https://ntrs.nasa.gov/search.jsp?R=19730016370 2020-03-11T19:47:21+00:00Z

N73-25097

NASA TECHNICAL MEMORANDUM

NASA TM X-2812

NASA TM X-2812

CASE FILE COPY

IMPROVED DESIGN OF A HIGH-RESPONSE SLOTTED-PLATE OVERBOARD BYPASS VALVE FOR SUPERSONIC INLETS

by John A. Webb, Jr., Oral Mehmed, and Kirby W. Hiller
Lewis Research Center
Cleveland, Ohio 44135

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • JUNE 1973

1. Report No. NASA TM X-2812	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle IMPROVED DESIGN OF A HIGH-RESPONSE SLOTTED-PLATE OVERBOARD BYPASS VALVE FOR SUPERSONIC INLETS		5. Report Date June 1973
		6. Performing Organization Code
7. Author(s) John A. Webb, Jr., Oral Mehmed, and Kirby W. Hiller		8. Performing Organization Report No. E-7399
		10. Work Unit No. 501-24
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.
		13. Type of Report and Period Covered Technical Memorandum
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D.C. 20546		14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

The electrohydraulically actuated slotted-plate bypass valve used to control the position of the normal shock during wind-tunnel investigations of supersonic inlets also has proven to be a valuable device for determining inlet dynamics and creating airflow disturbances. Operation of previous valves at high frequencies (to 100 Hz) for extended testing has resulted in numerous failures. This report presents an improved bypass-valve design which increases the cyclic tolerance of the device considerably over past designs. The use of dynamic limit criteria to obtain an optimum actuator-piston size results in a frequency response which is flat within ±3 decibels to 120 Hz for a peak-to-peak variation of 20 percent of full area.

17. Key Words (Suggested by Author(s)) Electrohydraulic servosystems; Fast-response systems; Supersonic inlet; Servosystem	18. Distribution Statement Unclassified - unlimited

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	23	$3.00

* For sale by the National Technical Information Service, Springfield, Virginia 22151

IMPROVED DESIGN OF A HIGH-RESPONSE SLOTTED-PLATE OVERBOARD BYPASS VALVE FOR SUPERSONIC INLETS

by John A. Webb, Jr., Oral Mehmed, and Kirby W. Hiller

Lewis Research Center

SUMMARY

The electrohydraulically actuated slotted-plate valve has proven valuable as a fast-response pneumatic valve for use as a disturbance or control device during wind-tunnel investigations of supersonic inlets. These valves have also made it possible to determine the pneumatic dynamics of the inlet over the frequency range from 0 to 100 hertz.

However, high cyclic use and acceleration forces which are the result of obtaining a high-response system have created failures in previously used valves. These failures included shearing of mounting bolts, wear of actuator-piston linkage resulting in backlash and eventual linkage failure, and damage to the actuator due to overtravel of the piston.

The improved design presented in this report minimizes all of the previously experienced problems. The new design incorporates a doweled actuator block, direct coupling of the actuator and valve moving element with a rod that has flexures to allow for misalinement, and hydraulic cushions at the ends of the piston travel. The dynamic performance is also improved by means of analytical response limit criteria which results in an optimum piston area.

Results indicate that the valve can operate for over 8 million cycles without any failures or degradation in performance. It can tolerate overtravel of the actuator piston for more than 4000 cycles with only a slight amount of piston-seal damage due to the hydraulic cushion orifices. The servosystem dynamic response is flat within ±3 decibels to 120 hertz for a peak-to-peak displacement of 20 percent of full stroke.

INTRODUCTION

Fast-response slotted-plate pneumatic valves have been used on research supersonic engine inlets to control the position of the normal shock in the inlet by bypassing excess inlet airflow (ref. 1). This high response capability makes bypass valves suitable as disturbance devices for control studies or for determining inlet dynamics over the range of frequencies from 0 to 100 hertz. Two such devices are described in references 2 and 3. It is possible to design and build such valves with accurately predictable dynamic-response characteristics as reported in references 4 and 5.

However, it has been learned (ref. 2) that the fast response capability of these devices predisposes them to a variety of failures. These failures are ascribable to the large acceleration forces and high cyclic use that result from their fast response capability. This might imply that such fast-response devices are inherently failure-prone and should only be used in short-duration experimental applications. However, a slotted-plate valve system that exhibits considerably improved reliability has been developed through a succession of design improvements. The purpose of this report is to discuss the problems which created failures in earlier designs, explain the improvements in the new design, and show the performance characteristics and the damage tolerance of a prototype valve of the new design.

PREVIOUS BYPASS VALVE DESIGN

The configuration of the earlier bypass valve design reported in reference 2 is shown in figure 1. A schematic representation of the valve and its servo control loop is presented in figure 2. These figures show how the plate valve was used to bypass air from the inlet in response to linear motion of the servoactuator. Closed-loop control of

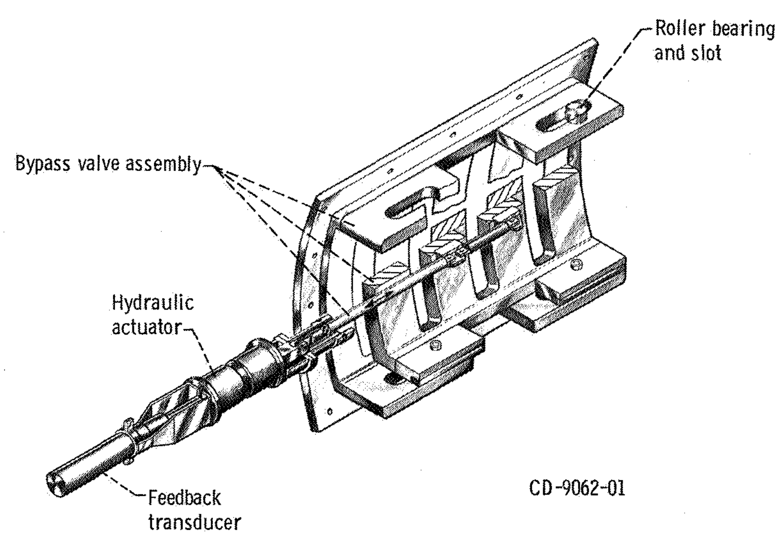


Figure 1. - Previous bypass valve design.

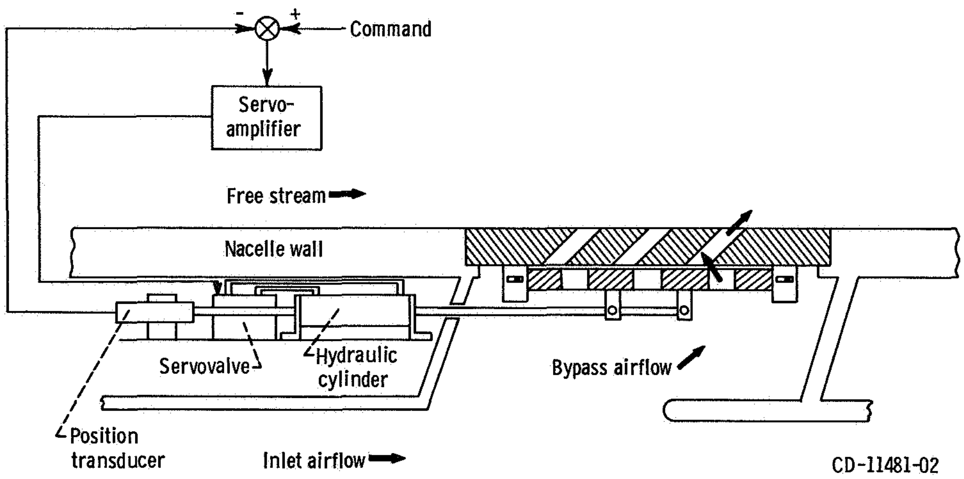


Figure 2. - Schematic of previous bypass valve servosystem.

the system was maintained with the electronic servo loop as explained in reference 2. The moving element of the valve was supported by cam-follower roller bearings which ran in slots in the valve stationary element. When the valve was closed, the pressure difference was in such a direction that it forced the two plates together. The movable plate was designed with sufficient thickness to prevent metal-to-metal contact due to the pressure loading. The spacing between the plates was about 0.16 centimeter. Teflon seals (not shown) of triangular cross section were mounted in the stationary element to reduce the valve leakage flow when it was fully closed.

The servovalve, actuator, and slotted-plate bypass valve were mounted separately on the inlet. The commercial piston-in-cylinder actuator was mounted with brackets to the inlet structure. Its moving element consisted of a two-piece steel shaft and an aluminum piston. It was connected to the moving plate with a pinned thrust link. Hydraulic oil was supplied to the actuator from a commercial, two-stage, electrohydraulic servovalve. Piston position was determined by means of a linear variable differential transformer. The full flow area, stroke, moving-element mass, and physical dimensions of this valve are indicated in table I. The valve exhibited a dynamic response which was flat to within ±3 decibels up to a frequency of 110 hertz (with a phase lag of 160° at 110 Hz) for a peak-to-peak amplitude of 14 percent of full stroke.

Various problems developed during extensive use of this valve at high cyclic frequencies. Some of the actuator's two-piece shafts came unscrewed, some aluminum pistons became loose, some of the actuators broke loose from their mounting bolts, backlash developed in all of the thrust link pins, most of the thrust links failed, and several of the hydraulic tubes cracked near their fittings. These problems resulted in the need for frequent repairs and caused delays in wind-tunnel test programs.

TABLE I. - PHYSICAL PROPERTIES OF BYPASS VALVES

Dimension	Previous valve	New valve
Actuator stroke, cm	2.54	[a]5.39
Slot length, cm	15.9	20.5
Slot width, cm	2.54	2.54
Number of slots	4	8
Valve full flow area, cm^2	161	401
Valve area to position gain, cm^2/cm	63.4	158
Mass of moving parts, kg	2.75	3.94

[a]Includes travel required for hydraulic cushion. "Full stroke" referred to in text is the 2.54 cm needed to stroke from full open to full closed.

The position transducer was mounted directly to the actuator shaft. This avoided the limit cycle oscillations which would have occurred if the position transducer had been connected directly to the valve moving element when backlash was present in the linkage. Such oscillations would have increased the rate of deterioration of the actuator and linkages. However, with the position transducer mounted to the actuator shaft, it was not possible to tell, in the control room, when the linkages failed.

An additional problem was created when the Teflon seals on the slotted plate became worn from repeated use, resulting in a changing leakage flow area. The use of seals also posed a difficulty in the assumption that valve area (which is the desired variable to control) is proportional to valve position. Depending on the geometry of the seals used, some nonlinearity may be introduced into this proportionality near valve closure. Difficulties also arose from conflicting requirements in the mounting of the seals. It was desired that the seals be easily replaceable, but an air-tight seal with the surface to which they were mounted was also required.

During operation of the inlet bypass valve, the actuator piston can be driven to the limit of its travel by accidentally applying an excessive command signal, opening of the feedback loop, excessive gain, or applying positive feedback. The earlier design relied on the mechanical travel limit of the actuator piston as the limit of total valve travel. This created excessive deceleration forces upon impact of the piston, which damaged both the actuator and the valve.

Most of these problem areas are due to the high response characteristics inherent in the valve design. When the previous valves were designed, the possibility that they would be used so extensively for high-frequency applications had not been anticipated. With a carefully redesigned valve, these problems can be reduced or eliminated, which results in a more reliable servosystem that could eventually be adapted as practical flight hardware.

IMPROVED BYPASS VALVE DESIGN

The need for additional bypass valves to continue tunnel testing of supersonic inlets led to a redesign of the slotted-plate valve and actuator to improve reliability without degrading dynamic performance. This section describes the improved design, with emphasis on the features unique to this valve which eliminate previous problems.

Figure 3 shows the improved bypass valve, actuator piston, servovalve, and feedback transducer. All components of the valve are mounted on the stationary element, which acts as a base for the entire valve assembly. This allows bench assembly and calibration, since the entire valve assembly can be removed and installed as one unit. A rod with integral flexures is used to couple the piston to the moving valve element. The moving element is supported by commercially available, crossed-axis, linear roller bearings to provide precise motion with negligible friction. A second transducer mounting is provided for a linear potentiometer to be used as a position indicator during steady-state tests.

The improved bypass valve is designed primarily for use as a control device for

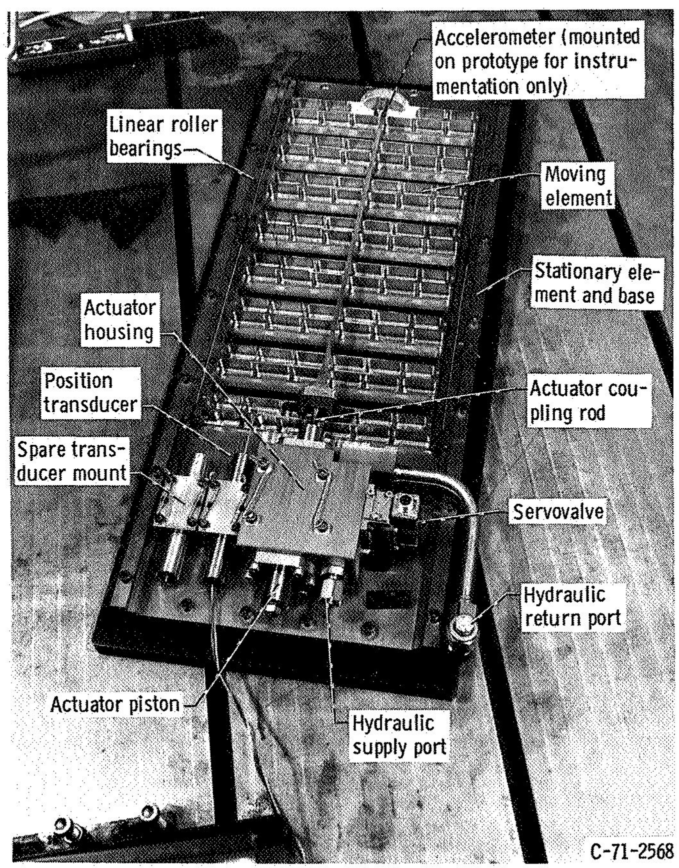
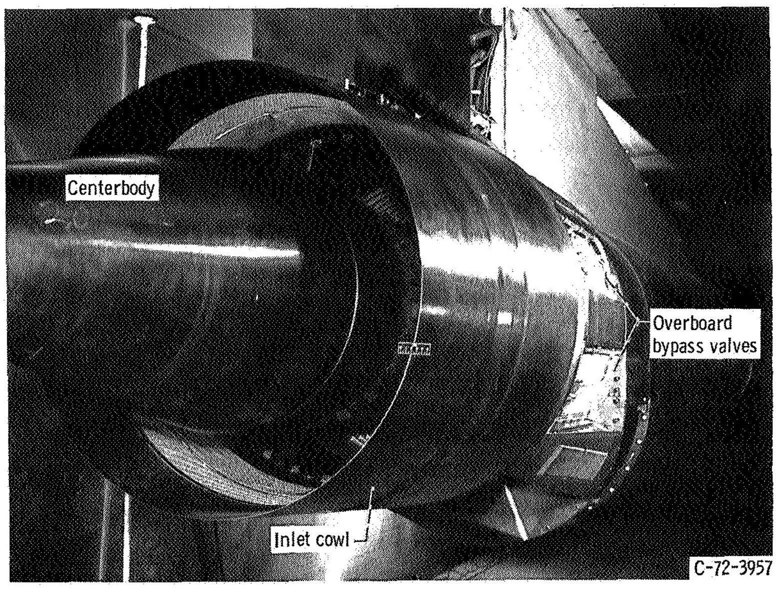
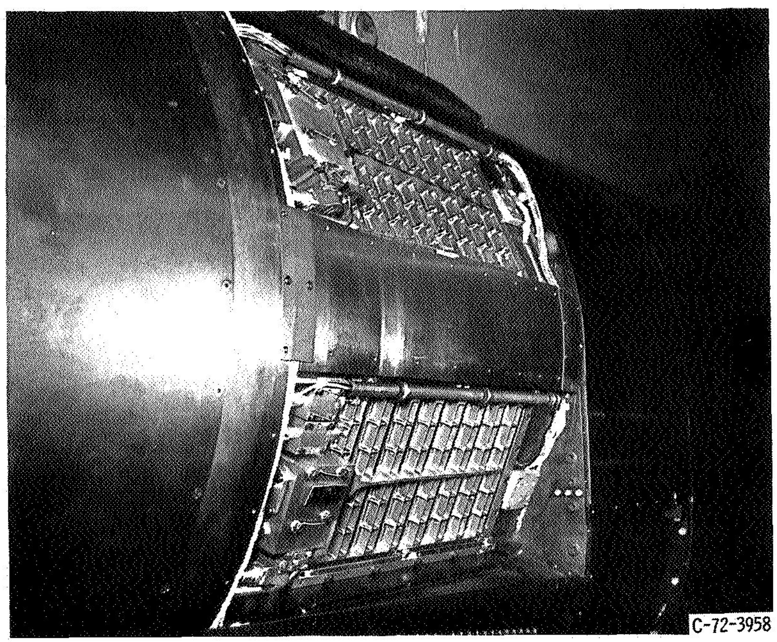


Figure 3. - Improved overboard bypass valve for supersonic inlets.

stabilizing supersonic inlets. Two typical applications of this valve are shown in figure 4. Figures 4(a) and (b) show an inlet that has eight of these valves around its periphery. In this configuration, the valves can be used for shock position control or as a source of internal airflow disturbances. Figure 4(c) illustrates the installation of five valves which were used to create a downstream airflow disturbance. This photograph was taken looking upstream into a duct mounted on the aft end of an inlet model which

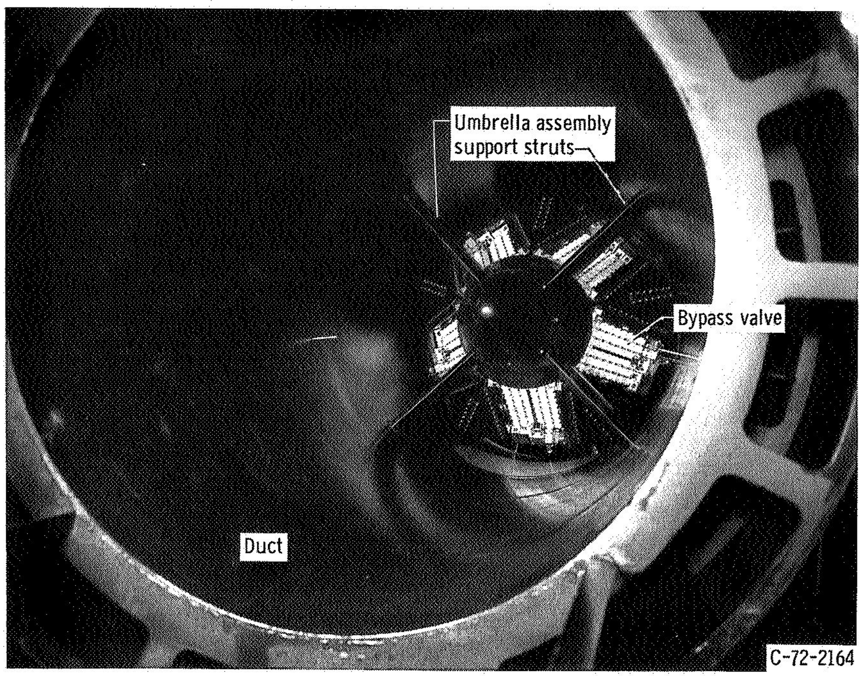


(a) Inlet with valves installed for bypass application.

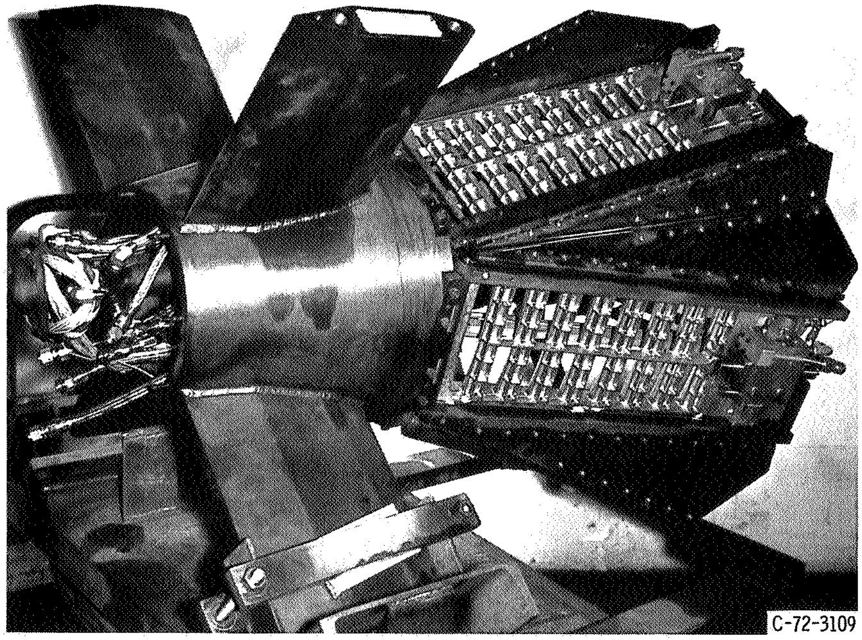


(b) Close-up view of valve installation on inlet.

Figure 4. - Typical applications of improved bypass valve.



(c) Installation as downstream disturbance generator.



(d) Close-up view of disturbance assembly.

Figure 4. - Concluded.

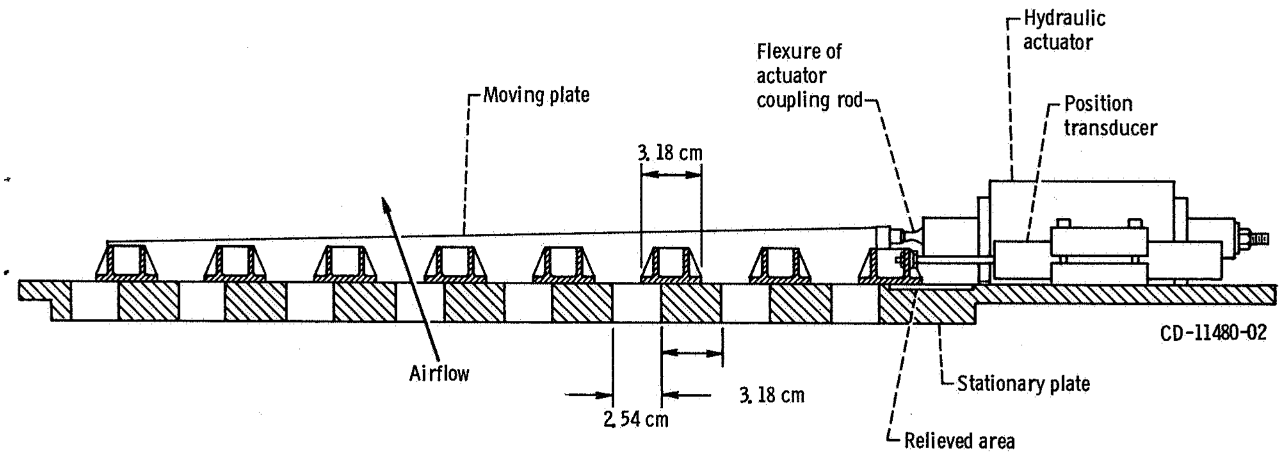


Figure 5. - Section view of improved overboard bypass valve.

was being tested. The valves are mounted on an umbrella type assembly which can open or close to various positions to obtain a desired amount of blockage. The bypass valves are then actuated to create disturbances in the amount of blockage. The unit is shown in the fully expanded position. A close-up view of the disturbance assembly is provided in figure 4(d).

A section view of the bypass valve is shown in figure 5. All seals between the moving and fixed element have been eliminated and leakage with the valve closed is controlled by the clearance between the plates. The two plates were specified flat and parallel to each other within 0.005 centimeter, and the clearance between the plates was set to 0.008 centimeter. This minimizes leakage flow during valve closure while assuring that no metal-to-metal contact occurs during operation. Pressure loading due to airflow is in the direction which tends to open the clearance gap further. The structure is designed for a 0.012-centimeter maximum deflection at midspan with the valve closed and with a pressure differential of 7.9 newtons per square centimeter across the valve. The lands of the valve are 0.64 centimeter wider than the slots to allow for an overlap of 0.32 centimeter at each edge when in the fully closed position. To maximize its structural strength and stiffness and to minimize its mass, the moving element was milled out of a plate, with stiffening ribs left in the material. The center rib, which connects to the actuator, helps to stiffen the structure and transmits the acceleration loads throughout the moving plate. Both plates were fabricated with 17-4 PH stainless steel hardened to a Rockwell C hardness of 35 to 42. Additional physical properties for this valve are presented in table I.

The linear roller bearings used to support the moving element of the bypass valve are shown in figure 6. The crossed-axis design of these commercially available bearings gives a precise linear motion with negligible friction. The end screws are used to

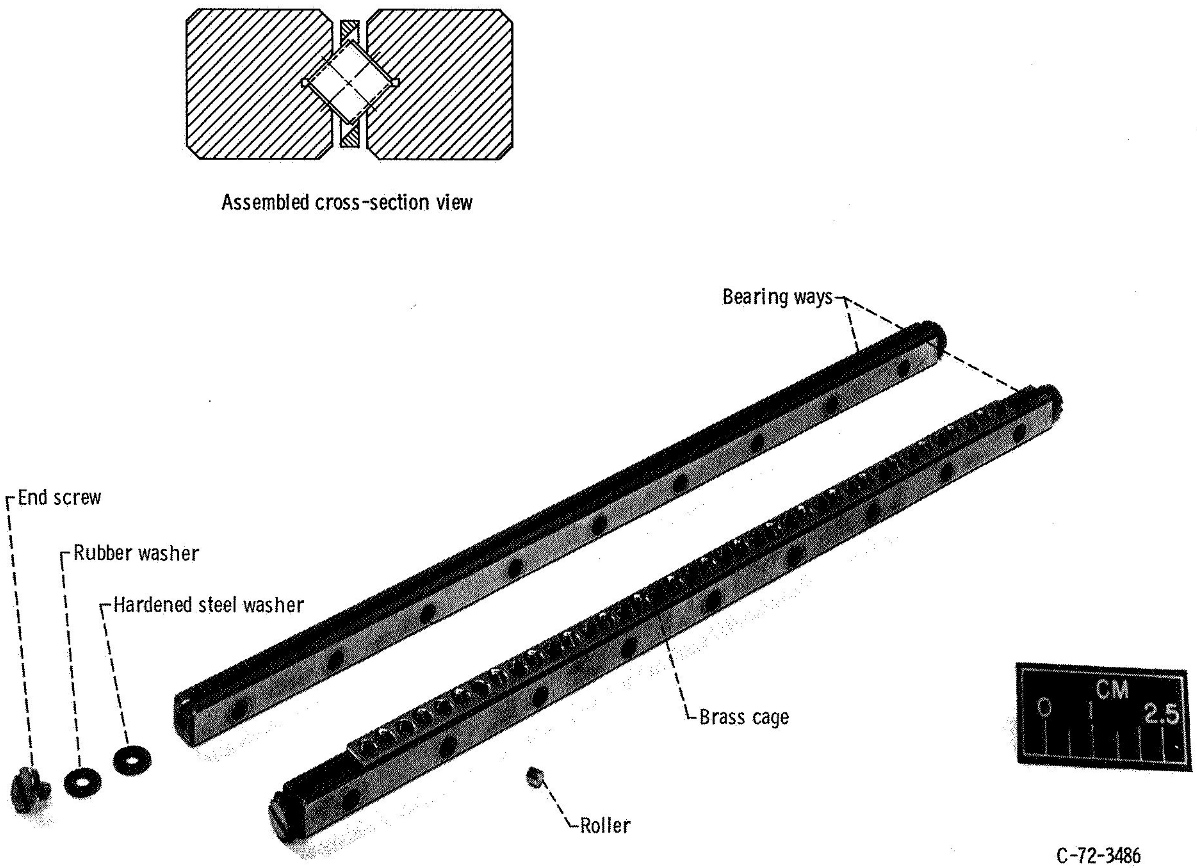


Figure 6. - Crossed-axis linear roller bearings used for bypass valve.

prevent disassembly of the bearing. During high-frequency motion, the rollers and cage work to one end of the bearing way and the end roller hits the end screw head repeatedly. To prevent the possibility of damage and failure of the end screws, a hardened steel washer is inserted between the screw head and the bearing way. A rubber washer is also placed between the screw head and hardened washer to provide some elasticity.

The actuator used to drive the bypass valve is shown disassembled in figure 7. The servovalve is mounted directly to the actuator block to minimize the length of the hydraulic line from the servovalve to the piston cylinder ports. This decreases the working-fluid volume and, therefore, keeps the hydraulic natural frequency (ref. 6) of the system beyond the range of interest (0 to 200 Hz). The hydraulic natural frequency for this actuator is 370 hertz. The servovalve is also mounted in such a way that its spool motion is perpendicular to the motion of the actuator piston. This eliminates the possibility of mechanical vibrations feeding back to the servovalve in a manner which

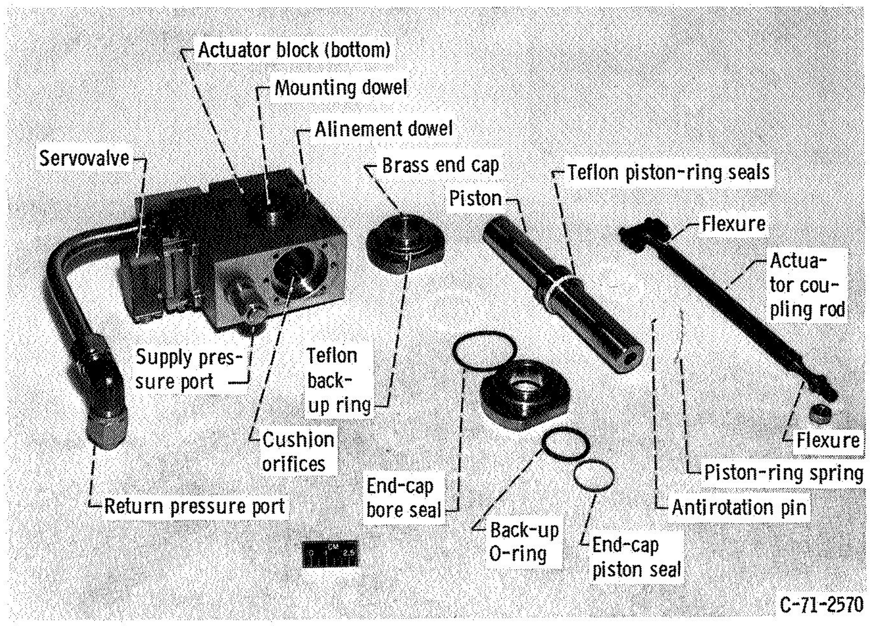


Figure 7. - Bypass valve actuator - dissassembled.

could cause instability during closed-loop operation.

The actuator block in figure 7 is shown with the bottom exposed to show the large dowel which is machined as a part of the block. The dowel fits into a hole in the base plate of the bypass-valve assembly to provide a secure mounting which eliminates shear on the actuator hold-down bolts. The small dowel is for positive alinement. The brass end caps for the actuator cylinder were designed with sufficient thickness to withstand the forces caused by piston impact and hydraulic overpressures. They seal to the cylinder bore with an O-ring and a Teflon back-up ring. The end caps seal to the piston with a Teflon seal which is backed up by an O-ring. The end caps provide the bearing surface for the piston. The bearing surfaces are lubricated with the working hydraulic fluid.

The piston has two pairs of Teflon piston rings for sealing to the actuator block bore. Seal pressure against the bore is maintained with metal springs which are placed under the piston rings as shown in figure 8. Rotation of the seals is prevented by antirotation pins. The piston is relieved between the seals to avoid the problem of unbalanced pressure forces causing binding and excessive seal wear when the piston travels over the cushion area.

To prevent damage from piston overtravel, a cushion in the form of a trapped volume of hydraulic fluid is obtained by porting the control flow from the servovalve through a series of graduated orifices on the actuator cylinder walls as shown in figures 7 and 8. If a malfunction occurs and the piston travels past its normal range of

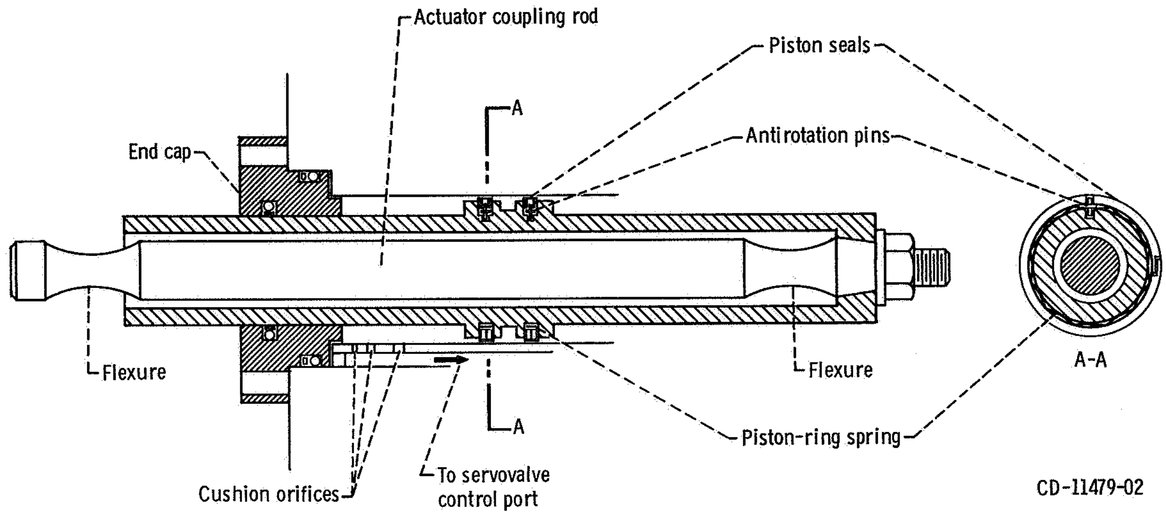


Figure 8. - Section view of assembled actuator.

operation, the piston-ring seals cross over these orifices, thus decreasing the flow area available for escaping hydraulic fluid. This decelerates the piston and reduces the high forces caused by driving the piston into the end cap. The antirotation pins in the piston assure that the seams of the piston rings do not line up with the cushion orifices. If the piston ring seams were allowed to pass across the orifices, the edges of the seams would be pressed into the orifices, and the piston lands would act as a shearing device, cutting away seal material on the orifice edges. This does not occur when antirotation pins are used, as will be demonstrated later.

The actuator coupling rod eliminates any pinned linkage between the acutator and the moving element. It is mounted within the hollow piston to allow close spacing between the valve and the actuator. One end of the rod is attached to the piston with a taper seat and self-locking nut (fig. 8). The other end has an integral flange that is attached with screws to the moving plate of the bypass valve. This rod is of circular cross section and is designed with a necked-down portion (flexure) near each end, which allows it to flex at slight angles to its centerline with relatively little force (a force of 3 N applied at the rod ends perpendicular to the rod axis can displace the mounting end of the rod 0.01 cm). However, the rod is also designed for high stiffness in the direction of valve motion and can withstand loads up to 13,400 newtons along its axis. This design allows for misalinements and deflections between the actuator and the valve moving element without causing high forces or wear in any component.

As can be seen from this discussion, the improved design eliminates most of the difficulties encountered with the earlier designs. Improvements such as the dowel pin in the actuator block, direct mounting of the servovalve, hydraulic piston cushions, and an actuator coupling rod with integral flexures increase the valve's ability to provide the endurance required for extensive usage and eliminates the control difficulties encountered with the earlier designs.

PREDICTED DYNAMIC PERFORMANCE

The dynamic performance envelope of the improved bypass valve can be predicted with the use of the technique described in reference 4. This reference derives three basic limitations on system performance: piston velocity limit, piston acceleration limit, and flapper flow limit. These limits were modified in reference 2 and applied to predict the performance of the earlier bypass valve design. Reference 5 presents an improved derivation of these limits with the use of an elliptical maximum power transfer envelope. A schematic of the servovalve is shown in figure 9.

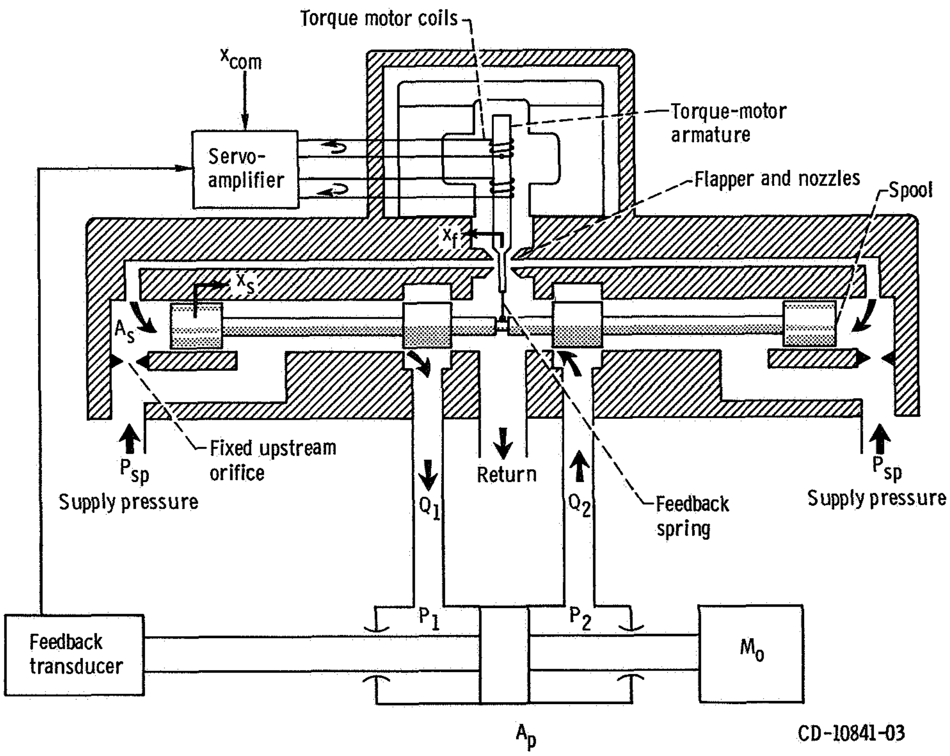


Figure 9. - Schematic representation of servovalve, actuator, and load.

The resulting relations are as follows:

Velocity limit

$$X_o \omega = \frac{\sqrt{2}\, Qr}{A_p} \tag{1}$$

Acceleration limit

$$X_o \omega^2 = \frac{2\sqrt{2}\, A_p P_{sp}}{3\, M_o} \tag{2}$$

Flapper flow limit

$$X_o \omega^2 = \frac{\sqrt{2}\, Qr\, q_{f(max)}}{x_{s(max)} A_s A_p} \tag{3}$$

Symbols are defined in the appendix.

The values used for the variables on the right of these equations are listed in table II. These result in straight-line plots of actuator displacement X_o as a function of frequency ω as shown in figure 10. The figure shows that the response is limited by the physical limits of the actuator for low-frequency operation, by the velocity limit for intermediate frequencies, and by the flapper flow limit for high frequencies. The intersection of the velocity limit and flapper flow limit lines can be derived as

$$\omega' = \frac{q_{f(max)}}{x_{s(max)} A_s} \tag{4}$$

TABLE II. - PHYSICAL CONSTANTS FOR IMPROVED BYPASS VALVE SERVOSYSTEM

Actuator piston area, A_p, cm^2	2.58
Spool end area, A_s, cm^2	0.342
Equivalent actuator and output load mass, M_o, $(N)(sec^2)/cm$	0.0394
Hydraulic supply pressure, P_{sp}, N/cm^2	2070
Maximum Flapper nozzle flow, $q_{f(max)}$, cm^3/sec	5.74
Servovalve rated flow, Q_r, cm^3/sec	505
Maximum spool displacement, $x_{s(max)}$, cm	0.0445

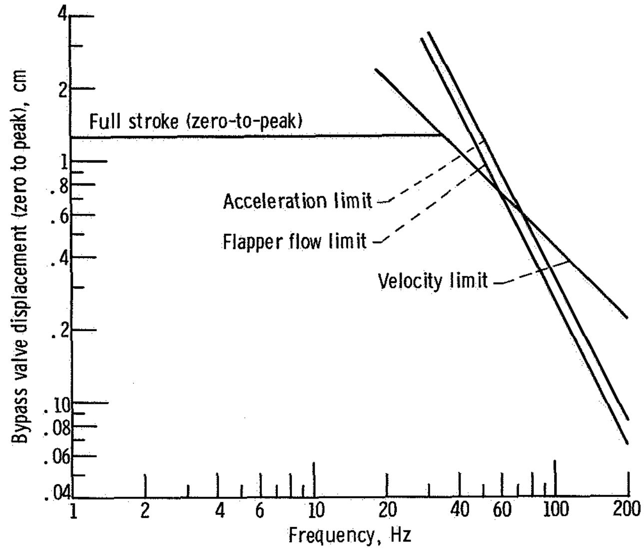


Figure 10. - Performance limit lines for improved bypass valve.

$$X'_o = \frac{\sqrt{2}\, Q_r A_s x_{s(max)}}{q_{f(max)} A_p} \tag{5}$$

In figure 10, this intersection occurs at a frequency of 60 hertz and a zero-to-peak displacement of 0.74 centimeter.

Reference 5 also derives the following relation for determining an optimum actuator piston area:

$$A_p^* = \sqrt{\frac{3\, M_o Q_r q_{f(max)}}{2\, P_{sp} A_s x_{s(max)}}} \tag{6}$$

If the piston area were exactly optimum, the flapper flow limit and acceleration limit would coincide (see ref. 4). Using the values in table II gives an optimum area of 2.33 square centimeters. The actual piston area listed in table II is slightly larger than optimum at 2.58 square centimeter; however, this is very close and does result in nearly the best possible frequency response obtainable with this system. (Exact optimum piston area was not chosen because of seal sizing considerations.)

EXPERIMENTAL PERFORMANCE

The improved bypass valve was tested to verify its dynamic frequency response capability and to check its ability to withstand the high cyclic requirements needed for wind-tunnel experimentation. It was also tested to determine its ability to tolerate the actuator piston being driven into the end stops. Testing was performed with a prototype valve securely mounted on a rigid structure. An accelerometer was mounted on the moving plate as, shown in figure 3.

The schematic diagram of the new bypass valve control system is similar to that shown in figure 2. The servoamplifier is described in reference 7. No compensation was added at the preamplifier of the servoamplifier to improve the system response. Since the frictional forces are negligible, no dither was used and no hysteresis was observed.

Dynamic Response

Dynamic response data were obtained with the servoamplifier gain adjusted to achieve a desired step response. Figure 11 gives a typical step response adjusted for 10 percent overshoot for a step amplitude of 20 percent of full stroke. The experimental frequency response data are plotted in figure 12 for the gain adjusted for 10 percent overshoot. Included on the plot are the limit lines determined in the previous section. This figure shows the ability of the flapper flow limit relation to predict the upper bounds of the frequency response. For a peak-to-peak displacement of 1.8 centimeters (70 percent of full scale) the -3 decibel point is at 60 hertz with a phase lag of 142°. At

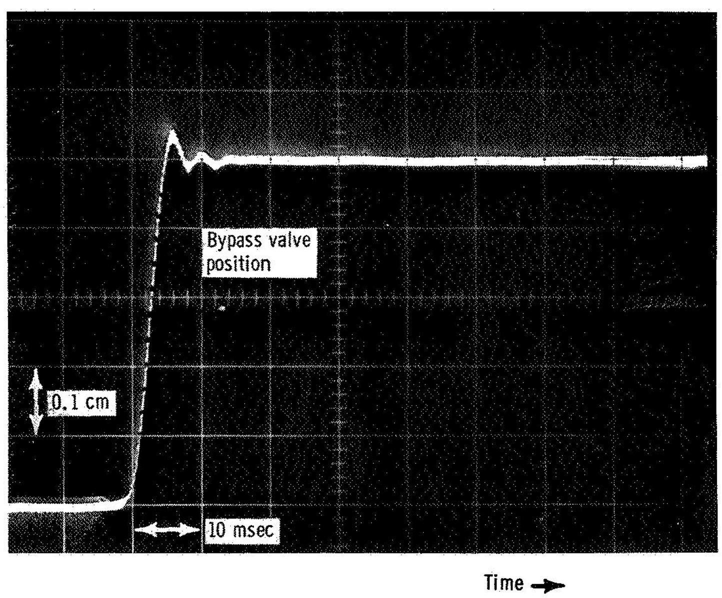


Figure 11. - Step response of bypass valve for a 20 percent of full stroke step.

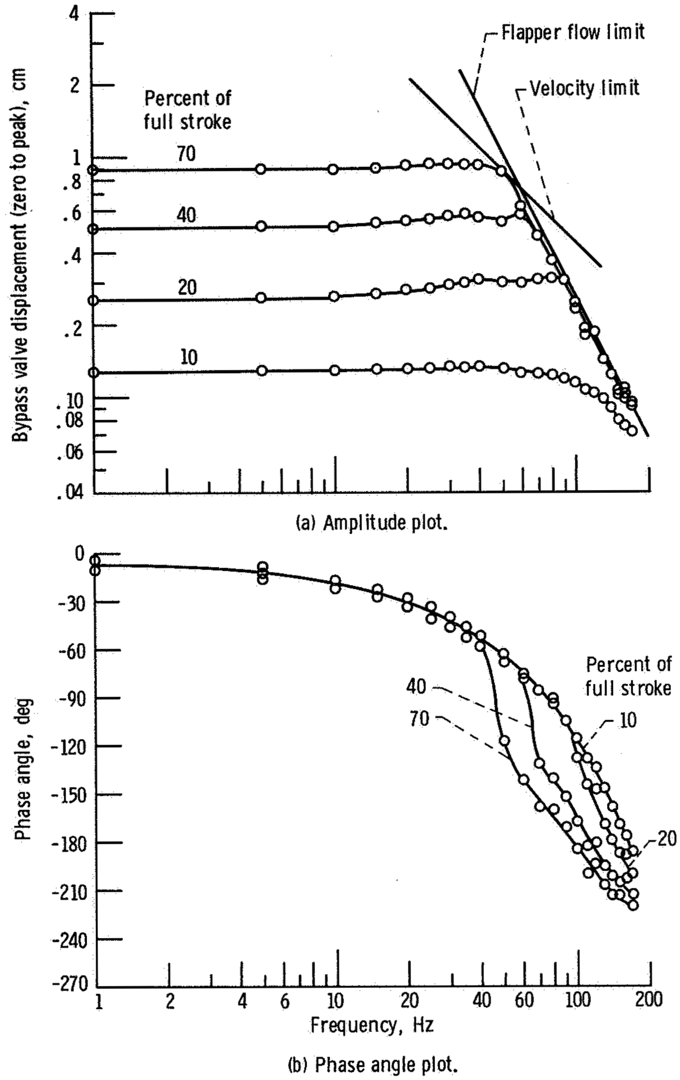


Figure 12. - Frequency response of improved bypass valve for servosystem gain adjusted to obtain 10 percent step overshoot.

a peak-to-peak displacement of 0.50 centimeter (20 percent of full scale), the -3 decibel point is at approximately 120 hertz and the phase lag is 157°. For smaller amplitudes, the linear closed-loop response does not reach the limit lines, as can be seen from the data for a peak-to-peak displacement of 0.26 centimeter (10 percent of full stroke).

It should be mentioned at this point that galling between the two valve plates was observed after running large-amplitude (40 and 70 percent of full stroke) responses. Galling occurred under the center of the first cross beam on the actuator side of the first slot. Since the actuator coupling rod is attached above and to the rear of the first cross beam (see fig. 5), it applies a twisting moment on that beam. Twisting apparently was sufficient during large amplitude actuation to cause the beam edge closest to the actuator to touch the base. This problem was remedied by relieving the base plate in the vicinity of the galled area. The relieved portion was a 10- by 4-centimeter rectangle centered under the piston rod and 0.3 centimeter from the first slot. No further galling appeared during additional testing.

Endurance Testing

Cyclic life test. - Because of its high response characteristic, this bypass valve will experience a large number of cycles when used for tunnel experimentation. To assure that the design being tested can tolerate such usage, an endurance test was run with the valve operating sinusoidally with an amplitude of 20 percent of full stroke. The servoamplifier gain was adjusted for 20 percent step overshoot. The frequency was swept linearly from 5 to 200 hertz and back. This actuation was maintained continuously with a sweep period of 4 minutes. The valve was stopped each hour and visually checked for possible failures.

This test was continued until an accumulation of 24 hours of running had been achieved. Using an average frequency of 100 hertz, this corresponds to over 8.6 million cycles. No problems concerning wear or vibrations appeared during this run. Figure 13 shows a comparison of the frequency response before and after the cyclic life test. There is no apparent change in response. A check of the piston seals and rings indicated very minimal wear.

Overtravel endurance test. - This test measured the ability of the new Teflon piston ring and antirotation pin design to tolerate the wear caused by crossing the cushion orifices.

The valve was operated with the feedback loop open and with a 1-hertz square wave of sufficient amplitude to saturate the servoamplifier output. This drove the actuator

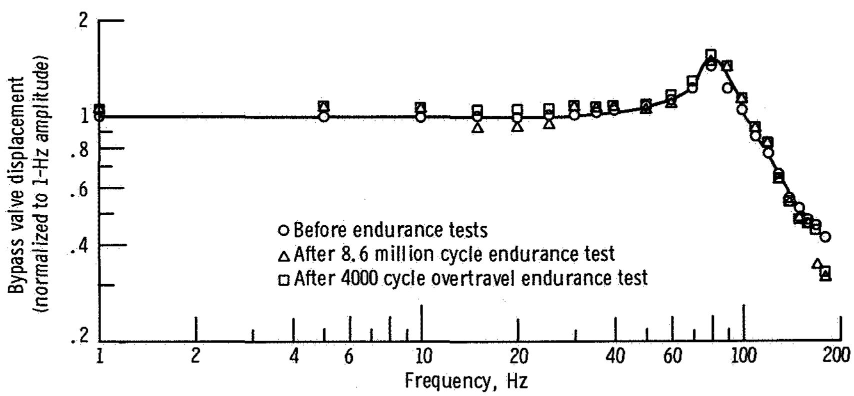
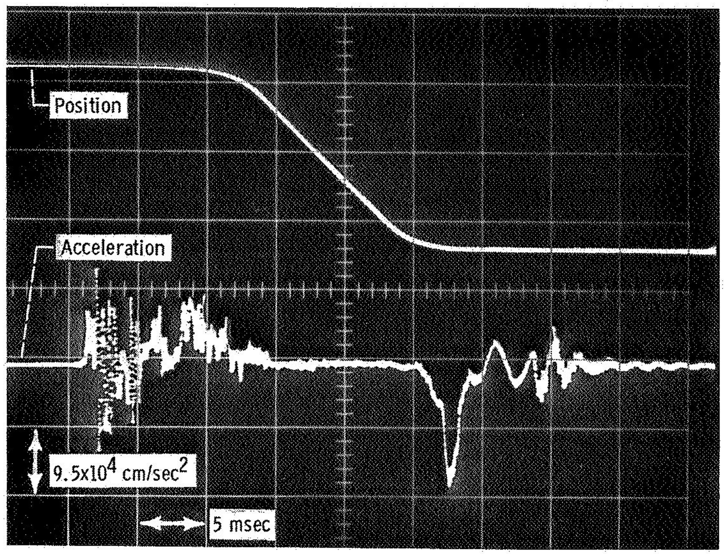
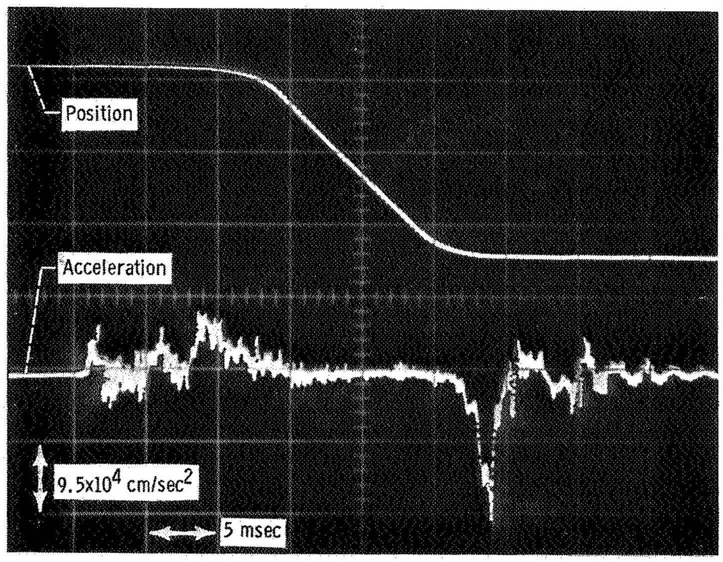


Figure 13. - Comparison of frequency response of bypass valve before and after endurance tests. Servosystem gain adjusted for 20 percent step overshoot.



Time

(a) After 1000 cycles.



Time

(b) After 4000 cycles.

Figure 14. - Bypass valve position and acceleration traces during overtravel endurance test. Position-transducer output saturates monitoring amplifiers at both ends of stroke because of piston overtravel.

piston into the hydraulic cushions in both directions. Accelerometer traces were monitored during this test to determine the point at which the seals failed. Failure could be determined by a large spike in acceleration which would indicate that the piston was hitting the end cap hard.

This test was continued for 4000 cycles, during which a record of position and acceleration was made every 500 cycles. Figure 14 shows the traces for 1000 and 4000 cycles. The acceleration spike at 4000 cycles had become slightly more peaked, indicating that the piston was hitting the end cap with more velocity than at 1000 cycles. However, even after 4000 cycles the acceleration spike was relatively small, and the seals still appeared to be functioning.

Again, frequency response was checked after this test and appeared to be unchanged, as indicated in figure 13. Visual inspection of the seals showed small ridges where material had been pressed into the orifices, but wear was not excessive, as shown in figure 15.

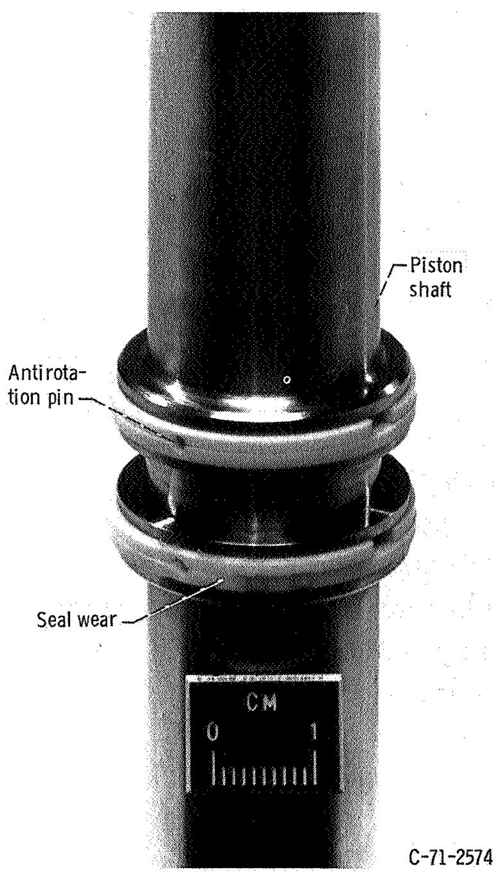


Figure 15. - Piston-ring seals after 4000-cycle overtravel endurance test.

Leakage Test

A test was performed to determine the airflow leakage of the bypass valve in the closed position. The plate clearance was 0.0076 centimeter, as mentioned earlier. At a pressure drop of 2.8 newtons per square centimeter, with a valve overlap of 0.010 centimeter, the equivalent leakage orifice area was 2.4 square centimeter.

At this clearance setting, the front valve edge overlap was varied from 0 (0.64 cm at the rear edge) to 0.08 centimeter (0.56 cm at the rear edge). The airflow through the valve remained constant for all positions, implying that the valve leakage was insensitive to overlap.

CONCLUDING REMARKS

Use of the electrohydraulically actuated slotted-plate valve as a fast-response pneumatic control valve has shown that careful attention must be given to design features which prevent the early failures common to these highly oscillatory devices. The new slotted-plate valve presented in this report has greatly improved endurance characteristics. Its features include mounting of hardware on a single plate which is also the stationary element of the valve, direct actuator coupling with a rod containing flexures to allow for misalinement, direct mounting of the servovalve on the actuator block, hydraulic actuator overtravel cushions, doweled mounting of the actuator block, and direct attachment of the position transducer to the valve moving element. These design improvements result in a cyclic endurance of better than 8 million cycles with negligible wear, and an actuator overtravel tolerance of better than 4000 occurrences.

Some galling between the valve plates occurred during large-amplitude response tests. The galling was caused by twisting of the moving-plate structure below the actuator coupling rod flange-attachment region. This problem was remedied for the present design by relieving the base plate slightly; however, this experience should be considered during the design phase for any future bypass valve designs.

The consideration of dynamic limit criteria to obtain an optimum piston area relation during design resulted in near-optimum dynamic performance for this bypass valve. (Optimum in this case means the fastest response obtainable given the constraints of fixed supply pressure, servovalve characteristics, and load mass.) The valve performance for peak-to-peak sinusoidal oscillations of 20 percent of full open area was flat to within ±3 decibels for frequencies up to 120 hertz.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 24, 1973,
501-24.

APPENDIX - SYMBOLS

A_p actuator piston area, cm^2

A_p^* optimum actuator piston area, cm^2

A_s servovalve spool end area, cm^2

M_o equivalent actuator piston and output load mass, $(N)(sec^2)/cm$

P_{sp} hydraulic supply pressure, N/cm^2

P_1, P_2 servovalve control-port pressures, N/cm^2

Q_r servovalve rated flow, cm^3/sec

Q_1, Q_2 servovalve control port flows, cm^3/sec

$q_{f(max)}$ maximum flapper nozzle flow, cm^3/sec

X_o zero to peak amplitude of sinusoidal position displacement, cm

X_o' zero to peak amplitude of sinusoidal position displacement at intersection of velocity limit and flapper flow limit, cm

x_{com} commanded position, cm

x_{fl} flapper displacement, cm

x_s servovalve spool displacement, cm

$x_{s(max)}$ maximum servovalve spool displacement, cm

ω frequency, rad/sec

ω' frequency at intersection of velocity limit and flapper flow limit, rad/sec

REFERENCES

1. Neiner, George H.; Crosby, Michael J.; and Cole, Gary L.: Experimental and Analytical Investigation of Fast Normal Shock Position Controls for a Mach 2.5 Mixed-Compression Inlet. NASA TN D-6382, 1971.

2. Neiner, George H.: Servosystem Design of a High-Response Slotted-Plate Overboard Bypass Valve for a Supersonic Inlet. NASA TN D-6081, 1970.

3. Baumbick, Robert J.: Device for Producing Dynamic Distortion Patterns at Inlets of Air-Breathing Engines. NASA TM X-2026, 1970.

4. Zeller, John R.: Analysis of Dynamic Performance Limitations of Fast Response (150 to 200 Hz) Electrohydraulic Servos. NASA TN D-5388, 1969.

5. Zeller, John R.; and Webb, John A., Jr.: Determination and Evaluation of Performance Limit Criteria of Fast-Response Electrohydraulic Servosystems. NASA TM X-2736, 1973.

6. Merritt, Herbert E.: Hydraulic Control Systems. John Wiley & Sons, Inc., 1967.

7. Batterton, Peter G.; and Zeller, John R.: Performance Characteristics of Improved Servoamplifier for Electrohydraulic Control Systems. NASA TM X-2167, 1971.

NASA-Langley, 1973 —— 3

E-7399