaces where none were permitted. Upon disassembly, the chamfers were confirmed. A thread sealant was applied to the affected edges to eliminate the pneumatic short, and the gage head was reassembled.

Using the original sensing shaft, the complete unit was tested for functional capability. Upon the application of a system supply pressure of 5 psi (34.5 kPa), the tip-contact forces shown in Table 7 were obtained. A comparison of this data to the original shows that a functional improvement was realized. The contact force obtained with the unit in the horizontal position indicates that a pressure drop was not the cause of the unit being inoperative. It also indicates that the design is correct in that the force is nearly the same as the 1-gram force for which the unit was designed.

Table 7. Prototype Contact Force After Sealing End Cap

Axis Orientation	Contact Force (g)
Tip Up	0
Horizontal	1.2
Tip Down	4.1

Before the sensing shaft design was changed, the exact design parameters were determined. Test shafts were designed with three different piston diameters and a means of adjusting their weight. Each was tested in the complete unit with a system air pressure from 2 to 6 psi (13.8 to 41.4 kPa). The results of these tests indicated that the optimum function was obtained using a 3.5-psi (24.1 kPa) supply pressure, a sensing shaft weighing 7.48 grams, and a piston diameter 0.001 inch (0.025 mm) larger than the original design specified. The data obtained is shown in Table 8.

Table 8. Results From Determination of Optimum Design Parameters

Axis Orientation	Contact Force (g)	Differential Pressure (psi) (kPa)
Tip Up	1.7	0.332 (2.3)
Horizontal	1.3	0.020 (0.1)
Tip Down	1.5	0.329 (2.3)

Varying the system supply pressure up to 6 psi (41.4 kPa) reduced the contact-force data range but increased the amount above the 2-gram limit. Lowering the pressure reduced the force but increased the data range. At 2 psi (13.8 kPa), the sensing shaft would not extend in the tip-up position.

Based on the data from the study, revisions were made to the design and the new sensing shaft was built. Testing of the total system did not produce data that was materially different from the preceding tests. The unit was considered acceptable and ready for functional testing.

Functional Testing

At the start of this project, the following three tests were determined to be sufficient for ascertaining the functional capabilities of a successful system:

- A test to define the dynamic response characteristics;
- A test to determine the measurement limitations that would be imposed by ambient vibrations; and
- A test to indicate the ability of the system to function with a constant contact force, irrespective of the angular position.

The dynamic response test was preceded by an engineering analysis to determine the expected limits. Assuming no function and no damping action within the air lines, the frequency value is

$$f = \frac{1}{2\pi} \sqrt{\frac{F}{md}},$$

where

f = frequency in hertz,

F = contact force in dynes,

m = combined mass weight of cylinders in grams, and

d = displacement of the gage-head probe in centimeters.

Calculations indicated that the theoretical value would be 12.8 ± 0.4 Hz. Using a variable speed 0.010-inch (0.26 mm) eccentric-diameter shaft, the ability of the unit to track the surface was tested. Ten observations were made and data were taken from an oscilloscope. At a low-speed rotation, the scope trace was adjusted to fit between major grid-lines. The rotational speed of the shaft was increased until 50 percent of either the peak or valley forms became displaced from the grid set-lines. The average frequencies then read from the scope were 9.1 Hz and 12.1 Hz. The smaller value reflects the kick-off position (the point of maximum sensing shaft extension), while the larger value reflects the lift-off position (the point of maximum sensing shaft retraction). While no attempt was made to determine the reasons for discrepancies in the data, friction and damping within the system undoubtedly would account for much of the variability.

The next study was designed to test the resistance of the unit to the effects of ambient vibrations. The unit was positioned to function in three planes, then subjected to increased frequencies. Approximately 10 microinches of displacement occurred at 42.5 Hz with an input amplitude of 0.00025 inch (6.35 μ m) in the tip-up position; the same amount of displacement occurred in the tip-down position at 40 Hz with an input amplitude of 0.00023 inch (5.84 μ m).

According to the findings of a study of ambient vibrations in the factory floor,¹¹ no area has been found where the frequency and amplitude levels equal those obtained in this test.

The final test was to determine the contact-force capability of the unit as a connected system and as a two-part system separated

by approximately 18 inches (457.2 mm). Figure 13 shows the test fixture and the contact-force readout instruments. The force-sensing element was a cantilever strain gage using a Cimeron Volt-Ratio Meter as a digital readout. Calibration of the system disclosed that a 1-gram contact force was displayed as 56 millivolts.

Figure 13. Test Setup for Determining Contact Force Versus Angular Position

Five angular positions were used to test the system: the vertical tip-up position, an inclination of 45° above horizontal, the horizontal position, an inclination of 45° below horizontal, and the vertical tip-down position. Ten readings were obtained at each position: five were obtained as the test fixture was rotated from the tip-up position to the tip-down position, and five other readings were obtained after rotating the test fixture

approximately 10 degrees past the tip-down position and returning through each position to the tip-up position. The average and standard deviation of each position were computed and a correction value for the effect of gravity on the strain gage was applied to each average force value to obtain a corrected value. The data obtained is shown in Table 9.

Table 9. Statistical Analysis of Connected System

Reading	System Angular Orientation*				
	Tip Up (mV)	45° Up (mV)	Horizontal (mV)	45° Down (mV)	Tip Down (mV)
1	95	63	91	137	131
2	91	61	103	140	133
3	91	59	85	135	131
4	95	60	91	131	131
5	93	63	89	135	127
6	97	60	95	140	131
7	95	60	91	141	131
8	97	60	89	142	133
9	99	63	91	140	130
10	93	63	85	139	131
Average	94.8	61.0	91.7	137.9	130.9
σ	2.7	1.6	5.0	3.6	1.8
Correction Value	0	0.7	2.0	3.8	4.5
Corrected Contact Force	1.7g	1.0g	1.3g	1.8g	1.5g
*Gage head cylinder weight: 7.3 grams; System air pressure: 3.5 psi (24.13 kPa)					

The final test was a study of the system's capability to function when the two parts were separated approximately 18 inches (457.2 mm). Figure 14 shows the setup used. Note that the electrical lead is fastened directly to the gage-head unit while

the system air supply line is attached to the counterbalance unit. Three air lines are needed to connect the two units pneumatically. The inside diameter of the hose is 0.125 inch (3.175 mm), and special hose adaptors are needed for attachment to the compensator. The test data obtained is shown in Table 10.

Figure 14. Disassembled Prototype in Angle Test Fixture

ACCOMPLISHMENTS

A comparison of the test data with the design requirements indicates that the air-actuated constant-contact LVDT gage-head system developed under this project functions within the design limits. That the system can function accurately at existing plant ambient-vibration levels indicates the possibility that systems can be developed to function at lower contact forces.

Table 10. Statistical Analysis of System With 18-Inch Separation of Sections

Reading	System Angular Orientation*				
	Tip Up (mV)	45° Up (mV)	Horizontal (mV)	45° Down (mV)	Tip Down (mV)
1	7.5	4.7	7.9	11.9	10.9
2	7.6	5.5	8.7	12.5	11.0
3	7.7	4.3	7.5	12.1	10.4
4	7.6	5.4	8.6	12.6	10.7
5	7.7	4.7	7.9	12.0	11.3
6	7.7	5.5	8.7	12.5	11.4
7	7.9	4.8	8.0	12.5	10.9
8	8.1	5.8	9.0	12.6	11.4
9	7.8	4.9	8.9	12.4	10.0
10	8.1	5.5	8.7	12.8	11.4
Average	7.77	5.11	8.31	12.39	10.94
σ	0.21	0.49	0.49	0.29	0.47
Correction Value	0	0.7	2.0	3.8	4.5
Corrected Contact Force	1.4g	0.8g	1.1g	1.5g	1.2g
*Gage head cylinder weight: 7.3 grams; System air pressure: 3.5 psi (24.13 kPa)					

FUTURE WORK

With dimensional changes and a reconstruction of the system, contact forces of less than one-half gram are believed to be easily obtainable. If only a fixed position of the gage head is desired, the use of a gage head similar to the engineering evaluation model can provide this capability. The study of the dynamic response characteristics indicates that consideration must be given to the surface texture of the part and the rotary speed if the system is to be used in this manner.

If the need arises for a transducer system having an adjustable though constant contact force regardless of angular position, the recommendation is made that consideration be given to the Battelle side-by-side piston concept described in the Appendix.

REFERENCES

¹ Deleted.

²H. J. Sneck and K. T. Yen, "The Externally Pressurized, Porous Wall, Gas-Lubricated Journal Bearing.1." *ASLE Transactions* 7, Chicago: 1964, pp 288-298.

³C. H. Robinson and F. Sterry, "The Static Strength of Pressure Fed Gas Journal Bearings, Porous Bearings," A.E.R.E. ED/R 1672, U.K.A.E.A. Research Group. Harwell, England: December, 1957.

⁴Ibid.

⁵Sneck and Yen.

⁶Robinson and Sterry.

⁷ Deleted.

⁸H. H. Richardson, "Static and Dynamic Characteristics of Compensated Gas Bearings," *ASME Transactions* 80, 1958, pp 1503-1509.

⁹ Deleted.

¹⁰Deleted.

¹¹C. W. Berry, Jr. and others, *Vibration Analysis* (Final Report). UNCLASSIFIED. Bendix Kansas City: BDX-613-926, November, 1973.

BIBLIOGRAPHY

Berry, C. W., Jr., and others. *Vibration Analysis* (Final Report). UNCLASSIFIED. Bendix Kansas City: BDX-613-926, November, 1973.

Grassam, N. S., and J. W. Powell. *Gas Lubricated Bearings*. London: Butterworths, 1964.

The Industrial Graphite Engineering Handbook. Union Carbide. New York: 1970 Rev.

Lenkei, A. "Close-Clearance Orifices," *Product Engineering*, 1965, p 57.

Richardson, H. H. "Static and Dynamic Characteristics of Compensated Gas Bearings," *ASME Transactions* 80, 1958, pp 1503-1509.

Robinson, C. H., and F. Sterry. "The Static Strength of Pressure Fed Gas Journal Bearings, Porous Bearings," A.E.R.E. ED/R 1672, U.K.A.E.A. Research Group. Harwell, England: December, 1957.

Sneck, H. J. "A Survey of Gas-Lubricated Porous Bearings," *Journal of Lubrication Technology*, October, 1968, p 804.

Sneck, H. J., and R. C. Elwell. "The Externally Pressurized, Porous Wall, Gas-Lubricated Journal Bearing," *ASLE Transactions* 8, Chicago: 1965, p 339.

Sneck, H. J., and K. T. Yen. "The Externally Pressurized, Porous Wall, Gas-Lubricated Journal Bearing.1." *ASLE Transactions* 7, Chicago: 1964, pp 288-298.

Tang, I. C. and W. A. Gross. "Analysis and Design of Externally Pressurized Gas Bearings," *ASLE Transactions* 5, Chicago: 1962, pp 261-284.

Appendix

TRANSDUCER DESIGN WITH SIDE-BY-SIDE PISTONS

A portion of a report prepared by
Battelle Columbus Laboratories,
Columbus, Ohio for the Bendix
Kansas City Division

Transducer Design With Side-By-Side Pistons

In the original design study, a side-by-side piston arrangement had been considered. It had been rejected only because with the 5/16-inch diameter LVDT, the transducer diameter would have exceeded 1/2-inch. Although no drawings were made of this concept, it was estimated that a 3/4 to 7/8-inch diameter would be required. However, when the end-to-end configuration failed to operate, it was concluded that even a 7/8-inch diameter operational transducer would be more desirable than none, so layouts were made to determine the feasibility of building a side-by-side configuration.

In making these layouts, a check of LVDT sizes and configurations was made, and it was learned that LVDT's with a diameter of 3/16-inch had become commercially available. With a 3/16-inch diameter LVDT, it was considered feasible to build a transducer with side-by-side pistons inside a 1/2-inch diameter package.

Figure 17 is an assembly drawing showing one configuration which might be used. This system would operate as follows: air would enter through a tube (9) and pass through cylindrical manifolds (4) and into the center of the pistons (3) and (5). The manifolds (4) would be two-piece units, with the two halves located relative to each other by spring wire snap rings (10) and located in the housing (1) by a socket head screw (11). Air would travel through the inside of the pistons (3) and (5) and out through the porous graphite bearings (2) to provide the air bearing support for the pistons. In addition, air would be throttled through a hole (12) in the rear of the probe piston (3), past adjustable set screws (8) to atmosphere. The adjustable set screws (8) would be adjusted to produce enough back pressure to generate the desired one gram probe contact force.

This system does not use the differential piston area concept to develop probe force, but instead the contact force is developed by pressure on the left hand end of the probe piston. The counterbalancing forces are developed by independently developed pressures acting between the manifolds (4) and the bearings (2). This separation of functions makes it unnecessary to control the sum of the counterbalance pressures, and because the sum of

Figure A-1. Transducer With Side-by-Side Pistons (Battelle Figure 17)

Figure A-2. Detail of Transducer With Side-by-Side Pistons
(Battelle Figure 17)

the pressures does not have to be accurately controlled, tolerances can be much lower than for the system with the differential piston area.

To develop the counterbalance forces and pressures, clearances are provided between the pistons (3) and (5) and the manifolds (4). Air flows through these clearances into the cylinder area between the manifolds and air bearings (2), and out to atmosphere past valve rings (7) which control the pressure in the cylinder cavities. Movement of the counterbalance piston (5) causes the valve rings (7) to throttle the air flow so that the counterbalance piston (5) is supported in its axial direction by air pressure. By making the piston areas and weights of both the probe and counterbalance pistons equal and by making the connecting passage (13) between the two pistons short and of large cross-sectional area, the same pneumatic forces will be developed on each piston and gravity forces will be cancelled.

The counterbalance part of the transducer is basically a servo system, and for the system to experience stable operation it is necessary that certain stability criteria be met. For the system shown in Figure 17, one way to make the system stable is to provide pneumatic damper pistons (6) and maintain the following relationship between weight and sizes:

$$1.0 \geq G(h^3/1.5\pi LD^3)^2(w/g)$$

The development of the above equation and the definition of the terms is given in Appendix F. A sample calculation is given in Appendix F, which shows that if 1/8-inch long damper pistons are used, if the piston weight is 0.003 lb (1.4 grams), and if G is about 0.3 lb/in. or less, then stable operation should be obtainable with a radial clearance of 0.0018 inch or less.

Another criteria for stable performance is that the spring impedance due to air compression in the damper cylinders must be high relative to damping impedance. Calculations were made which show that if the radial clearance

"h" were greater than 0.001 inch, and the length of the clearance volume "H" is not greater than 0.030 inch, then the spring impedance will be about 3 or more times the damping impedance.

Other features of the transducer shown in Figure 17 are that it can be disassembled, that the holes for the pistons are straight-bored or reamed holes with no shoulders, and that all air passages are drilled holes or low-tolerance holes made by electrical discharge machining. These features will significantly reduce the manufacturing cost of the unit. In addition, the piston weights are greatly reduced, providing increased frequency response and reducing the axial stability control problems.

DISTRIBUTION

	Copy
R. Bulcock, AEC-KCAO	1
V. C. Vespe, AEC-ALO	2
J. A. Freed, LASL	3
S. J. Buginas, LLL	4
J. Oberteuffer, LLL	5
H. Brinkmeier, Monsanto	6
R. F. Pippert, D/100	7
L. E. Borgman, D/113	8-9
E. L. Hawkins, D/141	10
K. A. Shroyer, D/141	11
H. Thompson, D/141	12
J. S. Carson, D/142	13
J. F. Hadley, D/142	14
R. E. Odell, D/143	15
D. H. Parker, D/143	16
H. N. Wilson, D/144	17
J. Parks, D/231	18
J. D. Corey, D/554	19-23
L. Stratton, D/554	24-26
R. P. Frohmberg, D/800	27
P. Keller, D/822	28
R. C. Ufford, D/824	29
R. E. Kessler, D/864	30