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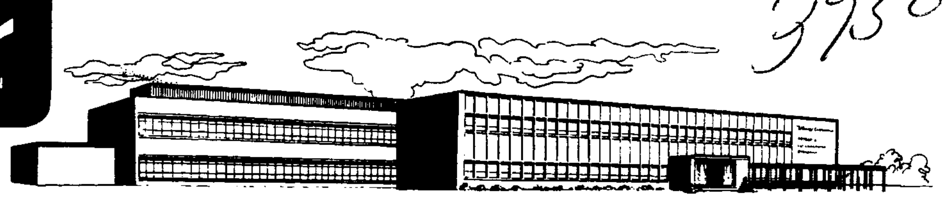
HYDRAULIC FLUID INTERACTION SERVOVALVES

Monthly Technical Report

1 February 1966 - 1 March 1966

THE Bendix CORPORATION



RESEARCH LABORATORIES DIVISION
SOUTHFIELD, MICHIGAN

CONTRACT NAS 8-11928

Report No. 3371

HYDRAULIC FLUID INTERACTION SERVOVALVES

Monthly Technical Report

1 February 1966 - 1 March 1966

Submitted to

National Aeronautics and Space Administration
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Huntsville, Alabama 35812

by

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SECTION 1

INTRODUCTION

As a result of recent technological advances in the field of fluid state (no-moving-part) control systems, it appears that improved hydraulic control systems are attainable. The hydraulic servovalve characteristics and their effect on system performance and reliability are well known. The application of fluid state technology to the hydraulic servovalve offers the potential of increased reliability, nonelectrical signal summing and part cost reduction. The servovalve is the most complex, sensitive and fragile component in a hydraulic control system. Therefore, the servovalve represents the item most likely to demonstrate effectively the results of applying concepts of fluid state technology. The objective of this program is to develop a hydraulic servovalve, utilizing the vortex valve fluid state device as the primary flow control element. The resulting servovalve concept will be evaluated, and will be compared with present state-of-the-art servovalves, by installation on and test of a typical rocket engine gimballing actuator.

SECTION 2

GENERAL PROGRAM DEVELOPMENT

The development of the hydraulic vortex servovalve will result in a control element of increased reliability. The magnitude of the improvement can only be estimated at this time, based on such parameters as the ratio between minimum orifice and channel dimensions, elimination of silting or very small particle jamming of sliding surfaces, and resistance to mechanical and thermal distortion. To establish some basis for comparison, a conventional spool-type servovalve and the vortex servovalve could be tested side by side while exposed to various environmental conditions, contaminated oil, thermal gradients, etc. As a result of this effort, a numerical comparison of the relative reliability could be established.

An area of additional interest is the trend toward recoverable boosters. This requires hardware and control system components that will resist sea water corrosion if the booster is immersed during the recovery activity. A complete servo actuator, including a fluid state feedback sensor, can be readily constructed of various corrosion-resistant materials, such as naval bronze and the 300 series stainless steels. Since future control systems will utilize fuel as the hydraulic actuation fluid, the distinct possibility of contamination of the tanks, and subsequently the control valves, actuator, etc., during recovery operations, is present.

SECTION 3

ACCOMPLISHMENTS THIS PERIOD

Servovalve Design

Detail drawings of the servovalve have been completed. Final checking is in progress. A design review with NASA has been scheduled for early in the next reporting period.

Torque Motor Specification

During the past reporting period, D. G. O'Brien, Inc., was visited to discuss the torque motor output force and stroke requirements. It was determined that, by increasing the flapper length and stroke approximately 22 percent, an output force and stroke compatible with the allowable differential current could be obtained. Appendix A presents a detailed discussion of the changes. The torque motor specification was revised to reflect the changes and a purchase order was issued. The revised torque motor specification is shown in Appendix B.

Servovalve Environmental Tests

Discussions were held with local test facilities capable of performing the required servovalve vibration tests. Requests for quotation were issued to those outside sources qualified to perform the tests.

Vortex Valve Tests

A series of tests was initiated to determine single output vortex valve performance at the pressure levels to be encountered in the servovalve. To date, performance testing has been carried out primarily at 1000 psi because of the flow limitation of the original Bendix hydraulic power supply.

A hydraulic power supply capable of delivering 30 gpm at 3000 psi has recently been installed in the Bendix Hydraulic Laboratory. This power supply will permit testing at pressure levels required by the hydraulic vortex valve. A comparison with the normalized data from previous lower pressure tests will be made.

SECTION 4

PROBLEM AREAS

The torque motor output force and stroke incompatability with the available power has been resolved by increasing the flapper length and stroke approximately 20 percent. The required torque motor force and stroke can be obtained with the allowable differential current of 0.012 amperes available from the servo amplifier.

No other significant problem areas have been uncovered during the past reporting period.

SECTION 5

PLANS FOR NEXT PERIOD

Plans for the next reporting period, which ends 31 March 1966, include the following subtasks:

(1) Complete checking of servovalve detail drawings, review with NASA and release for manufacturing.

(2) Complete vortex valve performance testing at P_{supply} = 2300 psi.

(3) Estimate servovalve performance,using higher pressure level data.

SECTION 6

PROGRAM SCHEDULE

The program plan is shown in Figure 1 and indicates the various subtasks and the planned period of accomplishments.

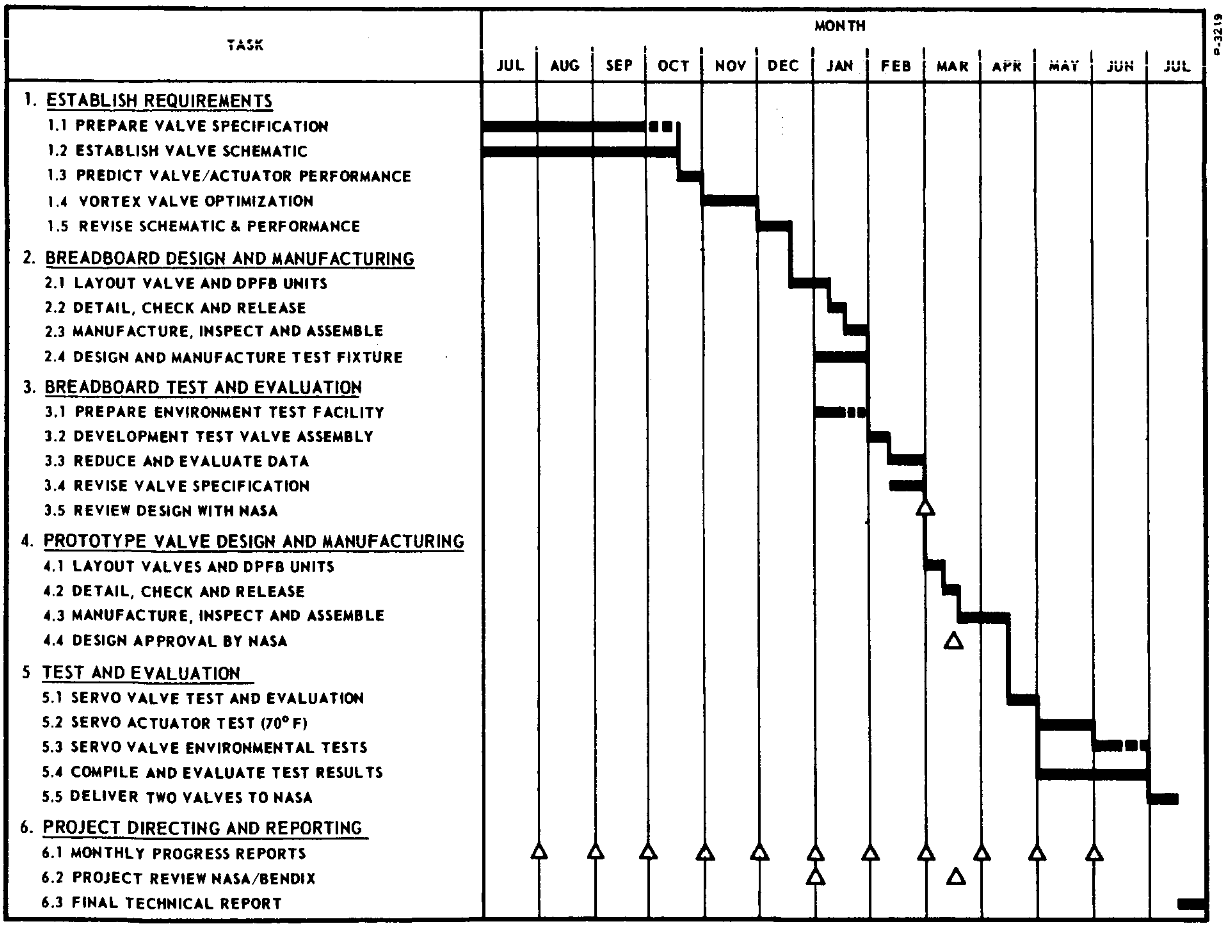


Figure 1 - Hydraulic Fluid Interaction Servovalve Program Schedule

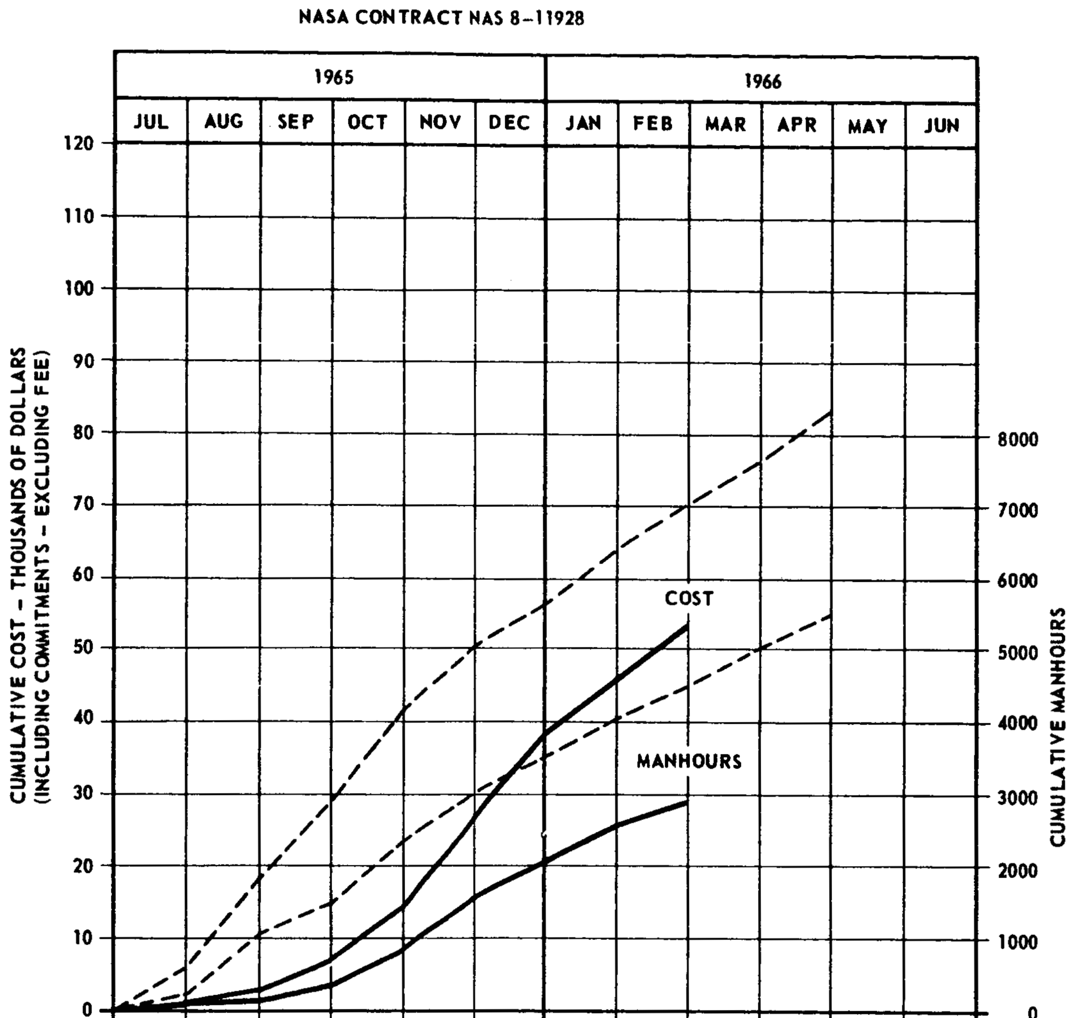
SECTION 7

MONTHLY FINANCIAL AND MANPOWER UTILIZATION REPORT

The cumulative manpower expenditures by category through 28 February 1966 are as follows:

	Hours
Engineering	1635
Technician	766
Miscellaneous	278
Shop	299

A graphic and tabular presentation of contract expenditures is given in Figure 2. It is anticipated that the contract work will be completed within the allocated funds; however, the scheduled period of accomplishement will be extended two (2) months, based on present estimates.



	JUL	AUG	SEP	OCT	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUN
FORECAST COST	6.71	18.26	29.69	41.88	50.88	57.70	64.09	70.89	76.80	83.45		
ACTUAL COST (EXCLUDING COMMITMENTS)	1.40	3.23	6.71	14.54	27.20	37.34	46.26	53.24				
CURRENT COMMITMENTS						.36	.07	.22				
TOTAL COST (INCLUDING COMMITMENTS)	1.40	3.23	6.71	14.54	27.20	37.70	46.33	53.46				
FORECAST MANHOURS	473	1118	1534	2437	3082	3572	4071	4561	5017	5533		
ACTUAL MANHOURS	72	193	410	906	1633	2071	2626	2978				

P-3219

Figure 2 - Hydraulic Fluid Interaction Servovalve Forecast and Actual Expenditures

APPENDIX A

TORQUE MOTOR FORCE REQUIREMENTS

APPENDIX A

TORQUE MOTOR FORCE REQUIREMENTS

In order to determine a torque motor output force and stroke that would be compatible with the allowable differential current, an analysis was made of the hydraulic spring rate of the reversed-flow flapper nozzle pilot stage. Figure A-1 presents the calculated force-deflection curve for different flapper lengths. The same flapper nozzle flow area was used for each length, and a constant vortex valve control port area downstream of the flapper nozzle was assumed. Therefore, for any flapper length, the same pressure would be obtained between the flapper nozzle and the control ports for equal (percentage-wise) values of stroke (i.e., for equal values of x/x_{max}, where x_{max} is the maximum stroke for the flapper length being considered).

In Figure A-1, the force-deflection curve used to specify the original torque motor design is designated by L = 0.870 inch. When the flapper length is 0.435 inch (see Figure A-1), the stroke, x, is halved, and, since the same flow area (πdx) is required, the nozzle diameter is doubled. The hydraulic force on the flapper is, therefore, increased by four (4). When the flapper length is 1.740 inch, the stroke is doubled, the nozzle diameter is halved, and the hydraulic force is reduced by four (4).

Since the negative hydraulic nozzle represents an unstable condition, a round cantilever spring device is attached to the flapper. Over the entire flapper stroke range, the cantilever spring rate should be of such a magnitude that the combined hydraulic and mechanical spring rates will always be positive. The torque motor, then, will always drive a net positive spring rate. In Figure A-1, a straight line representing the cantilever spring rate can be drawn next to any of the hydraulic spring rate curves. The difference between these curves is the net positive force that the torque motor must be capable of driving. The limiting slope of this line for each flapper length is the tangency point of the hydraulic spring rate curve. Since a large portion of the hydraulic force-stroke curve is linear, utilizing the tangent as the positive spring rate curve will cause the torque motor to drive a zero load over this portion of the curve. If the mechanical spring rate is not

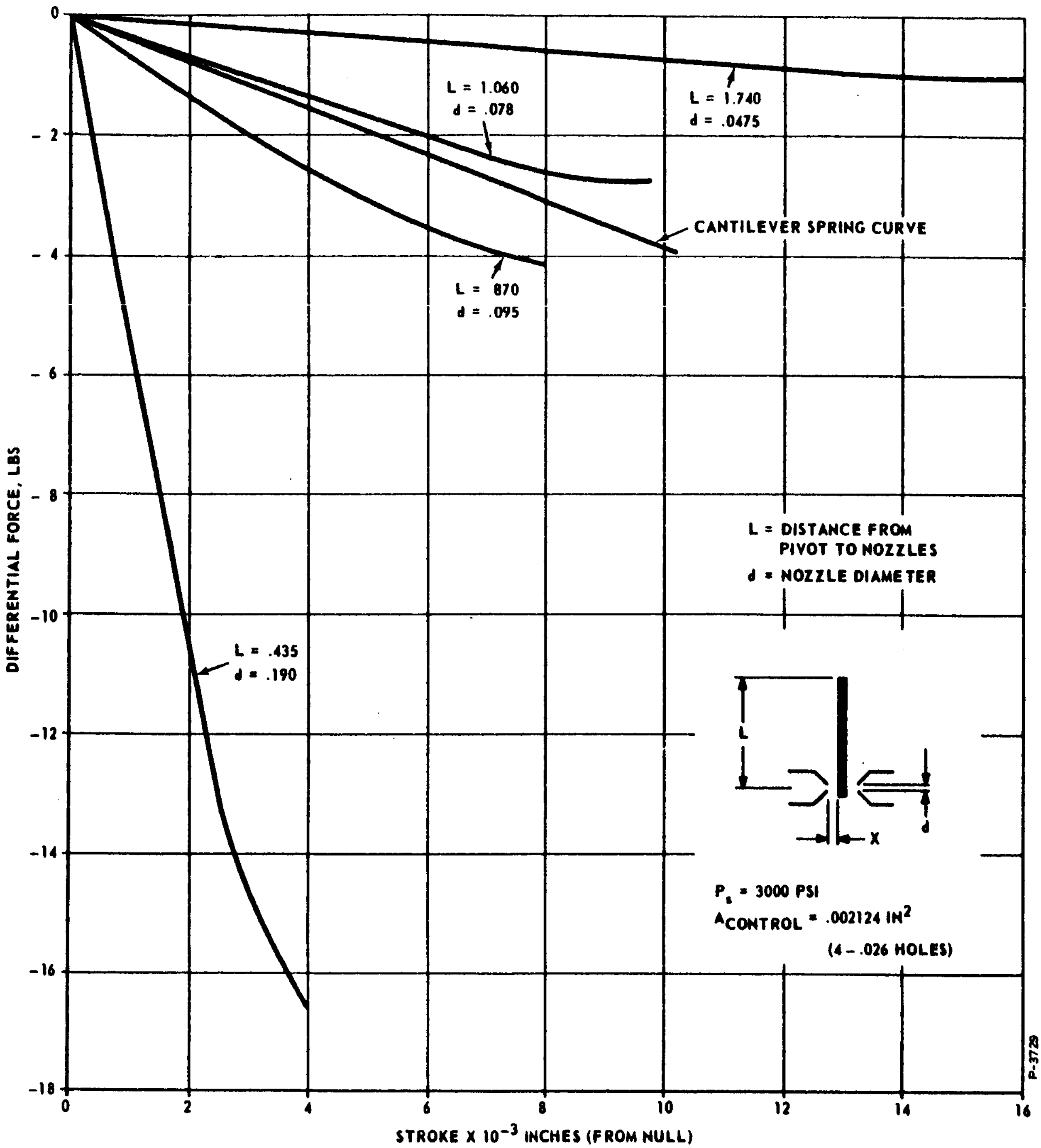


Figure A-1 - Hydraulic Spring Rate for Reversed-Flow Flapper Nozzle made tangent, the difference between the non-linear portion of the hydraulic spring rate curve and the cantilever spring rate curve increases rapidly, especially at full stroke.

If positive spring rate curves were drawn next to each hydraulic curve shown in Figure A-1, it would become apparent that, as the flapper length is increased, the difference between the two curves at full stroke would be reduced. However, although it is ideal to increase the flapper length, the flapper nozzle physical proportions become unrealistic. To maintain flapper nozzle flow control, the ratio of nozzle diameter to flapper stroke (from null) should never be less than 8. This will insure that at full stroke the flow area (πdx) of the unblocked nozzle is never greater than the nozzle circular area ($\pi d^2/4$). The curve of Figure A-1 for L = 1.060 inch represents this minimum d/x ratio. This increases the flapper stroke approximately 22 percent and results in a 40 percent decrease in force over the original design.

Technical discussions were held with D.G. O'Brien, Inc., to determine what maximum output force could be obtained with 0.012 amp differential current if the flapper length and stroke are increased approximately 22 percent in accordance with Figure A-1. D.G. O'Brien indicated that a 1-pound output force was realistic. As shown in Figure A-1, a cantilever spring force-deflection curve has to be drawn next to the hydraulic spring rate curve for L = 1.060 inch and d = 0.078 inch. This curve is drawn through the origin and through a point at maximum stroke (0.00975 inch) in such a way that the difference at full stroke is 1-pound. It can be seen that a small, positive, total spring rate is maintained over the entire stroke range.

The above changes were incorporated in the torque motor specification which appears in Appendix B and which was released as the X-4 revision. Since agreement was reached with D.G. O'Brien on the torque motor feasibility with these revisions, and since no other quotations were received, the purchase order was issued to D.G. O'Brien, Inc.

APPENDIX B

Torque Motor Specification

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2834-311		11272	DS-735	X-4

ENGINEERING SPECIFICATION

TITLE	DATE
Specification For An Electromagnetic Torque Motor	15 November 1965

This specification defines the requirements of a torque motor which will be used to drive the servovalve of a high performance hydraulic servo control system. The torque motor is to use construction techniques and materials suitable for use in a flyable primary control system.

1. Design

1.1 Type

The torque motor shall be a dry coil, permanent magnet polarized, two coil type with attached connector.

1.2 Weight

To be determined.

1.3 Installation

The torque motor shall conform to the space envelope, bolt pattern and output member as shown by Figure 1 in this specification.

1.4 Assembly

All threaded assemblies shall be positively locked to prevent loosening under vibration. Non-metallic adhesives shall not be used for assembly of the torque motor parts.

1.5 Connector

A connector shall be provided on the torque motor which will mate with Bendix Pygmy Connector PT06-8-4S.

1.6 Seals

The torque motor must be sealed to the servovalve body with standard MS "O" rings in appropriate grooves.

1.7 Amplifier

The amplifier is GFE and may be considered to be a constant current source.

PREPARED BY	CHECKED BY	APPROVED BY
T. A. Phillips		A. Pl...

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1.8 Coil Resistance

The resistance of each coil shall be 1000 ohms plus or minus 10%.

1.9 Polarity

The polarity of the unit shall be such that when the current in coil A-B is greater than the current in D-C the flapper motion is towards the connector end of the torque motor.

1.10 Coil Insulation

The insulation resistance between all pins connected together and case shall be greater than 50 megohms.

1.11 Coil Dielectric Strength

The coils shall withstand a voltage of 1000 vrms at 60 cps between coils and between coils and case.

1.12 Fluid

The unit shall be compatible with mil-o-5606 hydraulic fluid.

1.13 Internal Sealing

The output flapper of the unit shall contain a flexible seal capable of sealing system pressure of 3000 psi during normal operation.

1.14 Maximum Coil Current

The coil current shall not exceed 16.5 milliamps.

1.15 Quiescent Current

The quiescent current shall be 8.5 ± 2 milliamps per coil.

2.0 Environmental Conditions

The torque motor mounted to the servovalve will be subjected to the following environmental conditions in any combination and the torque motor shall be designed to meet the requirements of this specification during such exposure.

2.1 Temperature

The torque motor must operate within the specified performance envelope of Figure 2 throughout the temperature range of 0°F to +150°F immediately following a 4 hour soak at the selected temperature.

2.2 Vibration

The torque motor must withstand the vibration schedule of Procedure II of Mil-E-5272 under a non-operating condition with the servovalve filled with oil. All vibration schedule time shall be 60 minutes at 70°F. The torque motor shall meet the performance after being subjected to the above vibration schedule.

In addition, the torque motor will be subjected to a vibration schedule as follows during which a frequency response test of the servovalve will be conducted.

5 cps at 0.5 inch D.A.
10 cps at 0.5 inch D.A.
110 cps at 0.020 inch D.A.
500 cps at 0.012 inch D.A.

2.3 Shock

The torque motor shall operate within the specification performance after being subjected to a 40g shock of 11 millisecond duration in any direction.

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2834-311		11272	DS-735	X-4

ENGINEERING SPECIFICATION

TITLE	DATE
Specification For An Electromagnetic Torque Motor	15 November 1965

2.4 Altitude

The torque motor must operate from sea level to 300,000 ft. altitude within the performance specification. All electrical leads, connection, and coil construction shall be designed such that altitude induced electrical leakage will not effect the torque motor performance.

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2834-311		11272	DS-735	X-4

ENGINEERING SPECIFICATION

TITLE	DATE
Specification For An Electromagnetic Torque Motor	15 November 1965

3. Performance

Characteristic	Unit	Valve
Stroke	in	±.00975
Mid-position	lb	2.0
End of Stroke	lb	1.0
Hysteresis	%	<2
Resonant Frequency	cps	>100
Differential Current, Δ i (Range)	amp	±.012

Note: Torque motor output force requirements are shown in Figure 2.

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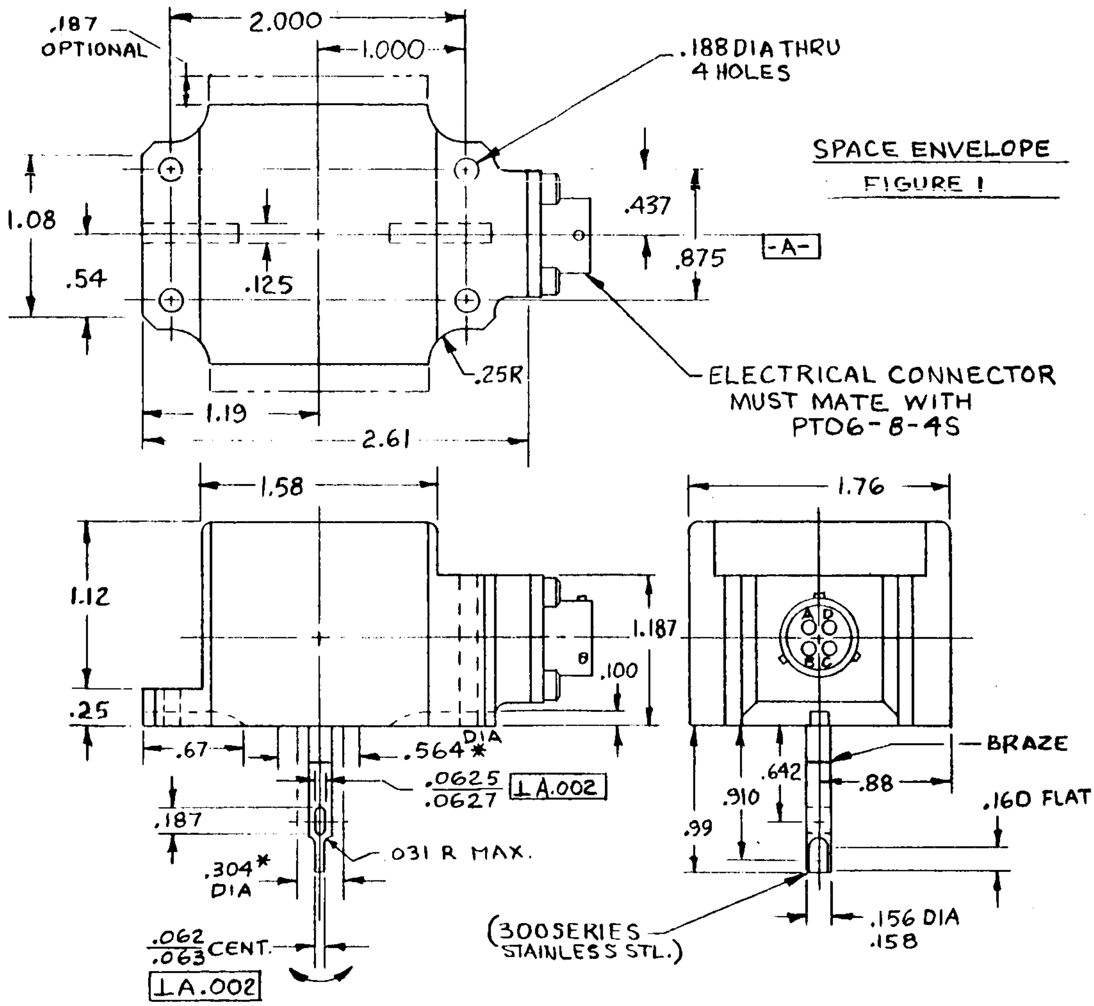
PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2834-311		11272	DS-735	X-3

ENGINEERING SPECIFICATION

TITLE	DATE
SPEC. FOR ELECTROMAGNETIC TORQUE MOTOR	1-28-66



* "O" RING SEAL SURFACE 32√. GROOVE TO BE PROVIDED IN MATING PART

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R. P. Jacobs		T. A. Phillips

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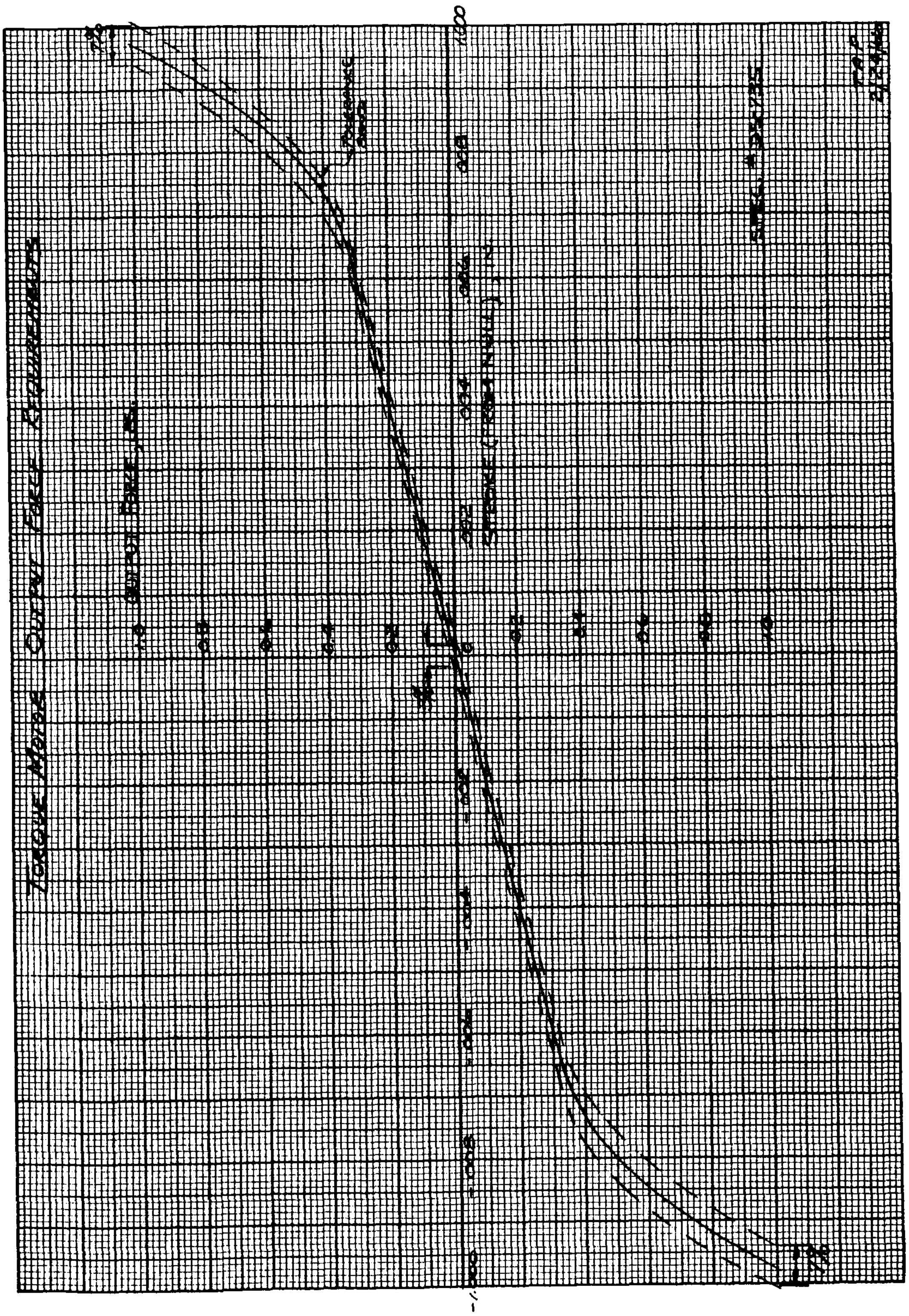

FIGURE 2