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TARADCOM

R and D LABORATORY

TECHNICAL REPORT

NO. 12177

A FLUIDIC ADAPTIVE DAMPER -

BREADBOARD DESIGN, FABRICATION AND TEST

FINAL REPORT

JANUARY 1979

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R.K. VanAusdal, Electrodynamics
Bendix Corporation
M.M. Steele, USATARADCOM
S.M. Tenney, USAERADCOM

by

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U.S. ARMY TANK-AUTOMOTIVE
RESEARCH AND DEVELOPMENT COMMAND
Warren, Michigan 48090

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Accession For

ABSTRACT

The purpose of this development effort was to achieve a practical implementation of a fluidic composite valve to replace the compression valve of a conventional shock absorber to improve vehicle control and performance. The effort was to result in the delivery of two breadboard adaptive fluid vibration dampers (shock absorbers) for Government evaluation.

Program results include the sizing, design, packaging, fabrication and testing of several modifications of a compact, inertially controlled arrangement of three parallel fluidic beam deflection amplifiers controlling a vortex valve. Fabrication techniques of the fluidic valve included photo-etching of steel amplifier discs that were diffusion bonded to form a sleeve for a conventional valve spool that serves as the inertial sensor. A unique two-electrohydraulic servovalve test circuit was produced that provided constant flow control over a wide range of load variations.

Two breadboard vibration dampers, one assembled and one unassembled, were delivered to the Government for evaluation.

1. INTRODUCTION

In July 1976, The Bendix Corporation, Electrodynamics Division initiated work on a contract with the Mobility Systems Laboratory, U.S. Army Tank-Automotive Research and Development Command (TARADCOM) for the design, manufacture and test of two breadboard adaptive fluidic vibration dampers for Government evaluation. The effort was to be based on the study performed on Contract DAAE07-75-C-0104 by Dr. R.E. Bowles as reported in TARADCOM Report No. 12086, October 1975.

The study developed a simplified fluidic composite valve to replace the compression valve of a conventional shock absorber. The fluidic unit incorporated an acceleration sensor which was to monitor wheel vertical acceleration and modify the compression valve damping coefficient as a function of the magnitude of the acceleraiton. The purpose of varying the damping coefficient is to reduce energy absorbed, promote damper longevity, reduce transferred energy to the vehicle, improve the ride quality, increase the wheel-ground-contact time and increase vehicle control and performance.

This report presents the history and results of the present program which produced three generations of the fluidic compression valve to attain satisfactory static results. It also produced a unique two-electrohydraulic-servovalve test circuit that provided constant flow control over a wide range of load variations.

2. OBJECTIVES

The objectives of the program were in general to design, fabricate and test a breadboard adaptive fluidic vibation damper incorporating an adaptive damper compression valve as defined in TARADCOM Technical Report No. 12086, Investigation of an Adaptive Fluidic Vibration Damper, for TARADCOM evaluation. To accomplish the objectives, the program tasks were established as follows:

- Size a preliminary vortex valve for immediate testing to determine the force ratio possible using a small hydraulic vortex valve.
- Conduct preliminary sizing and design layout of a manifold to connect the various parts of the simulated shock absorber.
- Make detail drawings from which parts can be made for all the required components.
- Fabricate all the required components.
- Design and fabricate the required test equipment, which includes a flow control test bench.
- Test the combination vortex valve-beam deflection amplifier-accelerometer (adaptive compression valve) as a unit, both statically and dynamically. If design deficiencies are identified as a result of testing, such deficiencies shall be isolated and corrected.

3. RESULTS/CONCLUSIONS

RESULTS

Adaptive Fluidic Compression Valve

After unsuccessful tests of two designs, a third configuration of the compression valve performed reasonably well during static tests. The spool type valve design, which used photo-etched discs that were fusion bonded to form the sleeve, is illustrated in Figure 1. Static test results, in which each spool of the three different assemblies of the latest design was manually positioned, are shown in Appendix A, Figures A-1, A-2, and A-3. These three figures show the resistance ratio as determined from the square of the ratio of measured flows for various spool positions from the highest resistance (lowest flow), or neutral position. In these tests, the lowest resistance (neutral) position was determined by pushing the disc-type-spring-centered spool until the lowest flow rate was attained; then the position adjustment screw was turned in small increments and the increased flow rate recorded.

As can be seen from the plots in Figures A-1 through A-3, the results of the three assemblies are in close agreement in overall magnitude of the achieved resistance ratio. While the highest ratio (from 3.0 to 3.5) is less than desired, it was considered sufficient for use in the breadboard fluidic vibration damper. Consequently, the No. 1 assembly was subjected to dynamic testing.

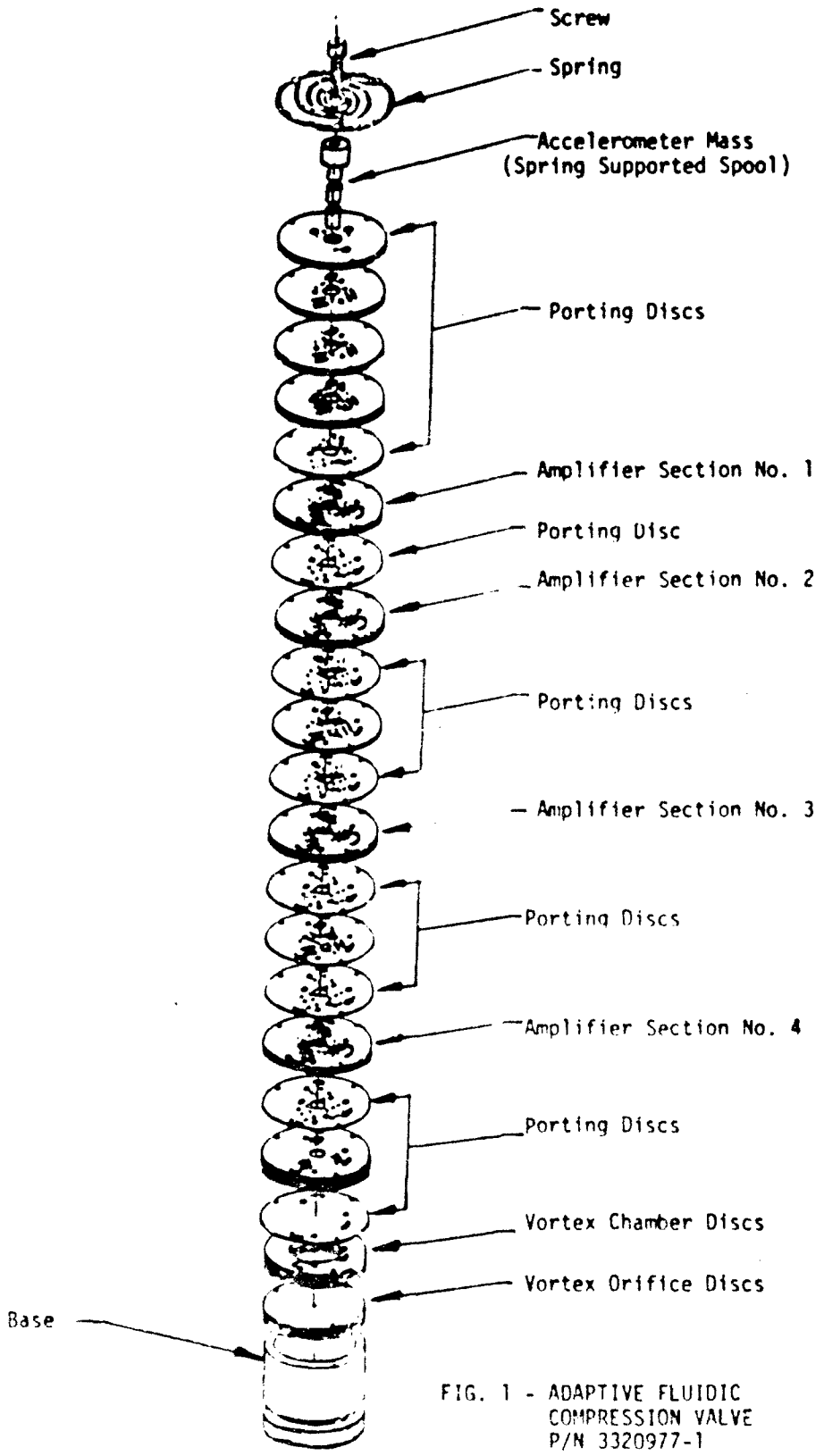


FIG. 1 - ADAPTIVE FLUIDIC COMPRESSION VALVE P/N 3320977-1

Dynamic testing was accomplished in the same test fixture used for static testing, reference Figure 2, which simulates the compression valve cavity of the Valve Assembly, P/N 3320976. This small test fixture was used so that it could be displaced in electrically programmed motion by a small electrohydraulic servoactuator in the test arrangement of Figure 3.

Results of the dynamic testing were extremely disappointing in that the valve produced only very small changes in resistance (pressure drop) and then only in response to relatively large magnitude-square wave displacements. No response could be detected with sinusoidal and triangular wave displacements. Numerous removals of the valve from the fixture -- in an attempt to find and remove sources of friction -- did not contribute to any appreciable improvement. At this point, it should be noted that prior to diffusion bonding, the assembly was clamped and the three locator pins removed. After bonding, some of the discs in the assembly were shown to have slipped approximately 0.006 inch diametrically. With the known precision of the etched discs and the 0.010 inch amplifier nozzle size of three amplifiers 120 degrees apart, there was no assurance of (1) the accelerometer bore being concentric, (2) the nozzle sizes being 0.010 inch, (3) the concentricity of receiver orifices, and (4) the vortex valve, which in itself is a disc assembly, was not misaligned. Normal manufacturing practice calls for leaving locator pins in place until after bonding.

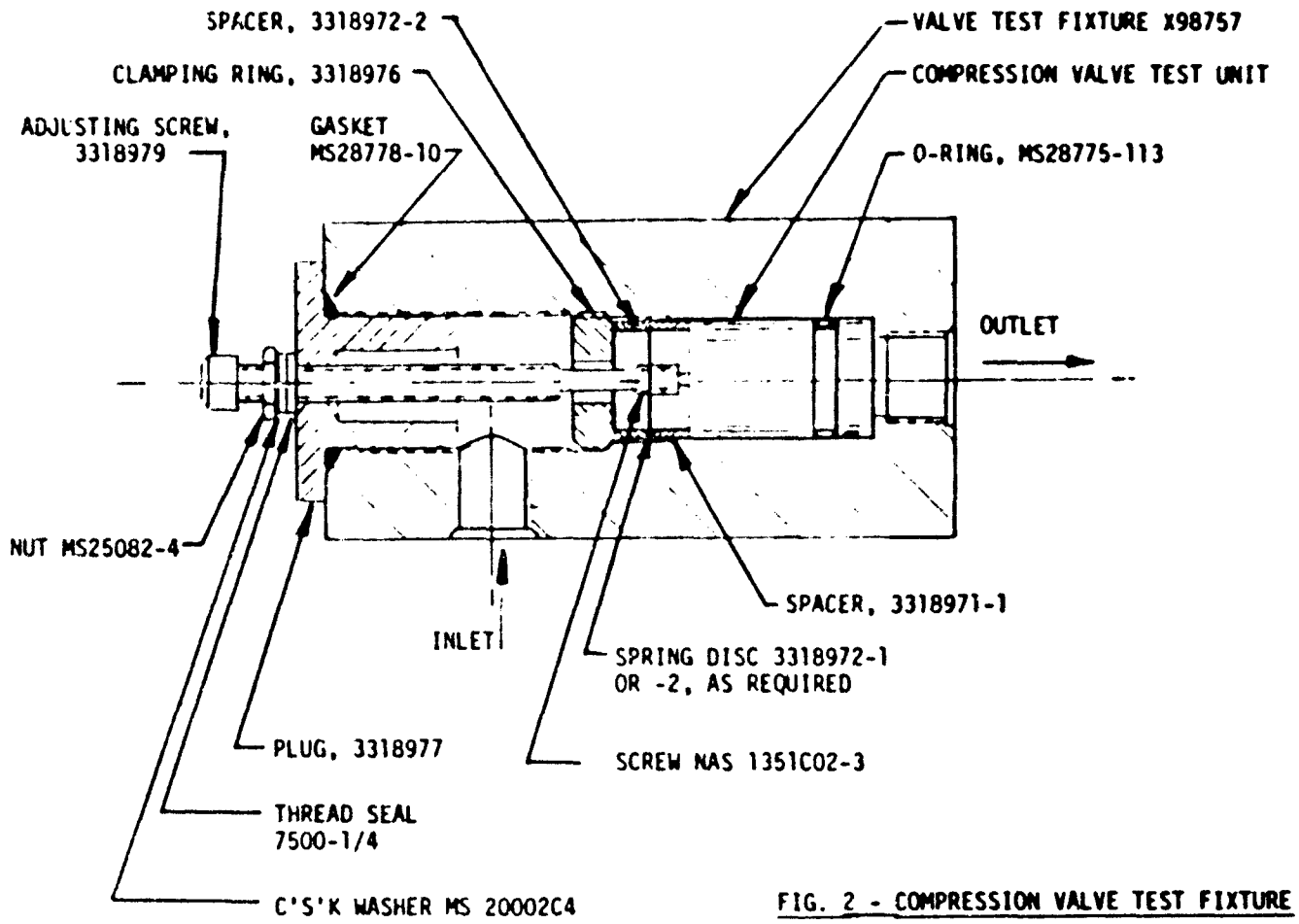


FIG. 2 - COMPRESSION VALVE TEST FIXTURE

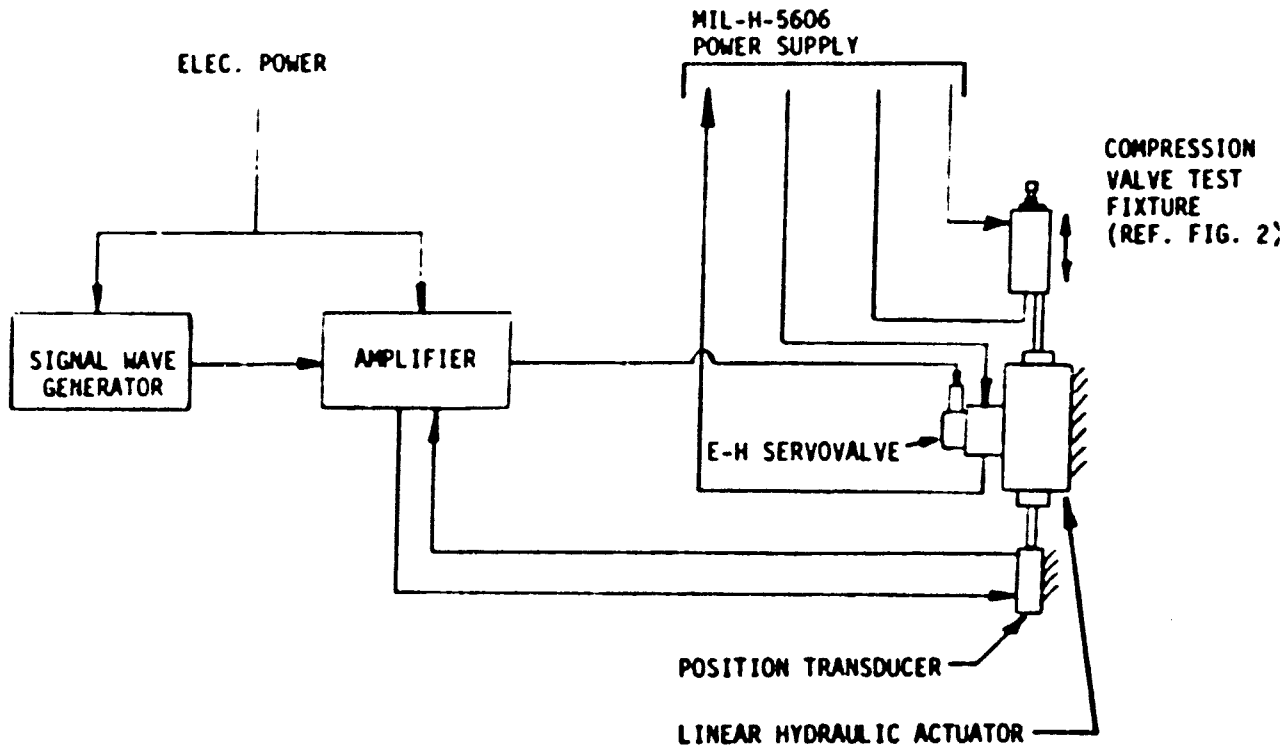


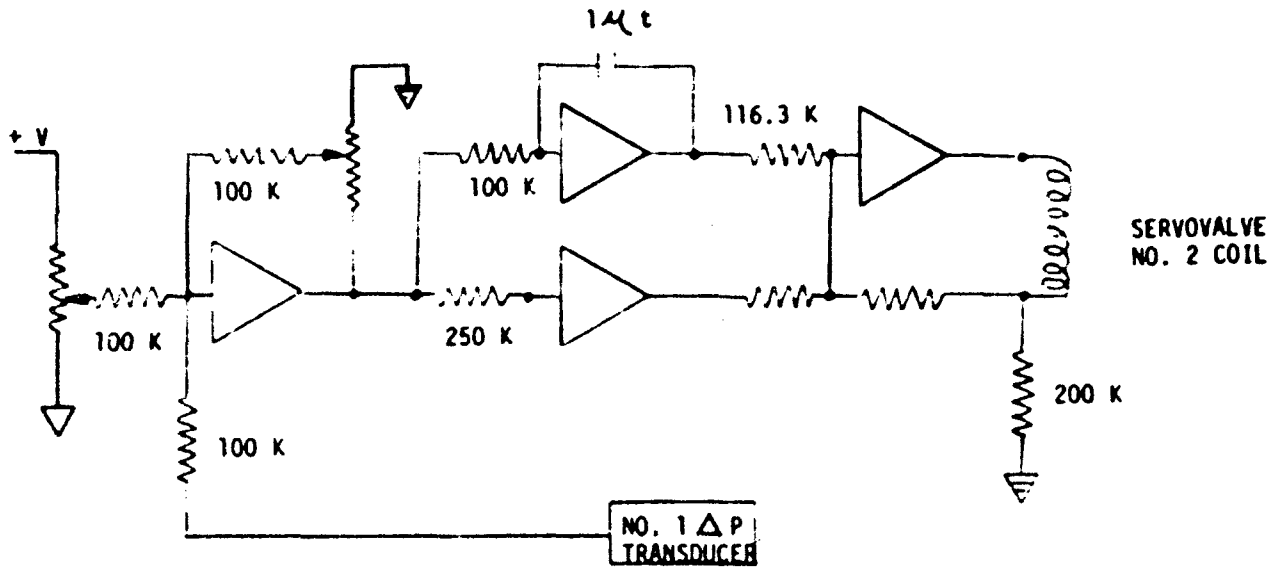
FIGURE 3 - COMPRESSION VALVE - DYNAMIC TEST ARRANGEMENT

Another factor that could have an influence on the dynamic performance is the flow force (Bernoulli force) generated by flow through the spool/sleeve metering apertures. This was not originally considered a significant probability; however, as discussed subsequently in the Conclusions, the force could be equivalent to an acceleration of 2 g's with only a flow of 0.2 gpm.

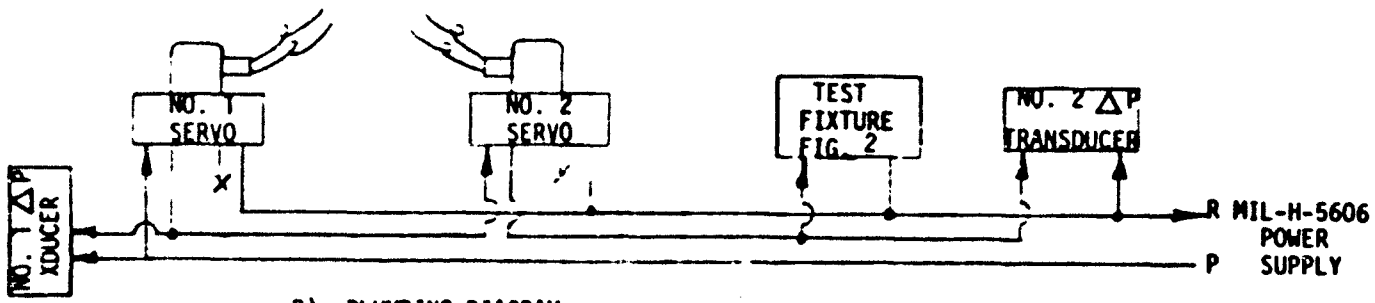
After consultation with TARADCOM personnel, and subsequent spool/sleeve lapping to reduce friction, it was decided to evaluate the performance of the valve in a fluid containing graphite in suspension, rather than the MIL-H-5606 hydraulic fluid previously used in testing, to further attempt minimization of friction. However, since the Bendix hydraulic test system could not be contaminated with the graphite because of the cost of subsequent decontamination and since Bendix had not planned on testing the complete Fluidic Shock Absorber Assembly (shown in Appendix A, Figures A-4 and A-5), it was mutually agreed to deliver one assembly filled with Arco Graphite Motor Oil to TARADCOM for evaluation.

Flow Control Test Bench

The flow control test arrangement required to maintain constant system flow during the resistance variations imposed by the compression valve during dynamic testing is shown in Fig. 4. This closed loop system with two electrohydraulic servovalves and a pressure feedback loop was designed to maintain a constant differential pressure ($P - C_1$) across the No. 1 servovalve for any given input current to the servovalve by varying the input command to the No. 2 servovalve. This caused its differential pressure ($C_1 - C_3$) to vary inversely with the pressure drop ($C_3 - R$) across the compression valve on test. Although no specific test data was generated in setting up the test arrangement, Fig. 5 -- showing the preliminary vortex valve test results -- indicates that flow control was accomplished. For example, at 3.5 milliamps current, the resistance (ΔP) ratio of 10.7:1 was attained with tangential-flow only to radial-flow only, while maintaining a constant flow rate in the system as indicated by Fig. 6. In this figure, it can be seen that the flow gain is identical for both radial and tangential flow modes for which the resistance ratio varied as stated above.



A) WIRING DIAGRAM



B) PLUMBING DIAGRAM

FIGURE 4 - FLOW CONTROL TEST SYSTEM SCHEMATICS

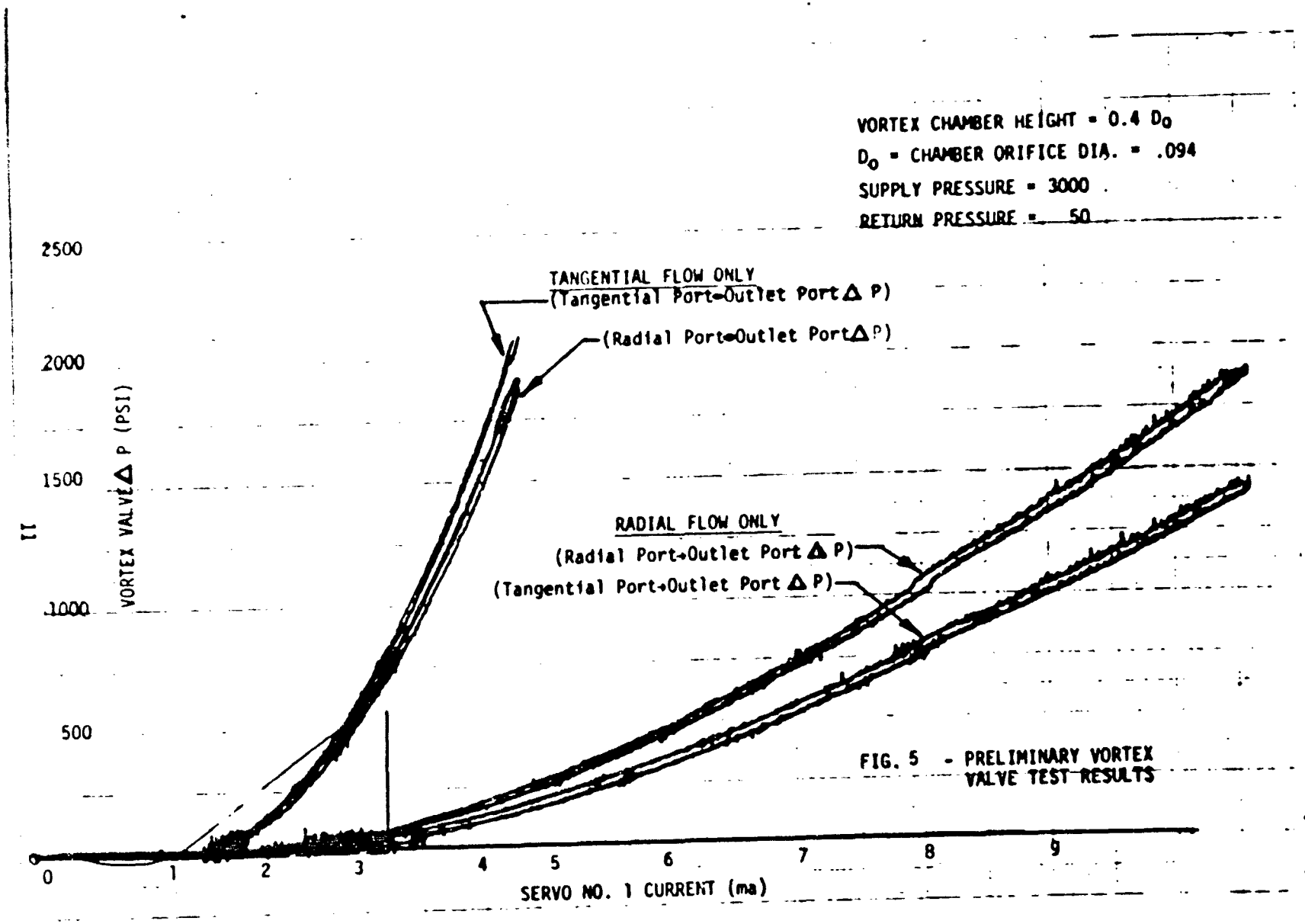


FIG. 5 - PRELIMINARY VORTEX VALVE TEST RESULTS

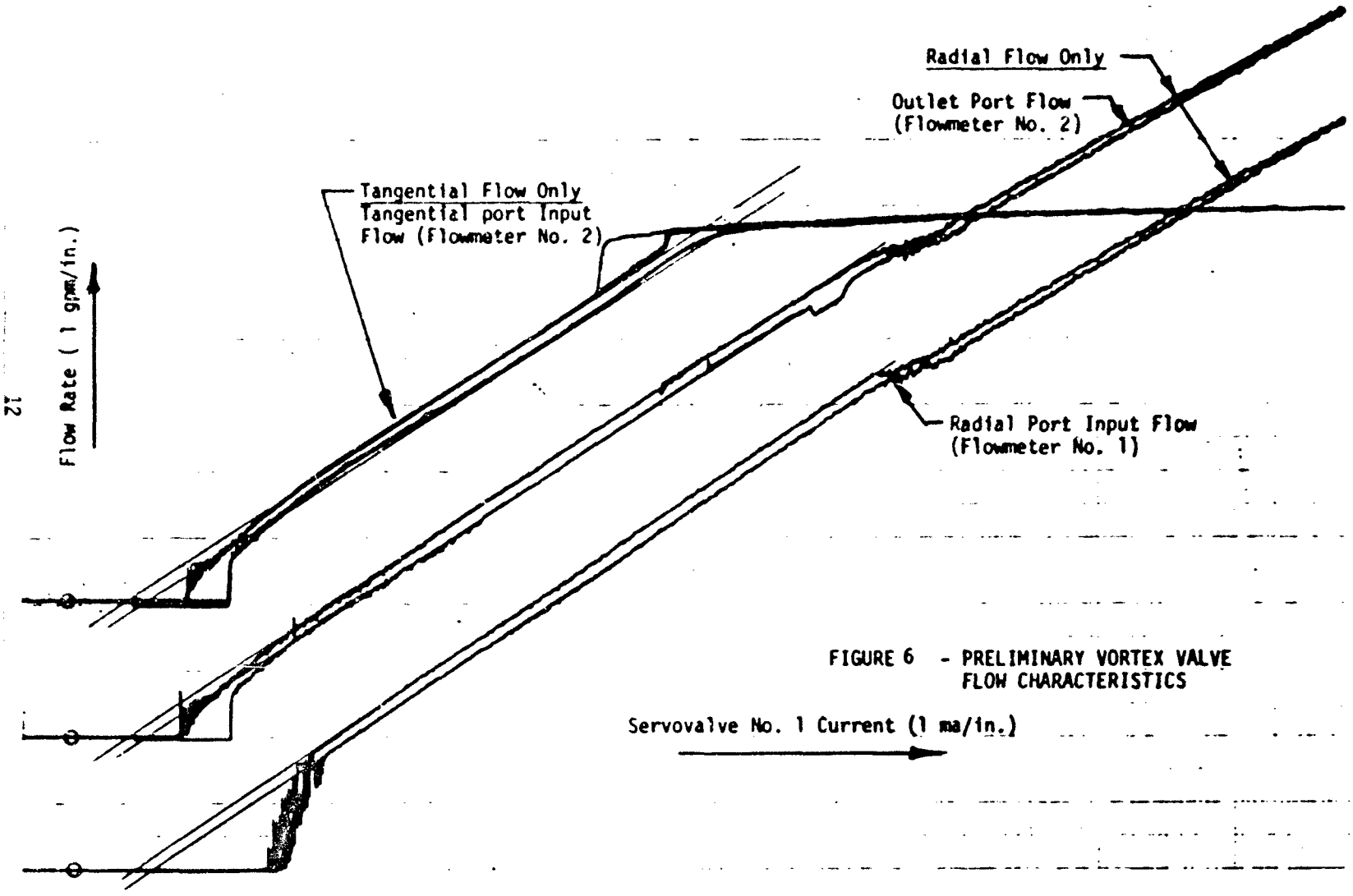


FIGURE 6 - PRELIMINARY VORTEX VALVE FLOW CHARACTERISTICS

Fig. 7 shows the effect on Servo No. 2 current as the current is varied on Servo No. 1. The load in each case was the preliminary vortex valve flowing in the tangential mode only and the radial mode only. This is the correct response if the total system ΔP is held constant and the ΔP across Servo No. 1 is held constant.

For the static case, the flow through Servo No. 1 is the same as the flow through Servo No. 2 which is the same as the flow through the vortex valve.

Thus, assuming 3000 psi system ΔP and 500 psi across Servo No. 1,

$$Q = K\Delta i_1 \sqrt{500} = K\Delta i_2 \sqrt{2500 - \Delta P} = K_v \sqrt{\Delta P}$$

$$(K\Delta i_2)^2 (2500) = \left[K_v^2 + (K\Delta i_2)^2\right] \Delta P$$

$$\Delta P = \frac{2500 \left(K\Delta i_2\right)^2}{K_v^2 + (K\Delta i_2)^2}$$

$$\therefore (K\Delta i_1)^2 (500) = (K\Delta i_2)^2 \left(2500\right) \left[1 - \frac{(K\Delta i_2)^2}{K_v^2 + (K\Delta i_2)^2}\right]$$

Solving for $(K\Delta i_2)^2$

$$\frac{K_v^2 (K\Delta i_2)^2}{K_v^2 + (K\Delta i_2)^2} (5) = \left(K\Delta i_1\right)^2$$

$$K_v^2 (K\Delta i_1)^2 + (K\Delta i_2)^2 (K\Delta i_1)^2 = 5 K_v^2 (K\Delta i_2)^2$$

$$K_v^2 (K\Delta i_1)^2 = \left[5 K_v^2 - (K\Delta i_1)^2\right] (K\Delta i_2)^2$$

$$(K\Delta i_2)^2 = \frac{K_v^2 (K\Delta i_1)^2}{5K_v^2 - (K\Delta i_1)^2}$$

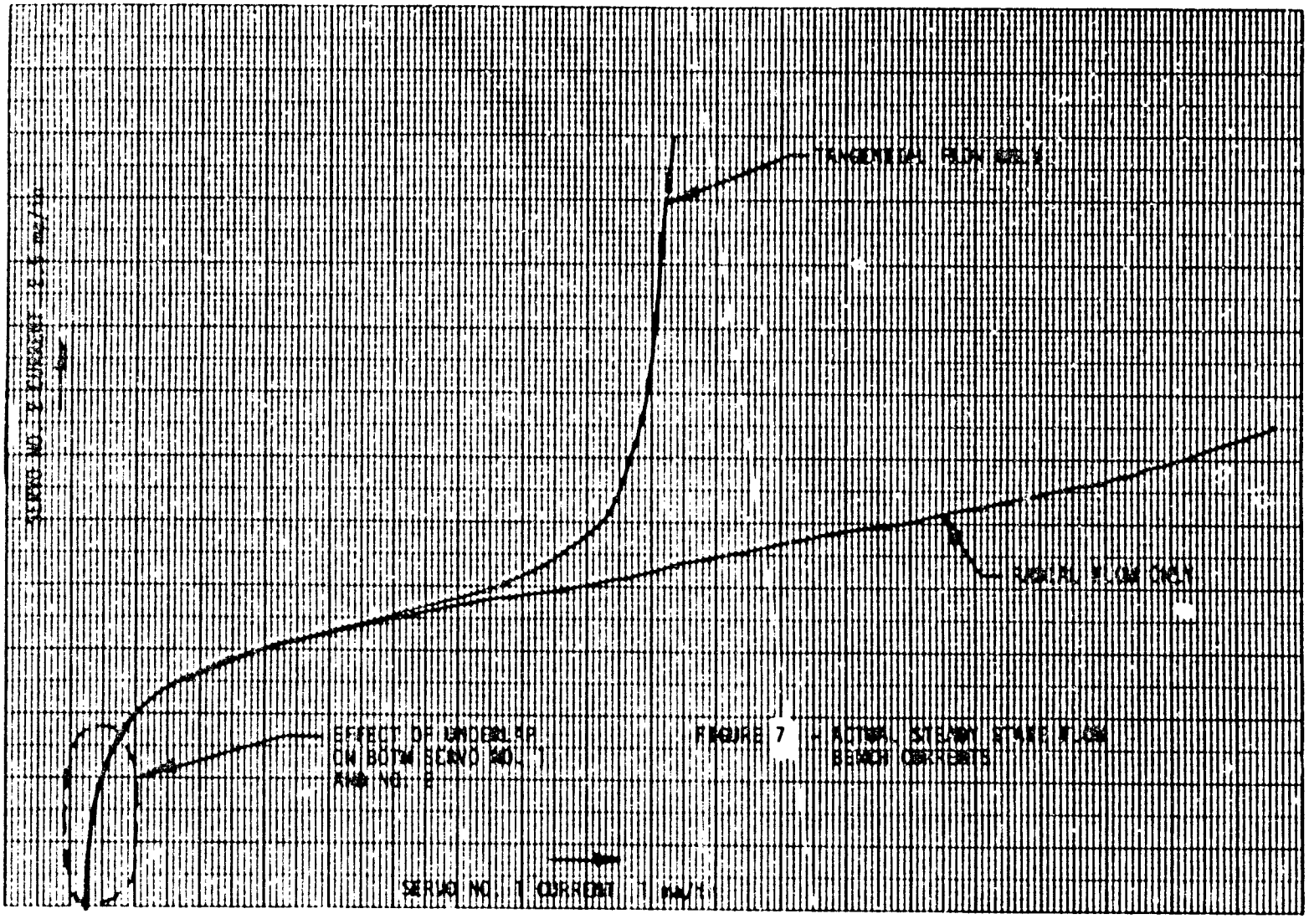


FIGURE 7 – ROTOR STEADY STATE FLOW BENCH DEFLECTS

K_v is the effective area of the vortex valve. If we assume that $K_v = 3K = 3$, we get the curve shown in Fig. 8. If we assume that $K_v = K = 1$, we get the other curve shown in Fig. 8. These curves are identical in shape to the curves of Figure 7 taken from the flow bench with the vortex valve. Therefore, we conclude that at least statically, the flow through the flow bench is only a function of Servo No. 1 current, not of the downstream resistance.

At this point in the damper development, the dynamic requirements of the damper itself were greatly reduced due to the lagged accelerometer and the nonlinear effects of the damping orifice. (The calculated response of the damper for various input velocities -- flows -- is shown in Appendix B, Figure B-4.) Notice that the frequency peaks are in the 1 to 4 Hz range; thus, no high response testing was done on the flow bench. The response of the system to step inputs seemed to be adequate for the required 4 cps response.

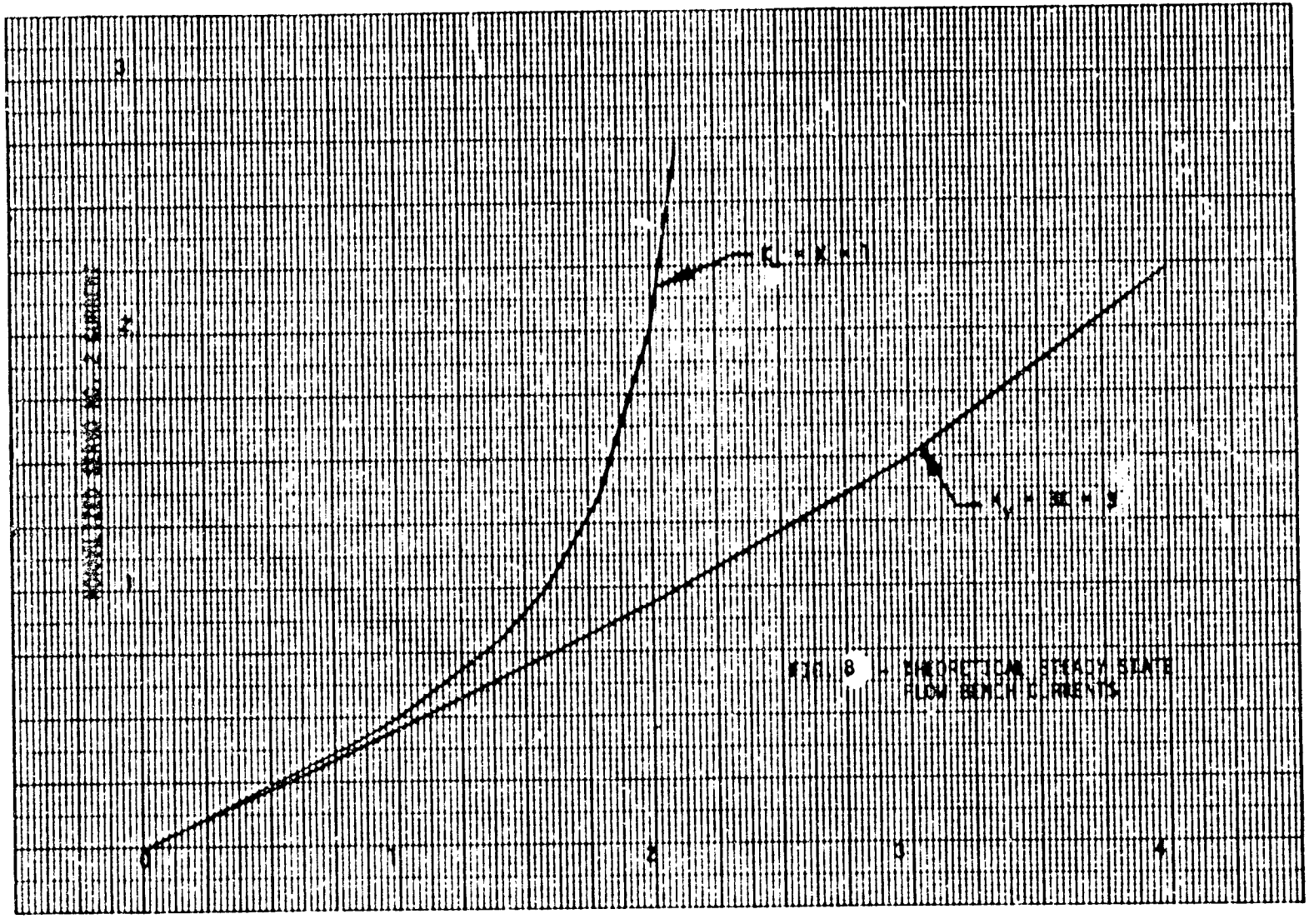


FIG. 8 - INCOMPRESSIBLE STEADY STATE FLOW BENCH ELEMENTS

CONCLUSIONS

Based on the results presented, the following conclusions are submitted:

- It seems feasible to produce a practical fluidic compression valve that can be incorporated in an adaptive vibration damper.

- The performance of the compression valve, although satisfactory hydraulically (by achieving a max. resistance ratio variation of 3.5:1), was poor in respect to input acceleration. Since repeated attempts at eliminating conditions that could induce friction between the valve spool and sleeve failed to improve performance, it is concluded that significant additional spool mass is required. Using a much larger spool mass means using a larger spool and a spool such as this would not need the added gain of the beam deflection amplifiers (the amplifiers only reduce the amount of flow required to go through the spool valve). Thus, the objectives of better static and dynamic performance could be realized.

- Another phenomenon that may have contributed significantly to the poor dynamic response of the spool mass to input acceleration is the Bernoulli force developed by flow through the spool/sleeve porting apertures. This force can be expressed:

$$F_B = C_d \sqrt{\frac{2\rho}{g} \Delta P} \quad Q \cos 69^\circ \qquad (1)$$

Where C_d = Aperture discharge coefficient

ρ = Fluid specific weight

g = Acceleration constant

P = Pressure differential across aperture

Q = Flow through aperture

So by assuming

$C_d = .65$

$g = 386 \text{ in/sec}^2 \ (9.804 \text{ m/sec}^2)$

$Q = .385 \text{ in}^3\text{/sec} \ (6.309 \times 10^{-6} \text{ m}^3\text{/sec})$

$\rho = .031 \text{ lb/in}^3 \ (.841 \times 10^4 \text{ N/m}^2)$

$\Delta P = 100 \text{ psi} \ (6.894 \times 10^5 \text{ N/m}^2)$

the force,

$$F_B = .65\sqrt{\frac{2\,(.031)}{386}\,100} \quad .385 \text{ Cos } 69^\circ$$

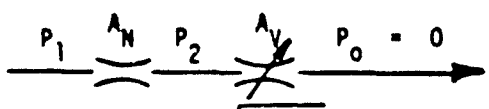
$$= .0114 \text{ lb. } (.051 \text{ N})$$

With a measured spool weight of 2.65 grams (.0058 lb), it is apparent that it would take nearly 2 g's to attain the flow assumed. If the ΔP is quadrupled $\left[400 \text{ psi } (27.58 \times 10^5 \text{ N/m}^2)\right]$, then the number of g's required would double.

- A force ratio of about 10/1 is possible with a small hydraulic Vortex Valve (Ref. Fig. 5), thus indicating that there is room for significant improvement in static performance.

- The best static performance was achieved with an amplifier nozzle area to vortex outlet area ratio of 1.13 to 1, indicating that sufficiently high velocity, and thus pressure drop, must be maintained through the amplifiers for them to function with the Vortex Valve. However, the resistance through the amplifiers is essentially constant while the resistance of the Vortex Valve is that resistance which is variable. High resistance in the amplifiers greatly diminishes the effective resistance ratio of the Vortex Valve.

Assuming a maximum possible force ratio of the Vortex Valve to be 10/1 and the area ratio of amplifier nozzle to vortex outlet to be 1.13, the maximum possible system force ratio can be calculated as follows:



$$Q = 103\, A_N \sqrt{P_1 - P_2} = 103\, A_V \sqrt{P_2}$$

If A_V varies from .885 A_N to .2798 A_N, then the maximum force ratio

$$\left(\frac{P_1 @ A_V = .2798\, A_N}{P_1 @ A_V = .885\, A_N}\right) \text{ is } F_{ratio} \approx \frac{\left(1 + \left(\frac{1}{.2798}\right)^2\right)}{\left(1 + \left(\frac{1}{.885}\right)^2\right)} = \underline{6.048}$$

Using the same analysis but allowing $A_{V\,MAX} = .333\, A_N$ and maintaining the maximum Vortex Valve force ratio at 10/1 gives a maximum pressure force ratio of 9.102.

The ratio of actual pressure ratio to theoretical pressure ratio, $\frac{3.2}{6.05} = .529$, indicates that there is still significant improvement to be made using the present design sizing.

- A system of achieving constant flow with varying load resistance was achieved using the two-servovalve flow bench as described previously. Its dynamic performance (approximately 4 cps at 90°) was considered to be adequate for dynamic testing of the final compression valve.

4. RECOMMENDATIONS

STATIC PERFORMANCE

- An investigation should be conducted to determine if impedance matching can improve the static performance of the beam deflection amplifier-vortex valve combination from the achieved 3.0 or 3.5 to 1 pressure ratio to closer to the 10 to 1 pressure ratio obtained with the preliminary vortex valve.

- Although the supply (or power) flow channel areas were increased as much as seemed practical in the space allowed for the final configuration of the beam deflection amplifier-vortex valve combination (without impacting the housing design), no analysis was made to evaluate the pressure loss or the effect on performance. Therefore, this evaluation should be made to assess its contribution, if any, to a poorer static performance than anticipated from the preliminary vortex valve results.

- Consideration should be given to conducting a design study of a spool and sleeve type valve to replace both the present spool valve and beam deflection amplifiers. Elimination of the beam deflection amplifiers should make it possible to realize a pressure ratio demonstrated with the preliminary vortex valve (10:1).

DYNAMIC PERFORMANCE

- A design study should be made to determine how much larger a spool mass is needed for the existing concept to achieve the required sensitivity to acceleration inputs and if it can be packaged in a reasonably sized envelope.

5. ADAPTIVE FLUIDIC COMPRESSION VALVE DEVELOPMENT

FINAL COMPRESSION VALVE

The final compression valve configuration which is shown in Fig. 9 consists of two integrated parts. The large cylindrical part (Fig. 9) is a fusion bonded assembly of steel (Sandvik 13C26) discs and a steel base containing the O-ring groove for installation. The steel discs which occupy the upper third of the sleeve are shown in Fig. 10, along with a U. S. dime (coin) for size comparison. The discs are stacked in an order that would form three parallel hydraulic circuits containing a total of 12 beam deflection amplifiers, a vortex valve and the necessary porting discs to interface with the spool as it is displaced relative to the stack of discs by acceleration of its mass. The spool that slides within the sleeve is the smaller cylindrical section of the smaller component in Fig. 9. The larger cylindrical mass that is integral with the spool is the major mass effected by acceleration. The spool material for this version of the compression valve is a tungsten (90%)-nickel (6%)-copper (4%) alloy (CMW 1000) with a density twice (16.96 gms/cc) that of steel. The spool mass is suspended by means of a disc type spring shown in Fig. 11. Although the -1 spring (0.015 wide spirals) is shown, the -2 spring (0.030 wide spirals) was used with the heavier tungsten-alloy spool. The -2 spring, which was made of 0.004-inch thick Sandvik 13C26 steel, was designed for a spring rate of .435 lbs/inch. The spring is clamped in place between two spacers by a threaded clamping ring within the installation bore of the valve housing, or test fixture. At the spool end opposite the large mass, a counterbore to a hole drilled

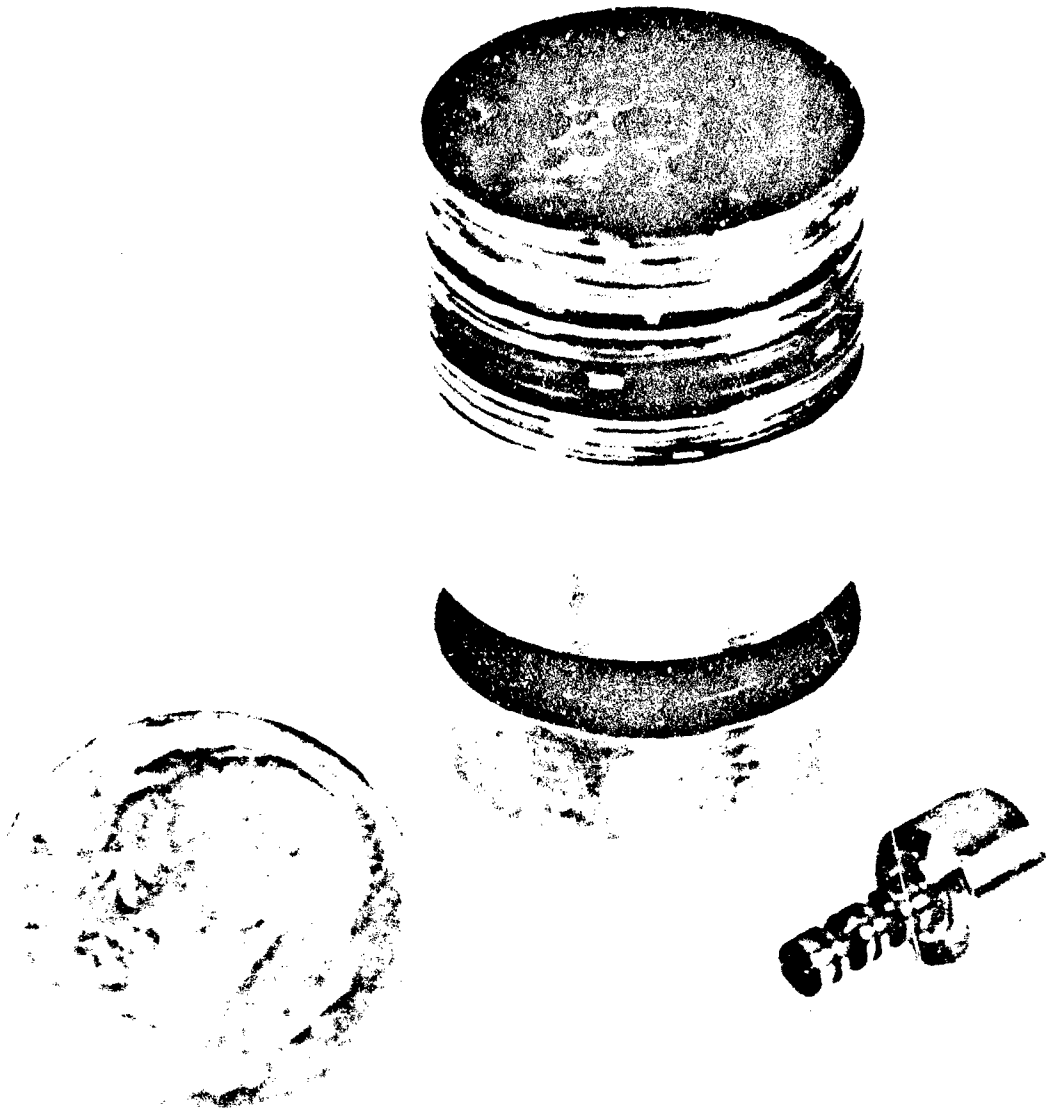


FIG. 9 - FINAL COMPRESSION VALVE MATCH ASSEMBLY
PART NO. 3320977-1

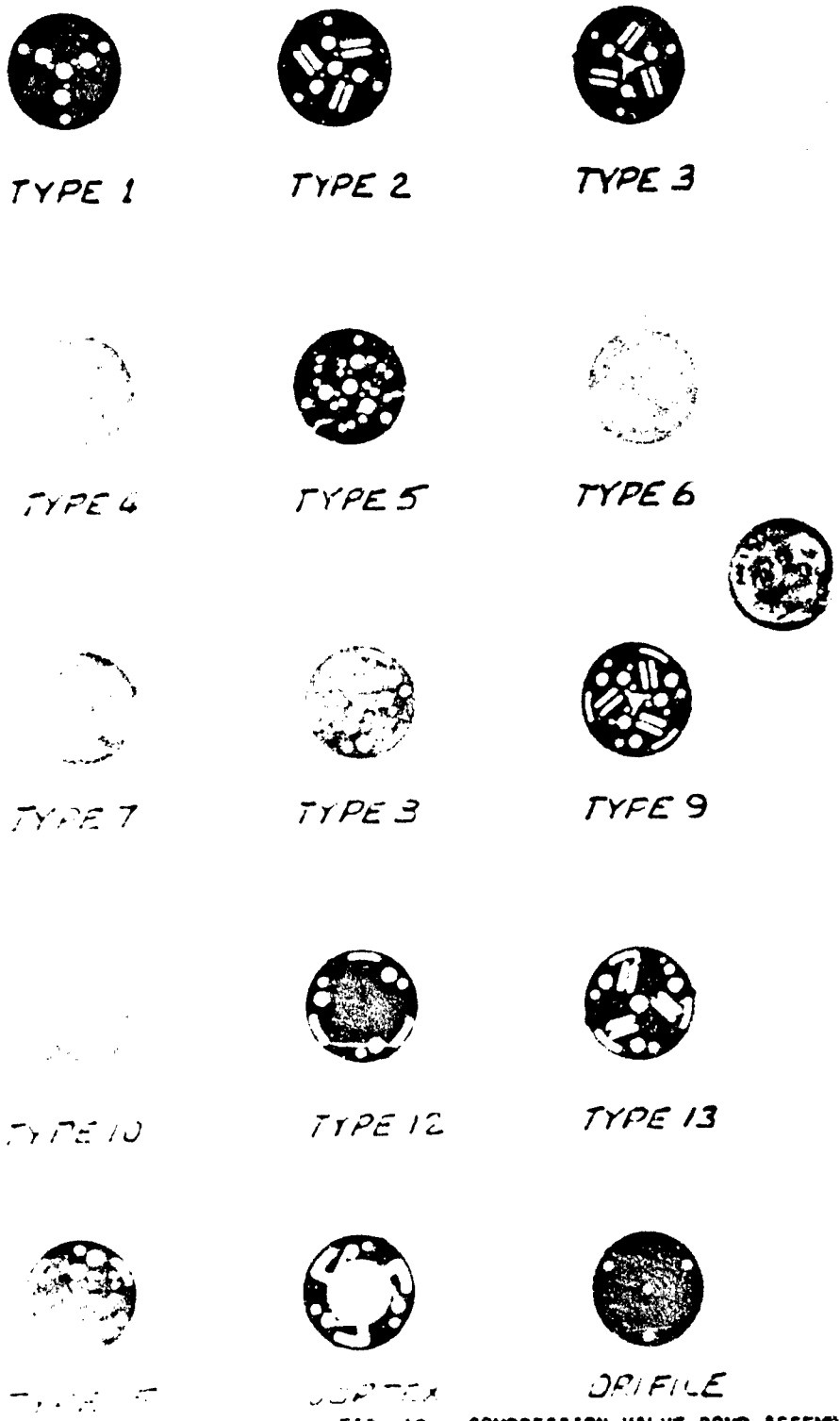


FIG. 10 - COMPRESSION VALVE BOND ASSEMBLY DISCS
PART NO. 3320998

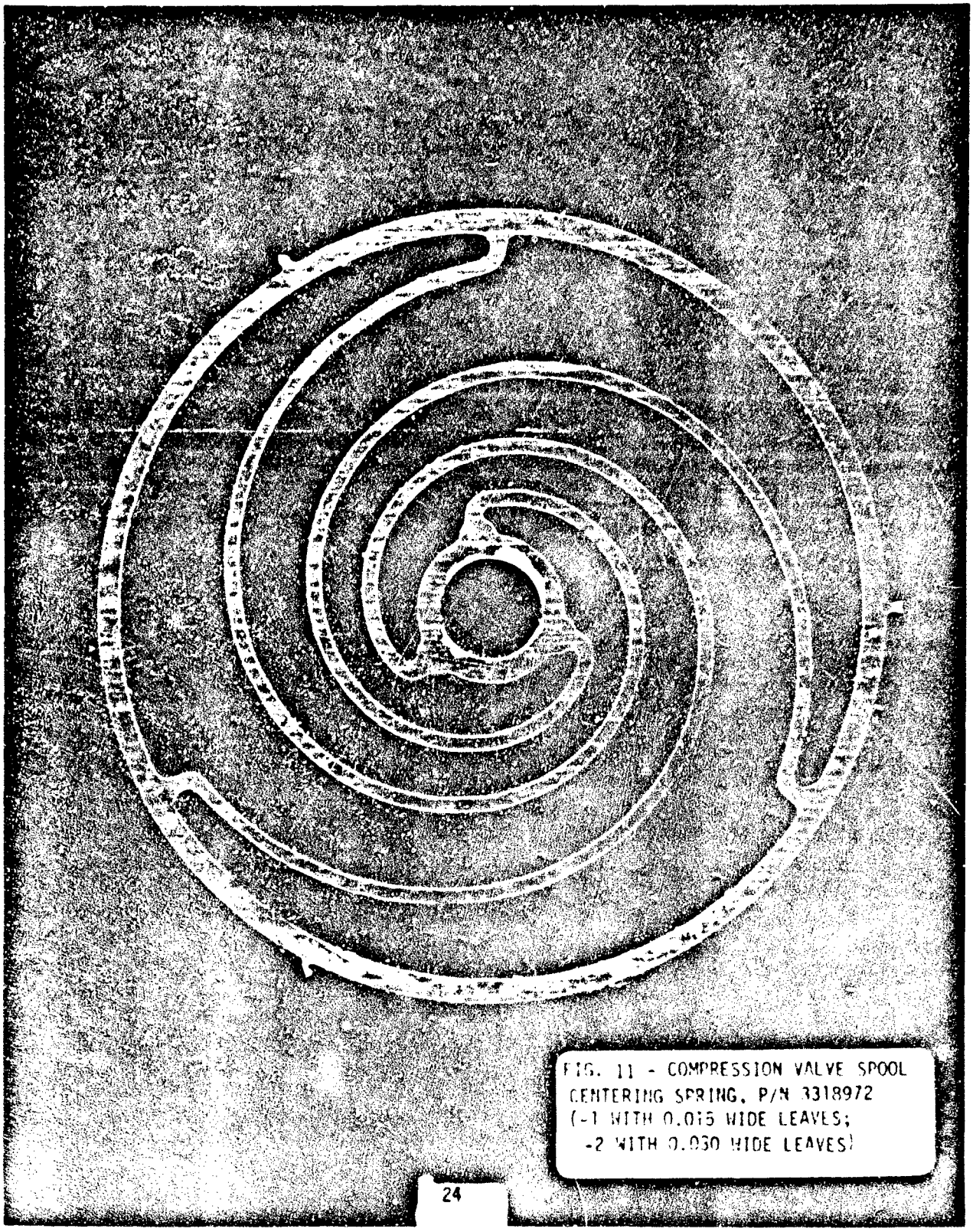


FIG. 11 - COMPRESSION VALVE SPOOL CENTERING SPRING, P/N 3318972 (-1 WITH 0.015 WIDE LEAVES; -2 WITH 0.030 WIDE LEAVES)

through the spool is used to accept a 0.062 OD, 0.028 thick sapphire orifice containing a 0.005-inch diameter metering hole. Consequently, as the spool mass is forced to move by the acceleration force applied, the displaced fluid in the "blind" spool bore is conducted through the orifice and spool hole back to the supply, or high pressure side of the valve, effecting a first order lagging of spool motion. This is done to comply with the Mod II Adaptive Fluidic Vibration Damper configuration pursuant to TARADCOM Report 12226 so that the compressive force is controlled by vertical wheel velocity rather than acceleration.

The four sections (three each) of beam deflection amplifiers that are enumerated in Figure 1 are made up of the number and type of discs called out in the figure. The three types of discs (6, 7, and 10) forming the 12 amplifiers are shown in enlarged form in Appendix A, Figures A-6, A-7, and A-8. The amplifiers control the flow to either the tangential or radial ports of the vortex valve and all are exclusively in the tangential output control mode when the spool is in the neutral or zero acceleration position. All amplifiers are in the radial output control mode when the spool is displaced (up or down) relative to the stack of discs (by acceleration force). With approximately 0.020 inch displacement either way, the vortex control mode is exclusively radial and for lesser displacements, the mode is a combination of tangential and radial. The amplifier configuration is based on a Harry Diamond Laboratories (HDL) amplifier design supplied

by that organization for the application -- after the original Bowles amplifier design was found to have been inadequate in tests conducted by Harry Diamond Laboratories, ERADCOM.

Flow is ported through the various other discs (shown in enlarged form in Appendix A, Figures A-9 through A-18) from the inlet of the compression valve through the applicable amplifier(s) and to the vortex valve. The vortex valve is also made up of discs as shown in Figures A-19 and A-20. A series of the vortex type discs form the vortex chamber into which an amplifier output flow is directed, either to the radial (low resistance), or tangential (high resistance) ports, or to both in proportion to the acceleration command.

MODIFIED COMPRESSION VALVE

The compression valve configuration shown in Figure 12 preceding the final design was very similar to the final design. The main difference between them, which improved performance approximately 200 percent, was the increased inlet/outlet flow channel sizes incorporated in the final configuration. All discs in the final design were revised to increase the flow area of the inlet and outlet channels by a factor of four.

Another minor difference between the final and previous compression valves was in the method of fabrication of the vortex valve. Except for the increased channel flow area for the tangential and radial ports,

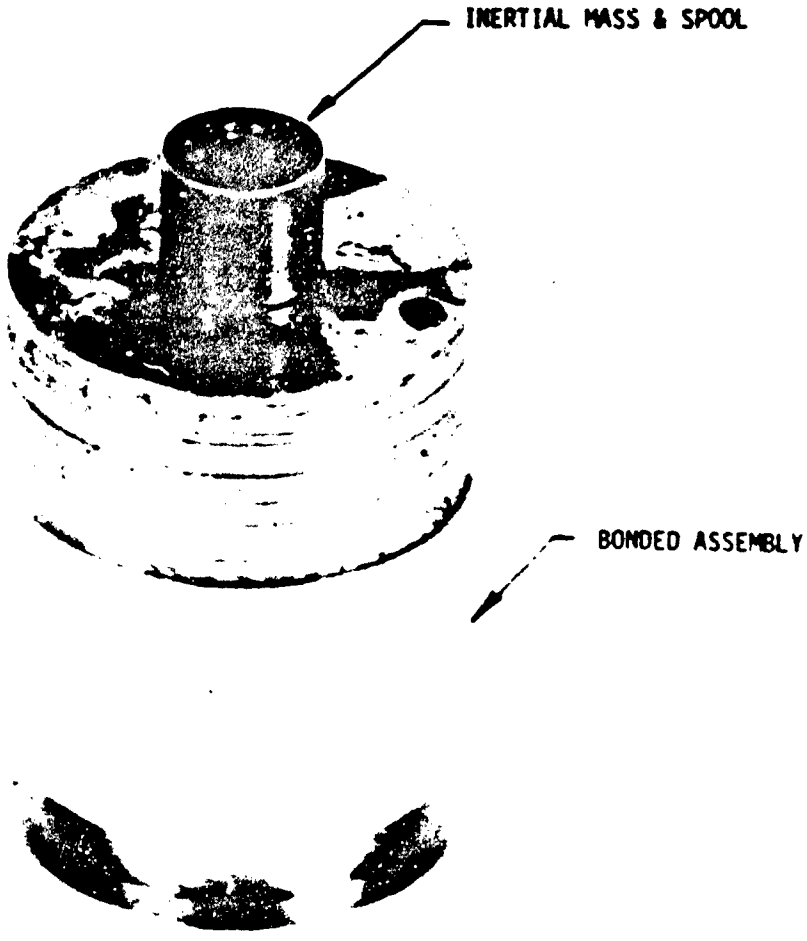


FIG. 12 - MODIFIED COMPRESSION VALVE
MATCH ASSY., P/N 3320977

the vortex valve sizing is the same. The vortex valve for the final design was formed by stacking discs as shown in Figures A-19 and A-20, whereas the previous vortex valve was machined from bar stock. However, even this solid piece vortex valve was fusion bonded to the stack of discs.

Also, this valve configuration used a BG-42 stainless steel spool rather than the tungsten alloy material selected for the final configuration. With the steel spool, the P/N 3318972-1 disc spring was used which was designed for a spring rate of 0.216 pound/inch.

The final difference was total nozzle area reduction to 0.0024 $in.^2$ as determined from "breadboard" tests conducted with the amplifiers and the Vortex Valve separated from each other so that the total nozzle area could be varied by adding or removing amplifier discs.

ORIGINAL COMPRESSION VALVE

The original compression valve shown in Fig. 13 appears to be very similar to the two designs that followed (previously described) because all were sized for the same installation envelope (Ref. Fig. 14). Also, the beam deflection amplifier section of the large cylinder in Fig. 13 (from the top down to the obvious parting line) was fabricated by stacking photo-etched discs as shown in Fig. 15. However, the beam deflection amplifier configuration was based on the Bowles design as shown in TARADCOM Report No. 12086. In addition, the vortex valve configuration was based on the Bowles design, which depended on development of the fluid swirl in the amplifier block as shown in the TARADCOM report. In Fig. 13, the comparable thing was done as can be noted by the group of discs between the amplifier group and the one-piece vortex valve. Unlike the next two designs that followed, in which the vortex valve was fusion bonded to the stack of discs, the vortex valve in the original design was secured to the stack by the three roll-pins which can be seen at the top surface in Fig. 13. The Vortex Valve and the amplifier block were loaded together when mounted in the fixture to form a metal-to-metal seal between the amplifier block and the Vortex Valve.

When this design showed almost no resistance change with spool position, a different Vortex Valve was used with the Bowles Amplifier block. The Vortex Valve is shown in Fig. 16. It is smaller than the others because its chamber diameter had to be smaller than the swirl passages in the amplifier block. Its outlet diameter was chosen to maintain the 5.8 to 1 ratio of chamber diameter to outlet diameter used in the previous design.

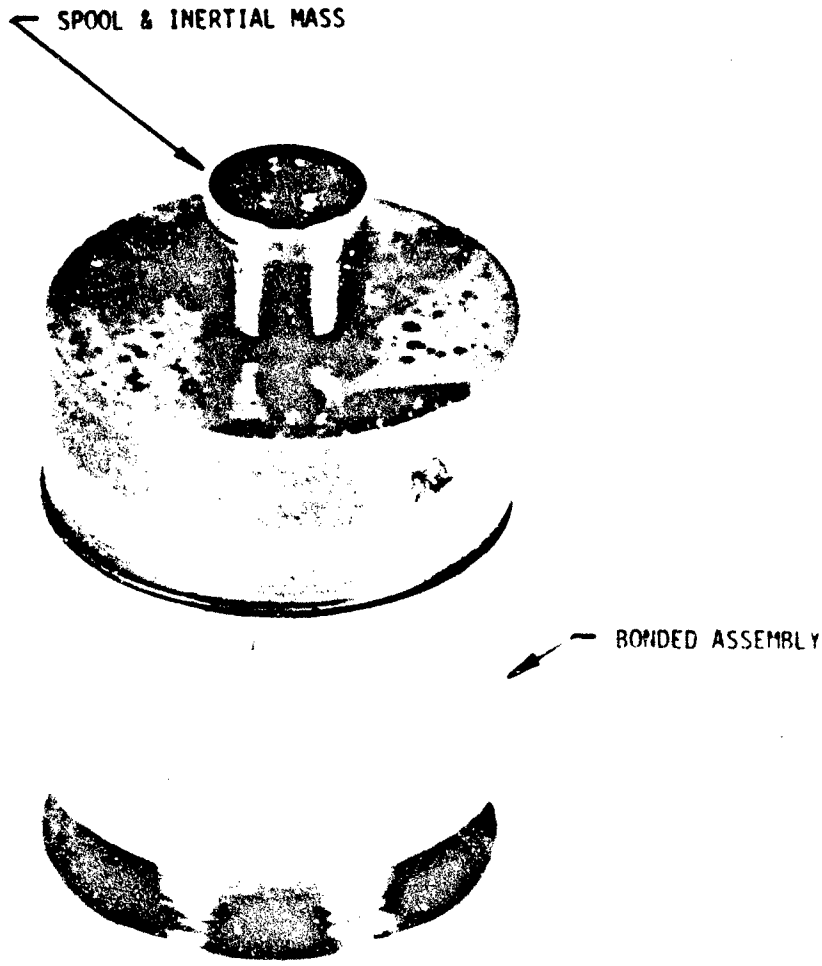


FIG. 13 - ORIFICIAL COMPRESSION VALVE
MATCH ASSY., P/N 7322931

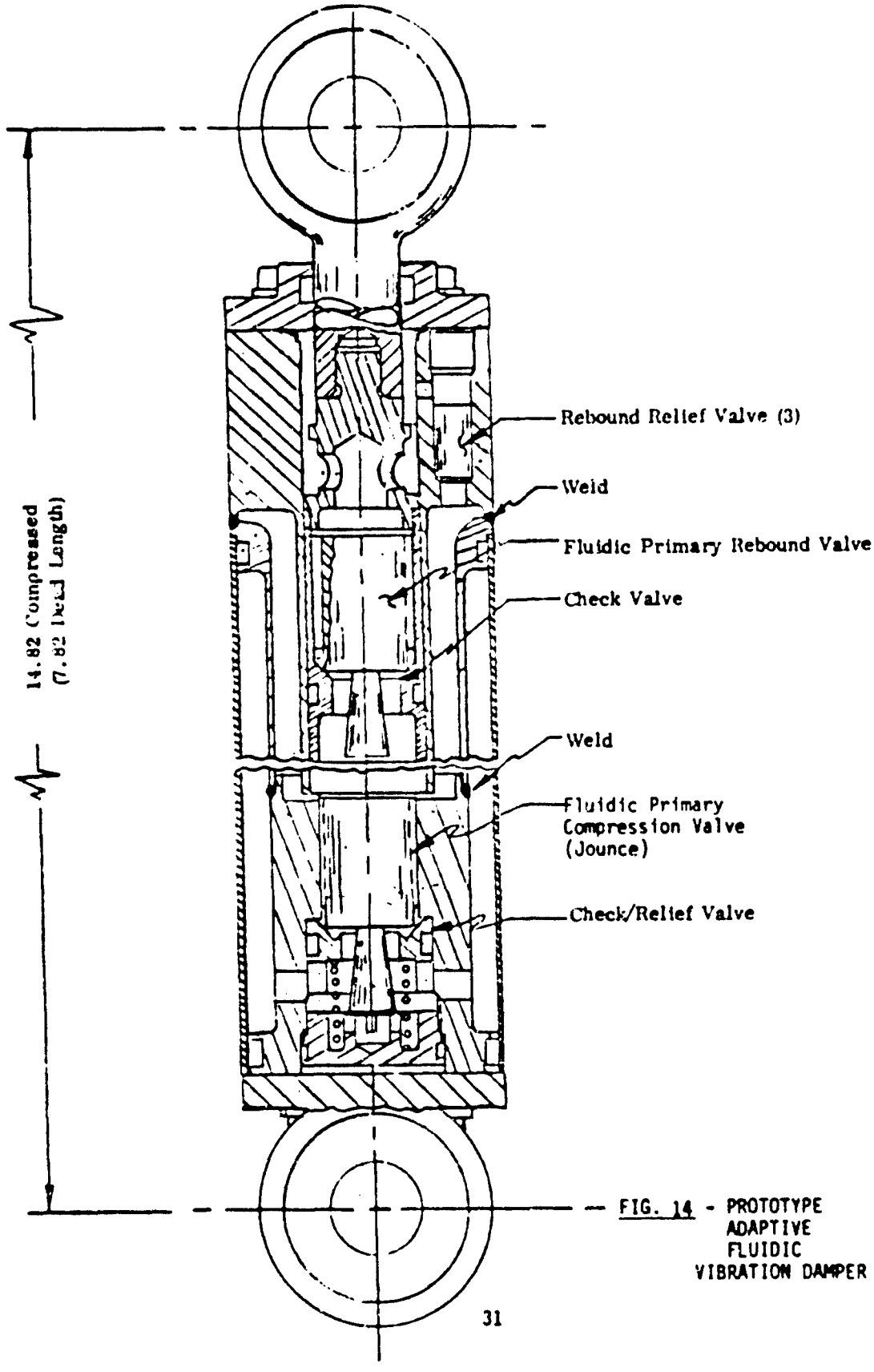


FIG. 14 - PROTOTYPE ADAPTIVE FLUIDIC VIBRATION DAMPER

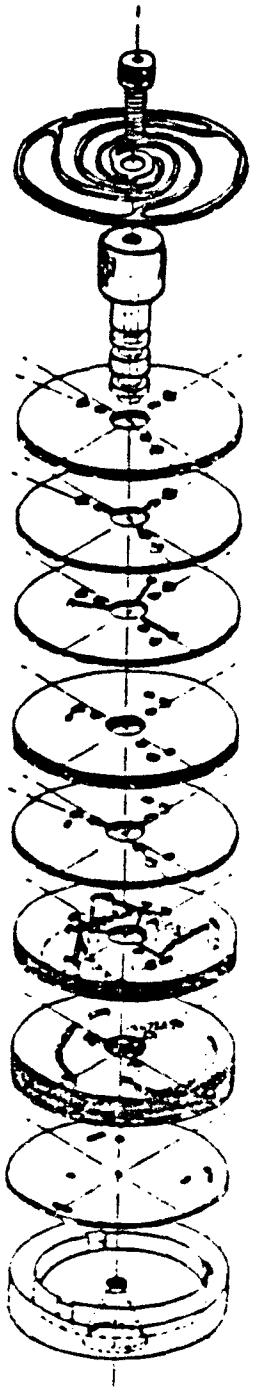
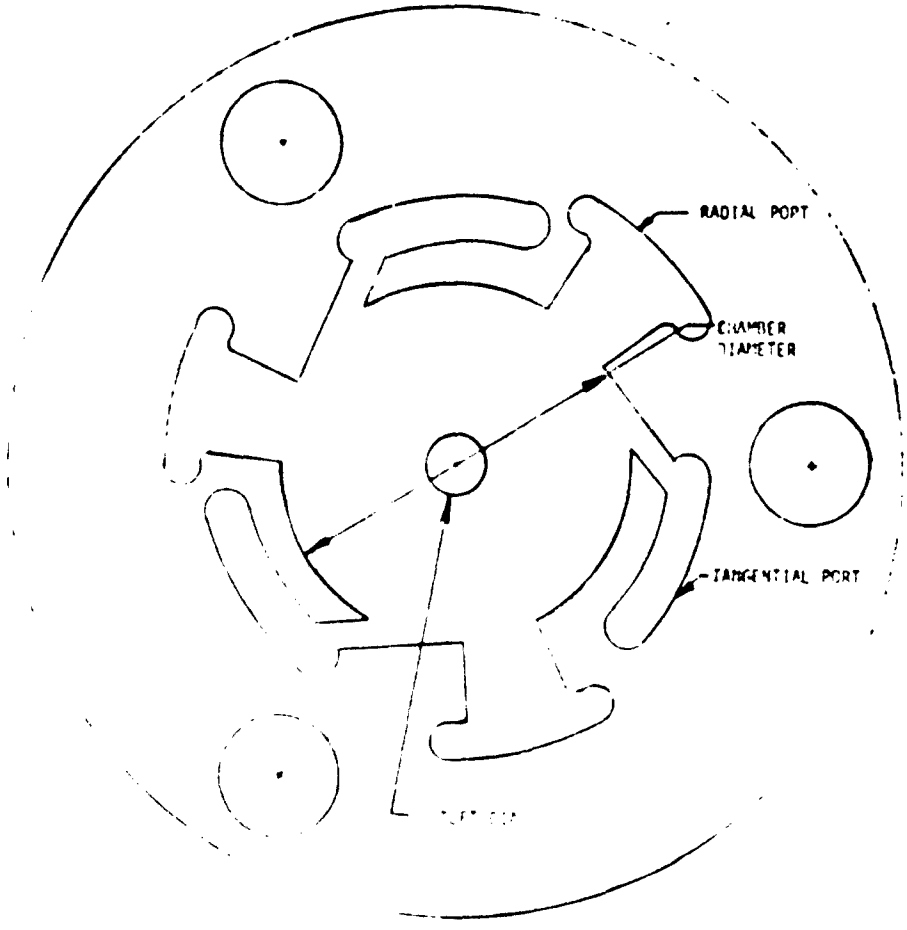


FIG. 15 - ADAPTIVE FLUIDIC COMPRESSION VALVE, PART NO. 3318981



VORTEX VALVE

FIG. 16

APPENDIX A

SUPPORTING DATA AND PHOTOGRAPHS

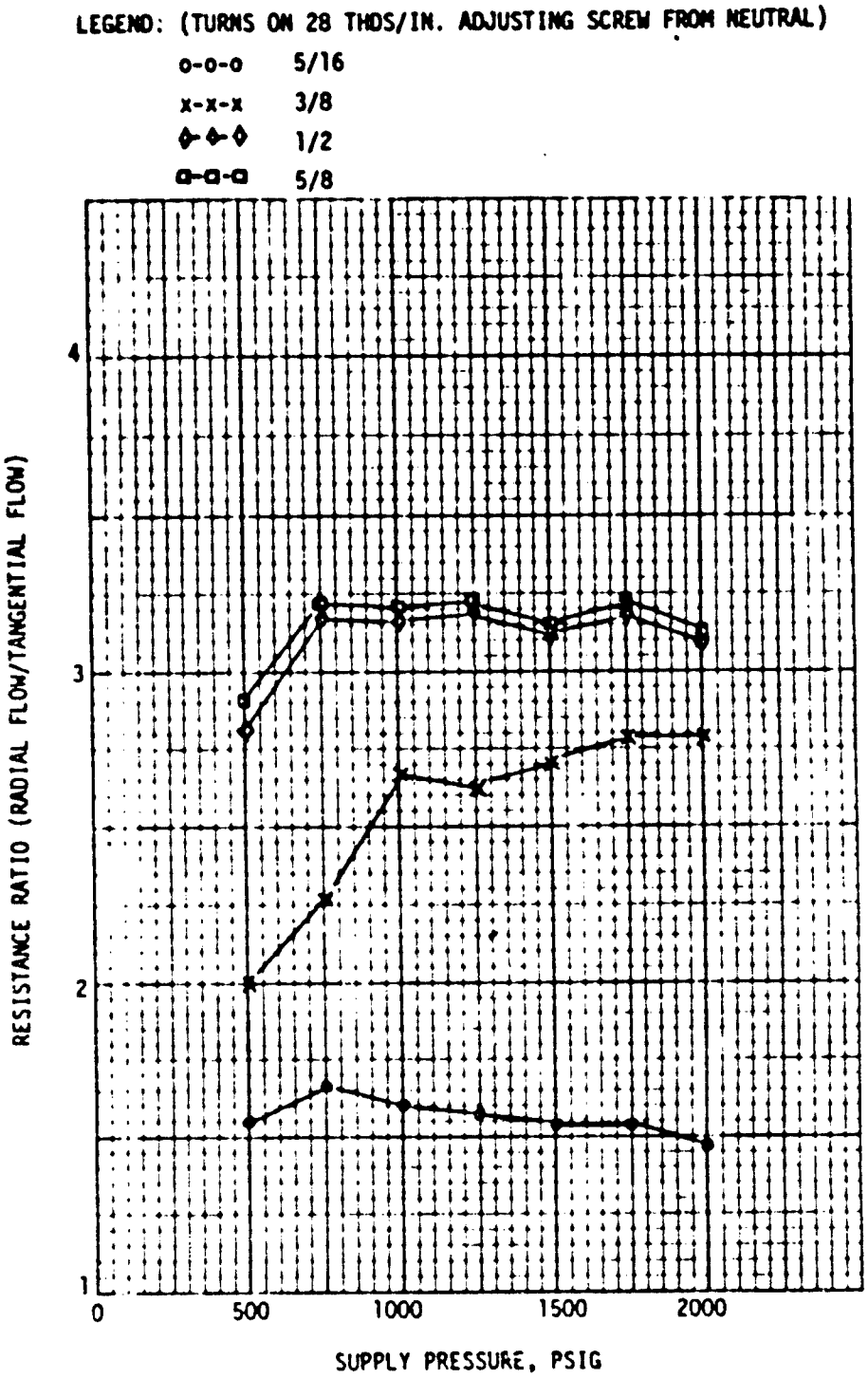


FIG. A-1 - STATIC TEST RESULTS, MATCH ASSEMBLY NO. 1, P/N 3320997-1

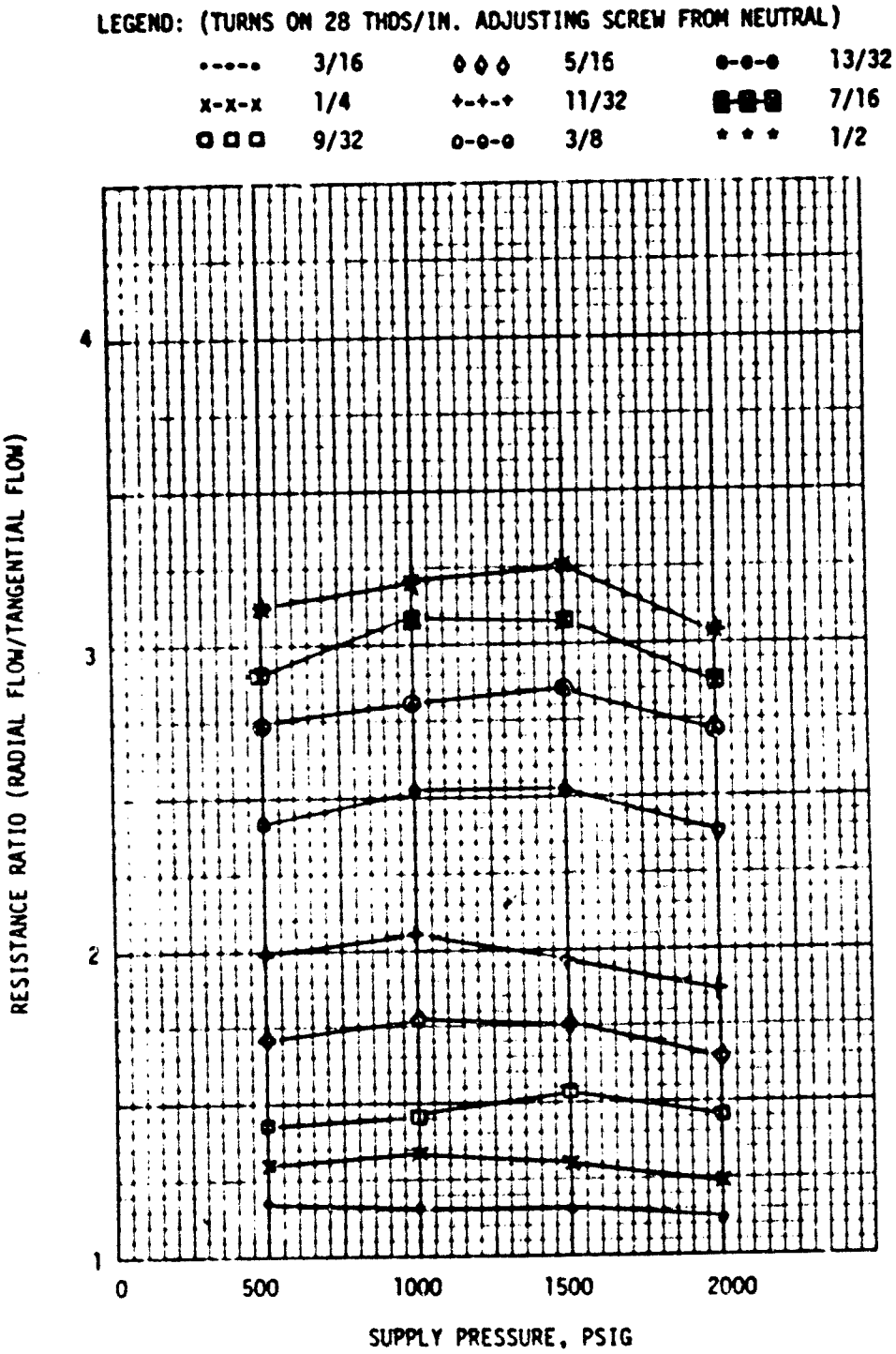


FIG. A-2 - STATIC TEST RESULTS, MATCH ASSEMBLY NO. 2, P/N 3320997-1

LEGEND: (TURNS ON 28 THD/IN. ADJUSTING SCREW FROM NEUTRAL)

•-•-• 1/8

x-x-x 1/2

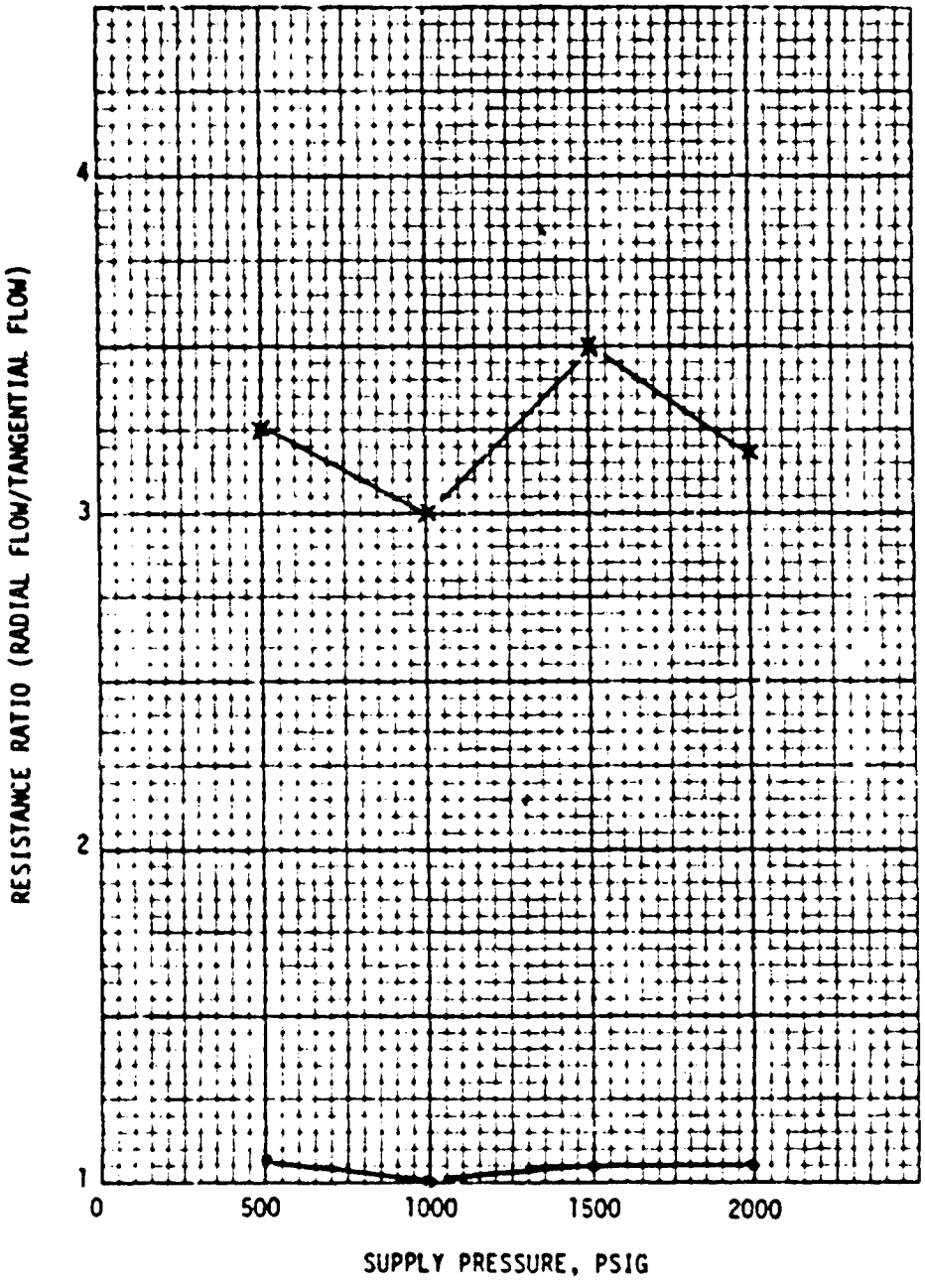


FIG. A-3 - STATIC TEST RESULTS, MATCH ASSEMBLY NO. 3, P/N 3320997-1

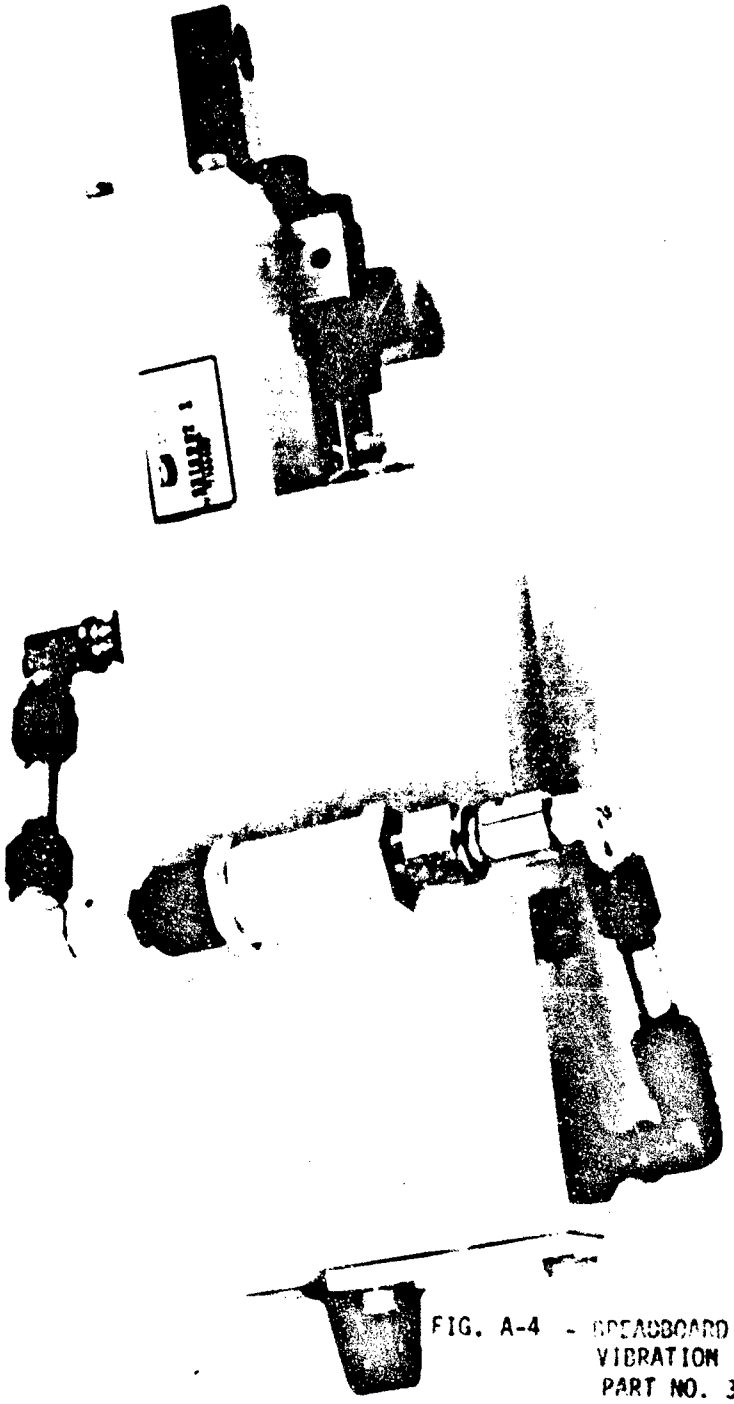
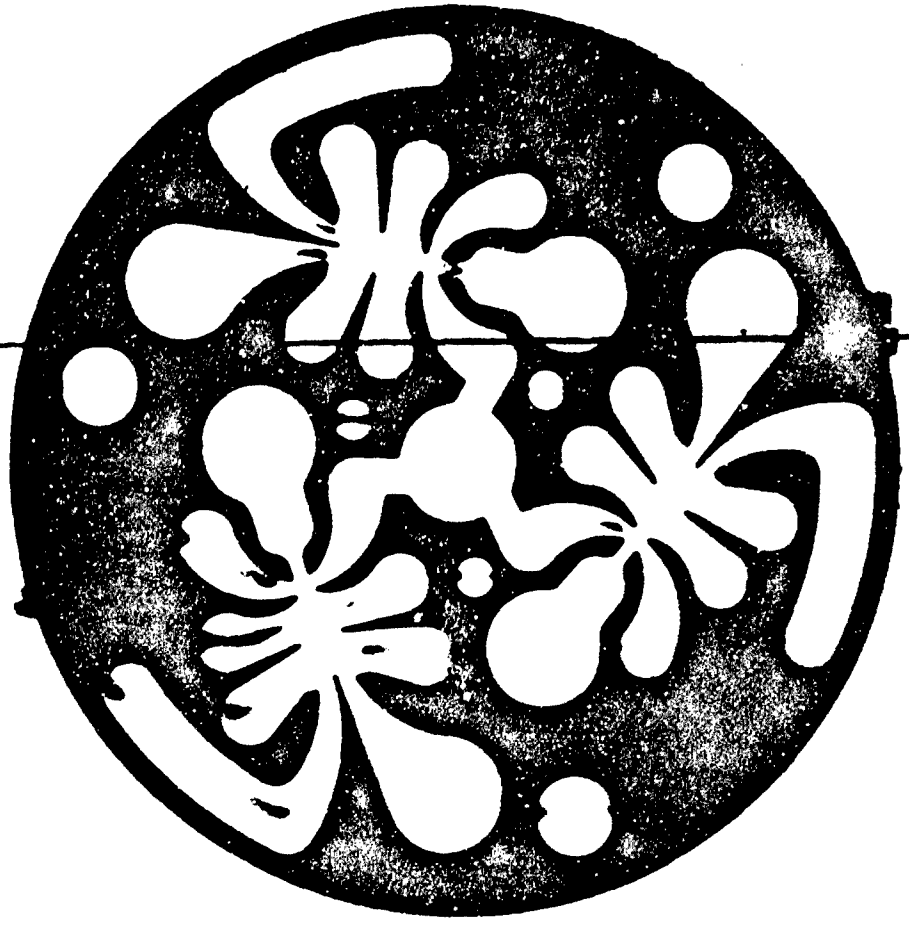


FIG. A-4 - BREADBOARD ADAPTIVE FLUIDIC VIBRATION DAMPER PART NO. 3318982-1

FIG. A-5 - BREADBOARD ADAPTIVE FLUIDIC VIBRATION DAMPER PART NO. 3318982-1

TYPE 6

FIG. A-6 - AMPLIFIER DISC, P/N 3320966-1
(0.004 THICKNESS)



TYPE 7

FIG. A-7 - AMPLIFIER DISC
PART NO. 3320966-2
(0.004 THICKNESS)

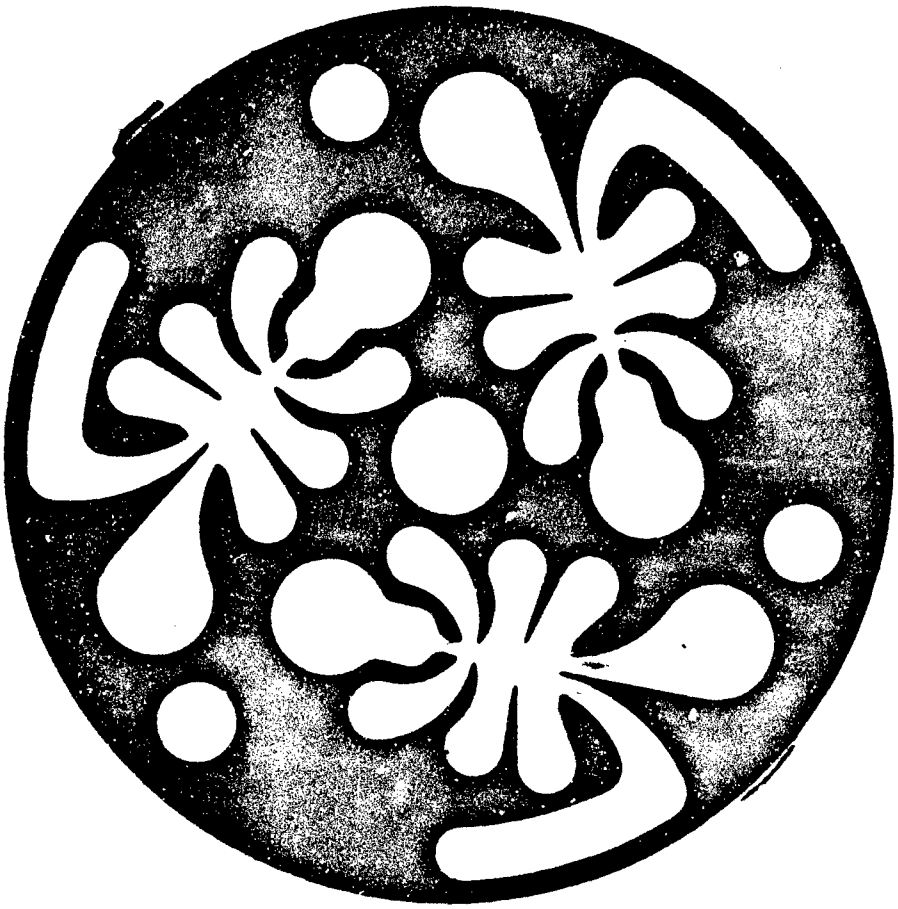


FIG. A-8 - AMPLIFIER DISC
P/N 3320966-3
(0.004 THICKNESS)

TYPE 10

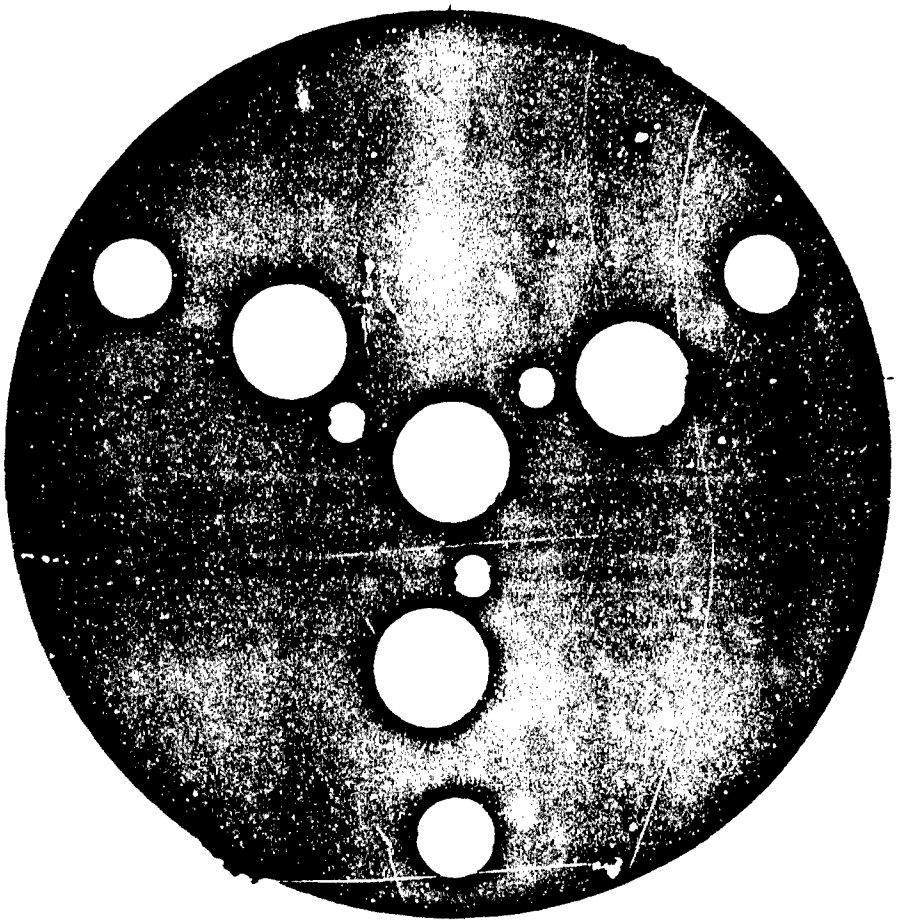


FIG. A-9 - PORTING DISC
P/N 3320967
(0.010 THICKNESS)

TYPE 1

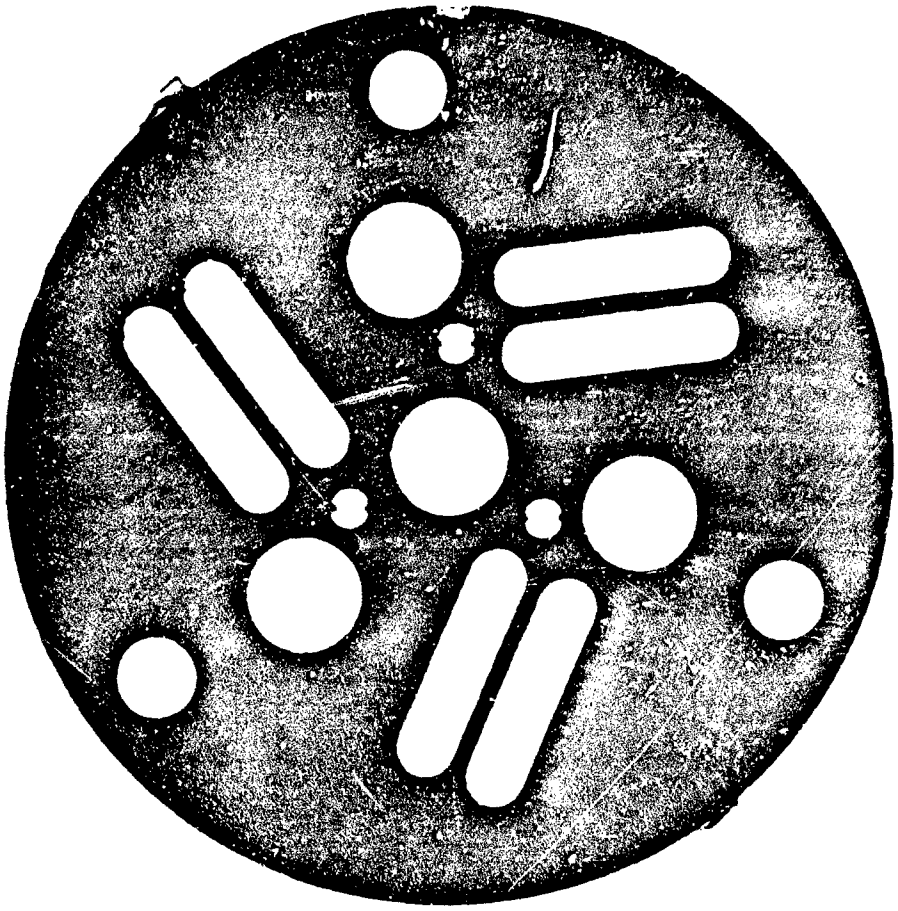


FIG. A-10 - PORTING DISC
P/N 3320968
(0.010 THICKNESS)

TYPE 2

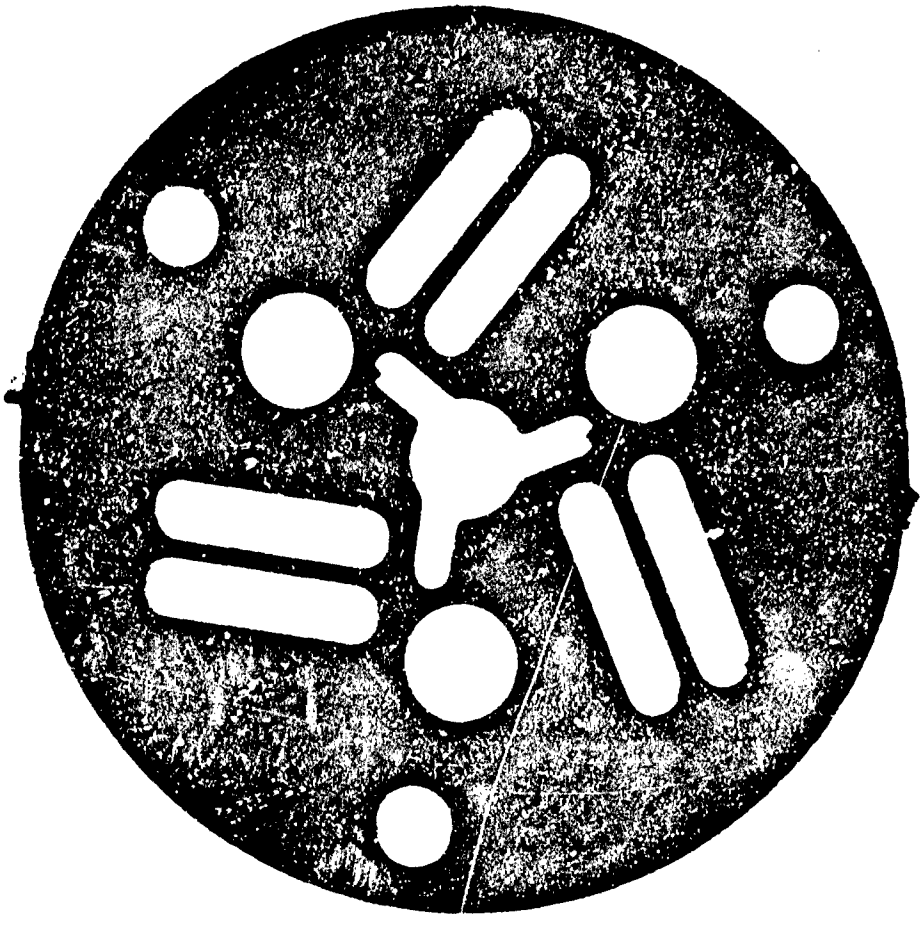


FIG. A-11 - PORTING DISC
P/N 3320969
(0.010 THICKNESS)

TYPE 3

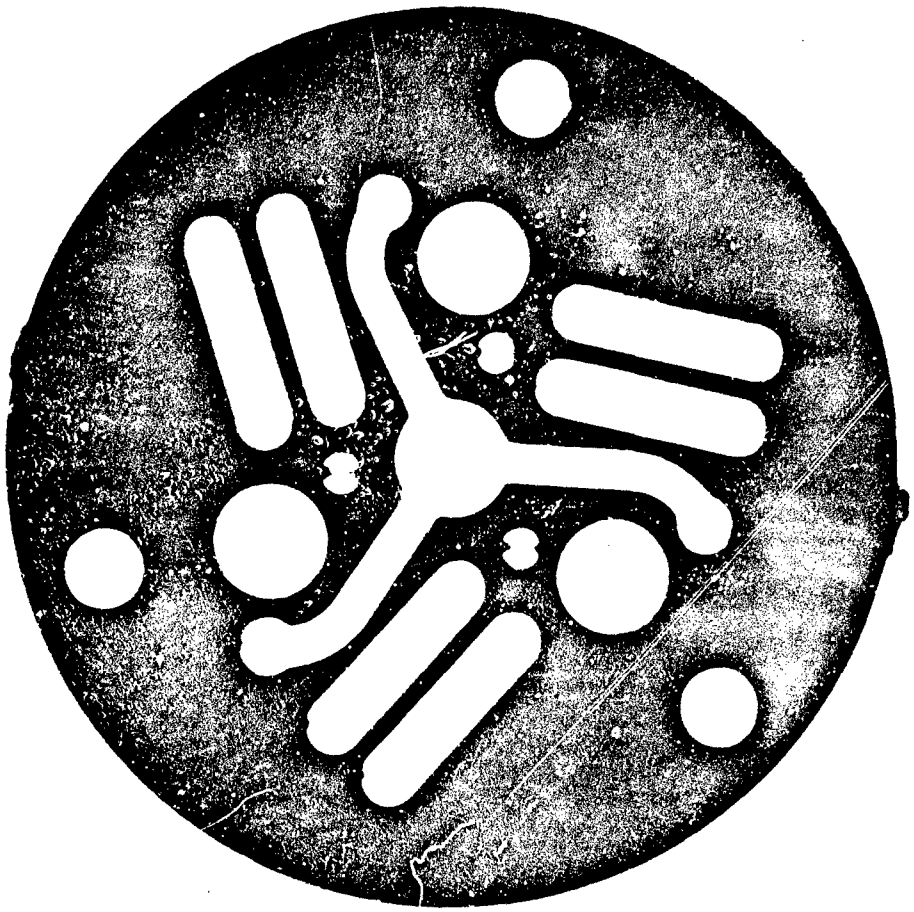


FIG. A-12 - PORTING DISC
P/N 3320979
(0.010 THICKNESS)

TYPE 4

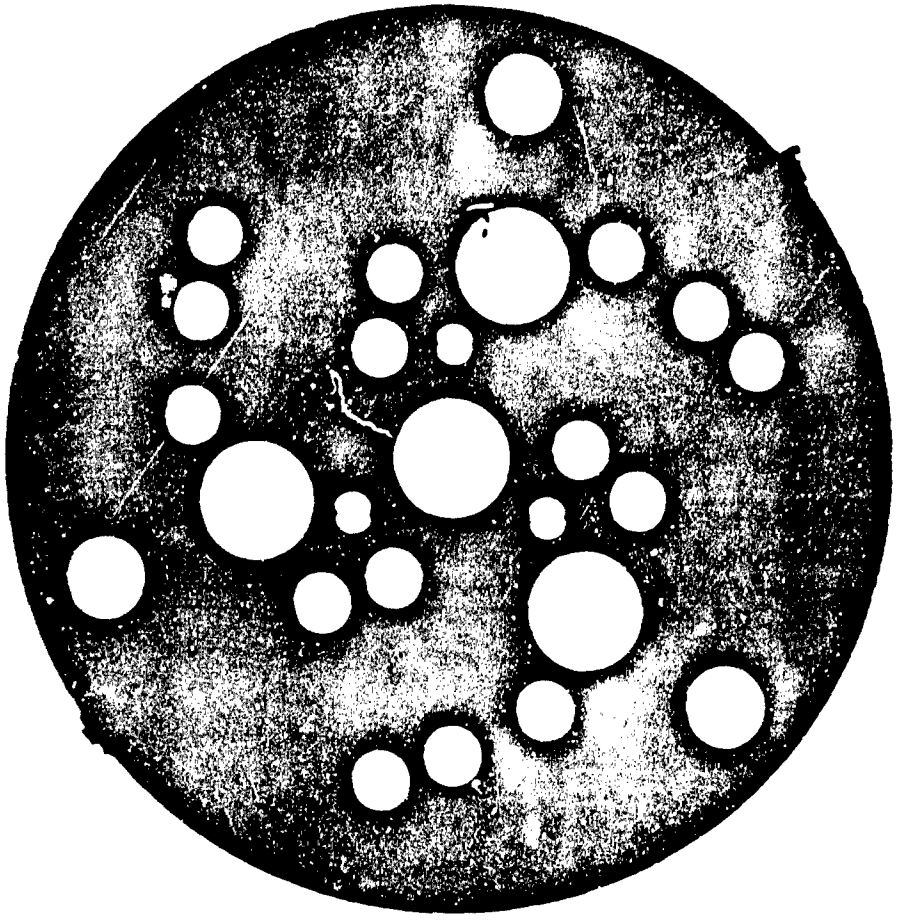


FIG. A-13 - PORTING DISC
P/N 3320991
(0.010 THICKNESS)

TYPE 5

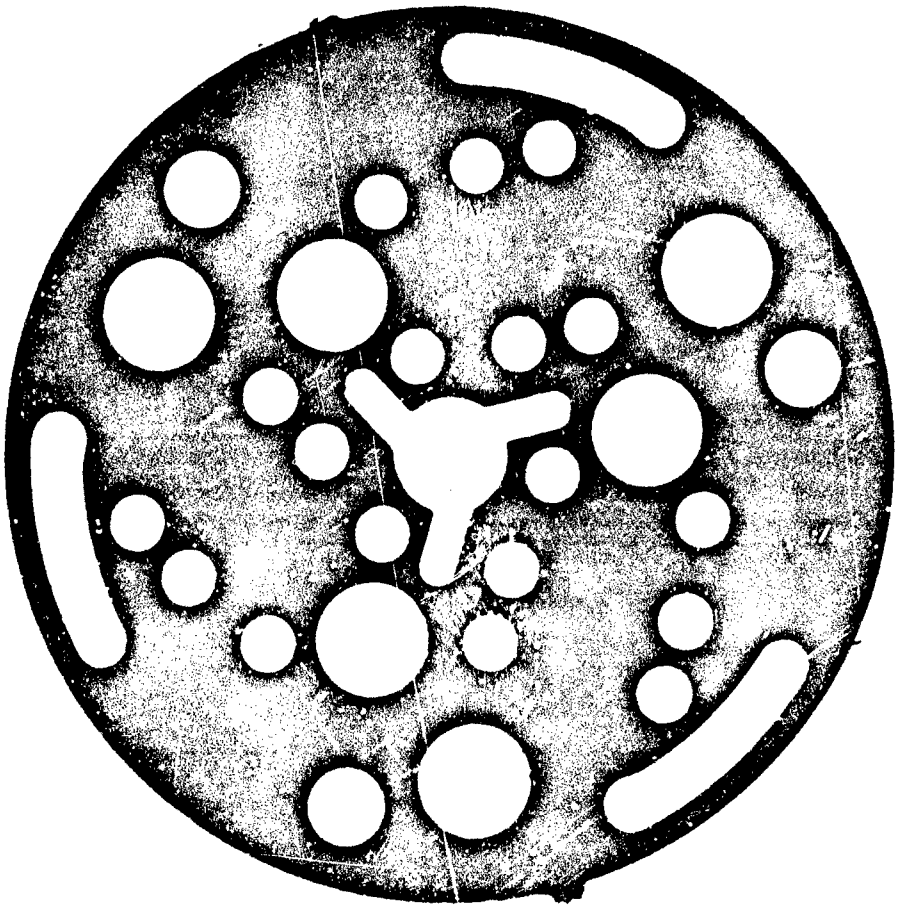


FIG. A-14 - PORTING DISC
P/N 3320992-1
(0.004 THICKNESS)

TYPE 8

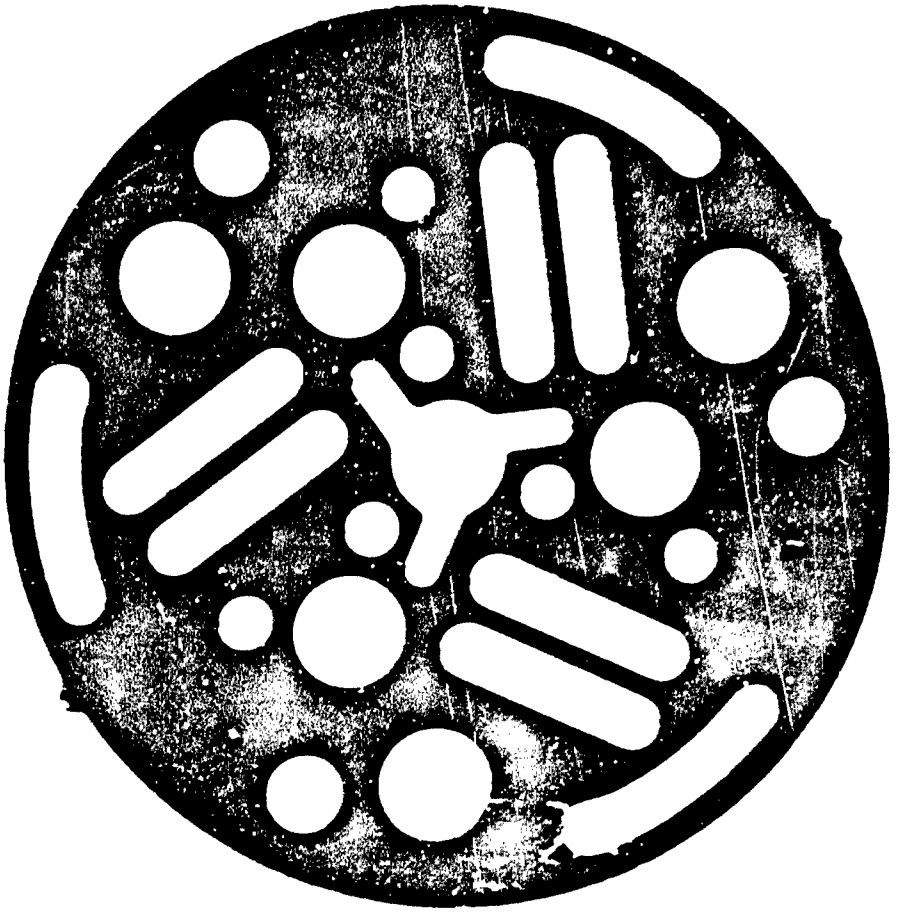


FIG. A-15 - PORTING DISC
P/N 3320993
(0.010 THICKNESS)

TYPE 9

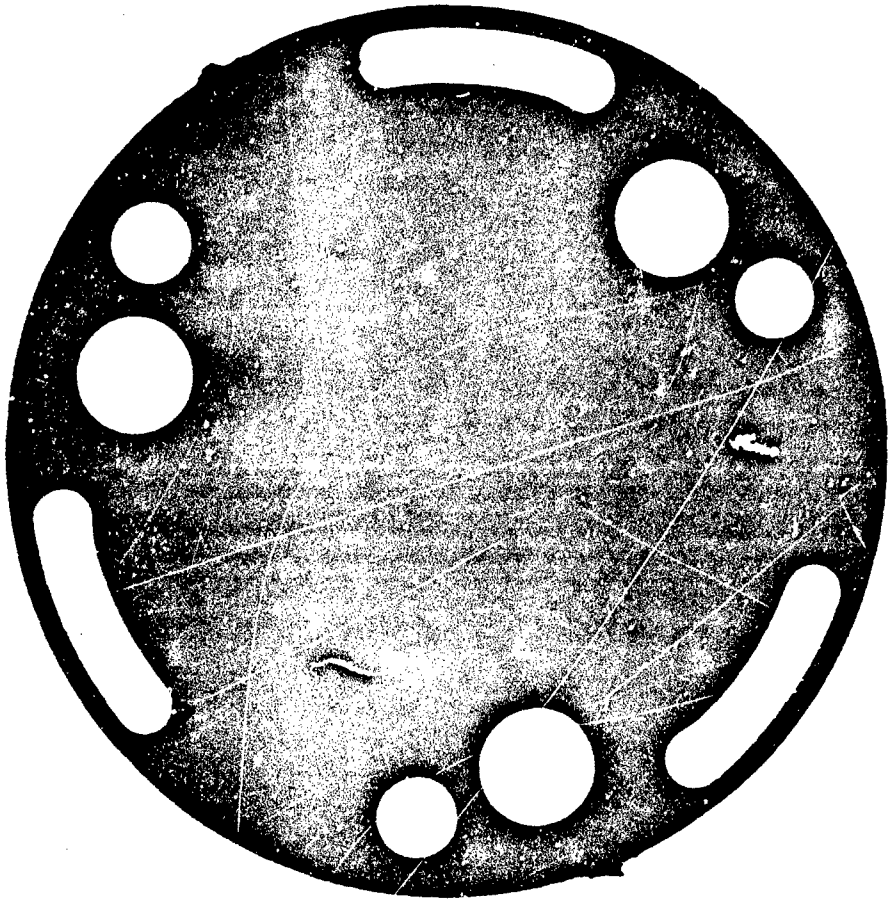
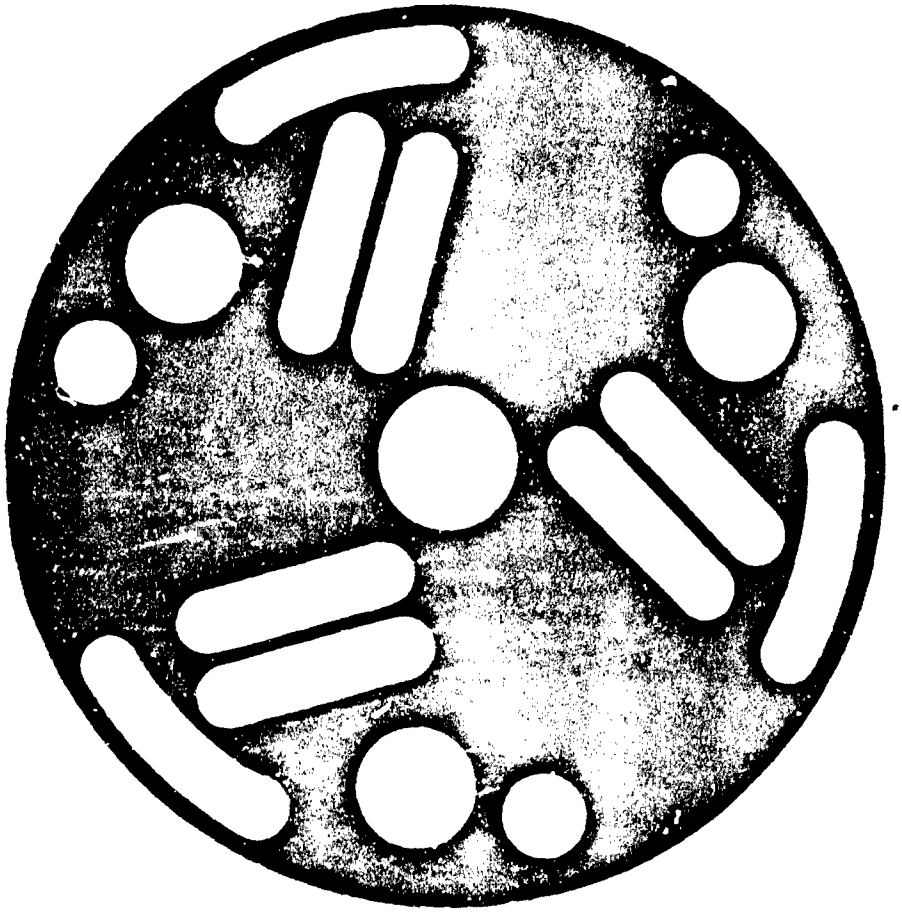


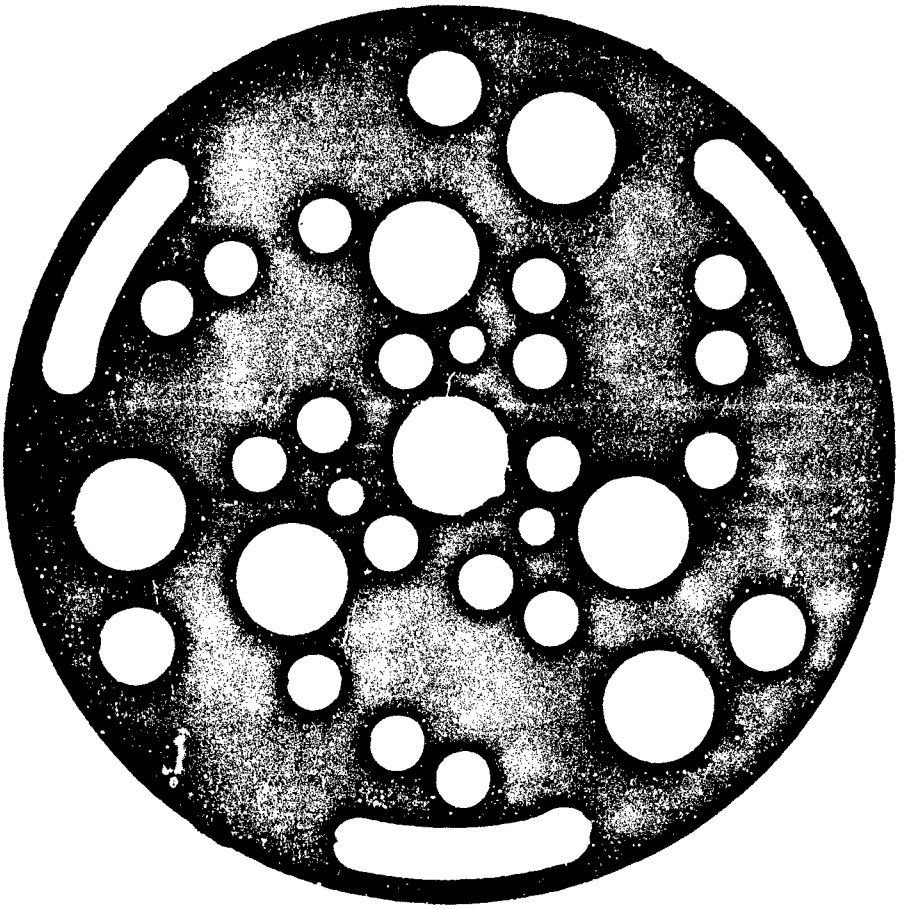
FIG. A-16 - PORTING DISC
P/N 3320995
(0.010 THICKNESS

TYPE 12



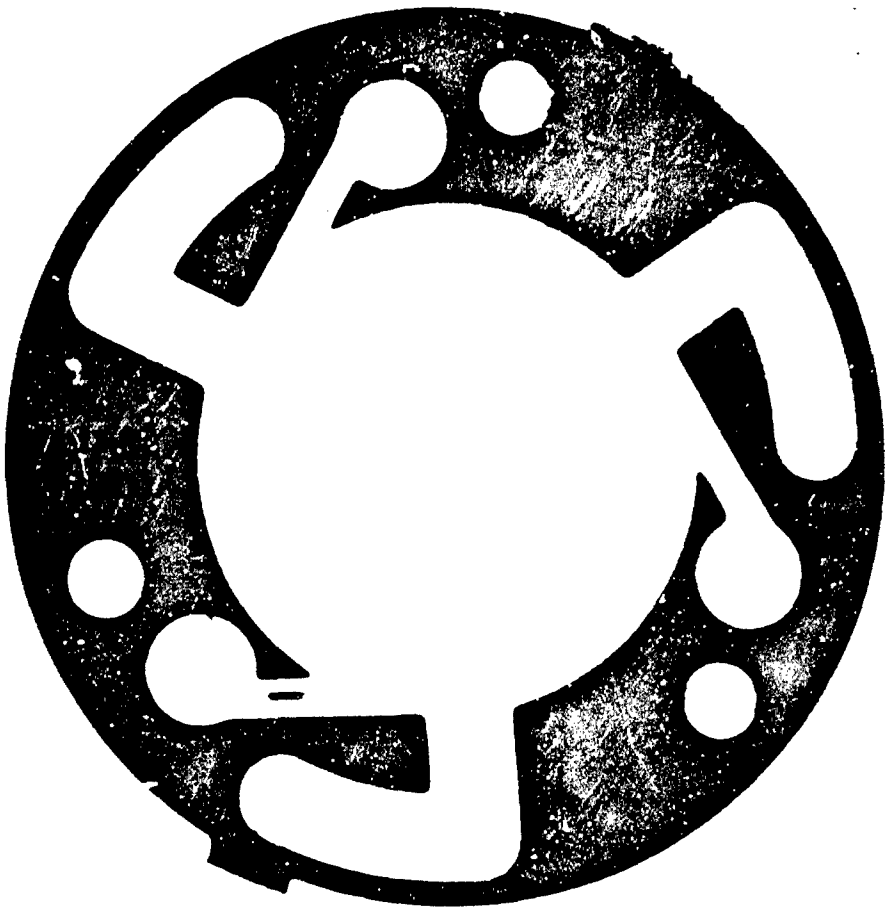
TYPE 13

FIG. A-17 - PORTING DISC, P/N 3320996
(0.010 THICKNESS)



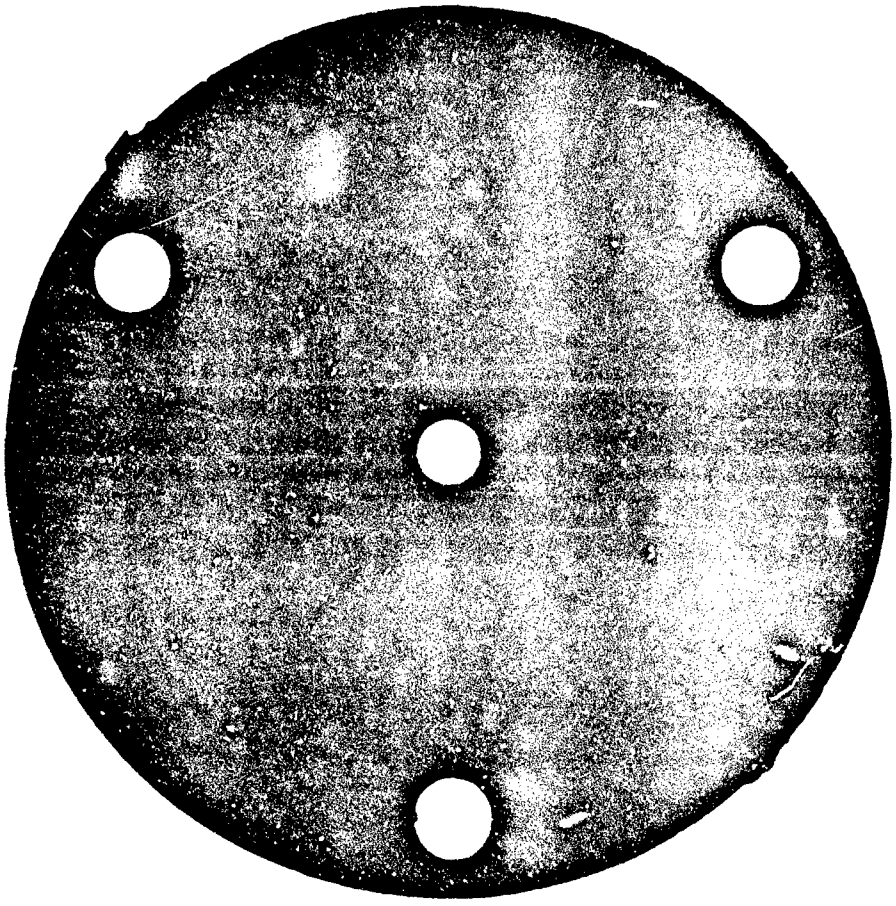
TYPE 15

FIG. A-18 - PORTING DISC, P/N 3320992-2
(0.004 THICKNESS)



VORTEX

FIG. A-19 - VORTEX VALVE CHAMBER DISC
P/N 3321149 (0.010 THICKNESS)



ORIFICE

FIG. A-20 - VORTEX VALVE ORIFICE DISC
P/N 3321148 (0.010 THICKNESS)

APPENDIX B

COMPRESSION VALVE SIZING

APPENDIX B

COMPRESSION VALVE SIZING

The maximum acceleration, "Force-Velocity" Curve, Fig. B-1, passes through 300 lbs. and 4.25 ft/second. Using the latter as a design point, paragraph 3.5.3.2 and equation 45 of TACOM Report No. 12086 gives,

$$A_1 = \frac{\dot{Q}\sqrt{Kmin}}{3C_d \rho V_1}$$

where $\dot{Q}$ = flow rate = V_s AR (12)

V_s = shock velocity ft/sec = 4.25

A_R = shock rod area = .3068 in^2

V_1 = flow velocity at Vena Contracta = $\sqrt{\frac{600}{A_r \rho}}$

ρ = fluid density = 8×10^{-5} $lb\text{-}sec^2/in^4$

C_d = orifice coefficient = .608

A_1 = power nozzle size for the beam deflection amplifiers

Kmin = of total system pressure losses in the radial or maximum acceleration mode.

Kmin = 1.75

Solving, V_1 = 4,944 in/sec

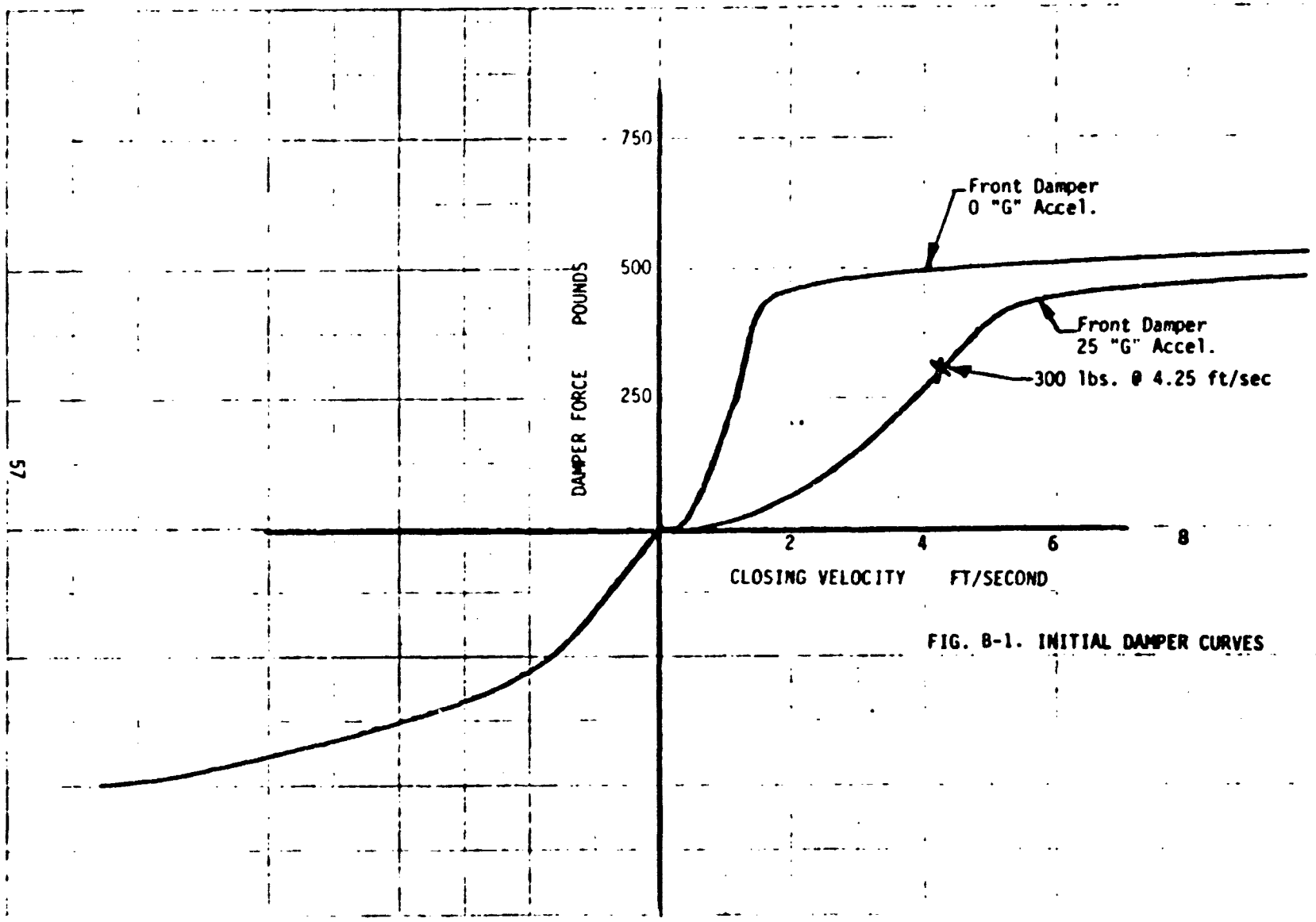


FIG. B-1. INITIAL DAMPER CURVES

$\dot{Q} = 15.65 \text{ in}^3/\text{sec}$

$$A_1 = \frac{15.65 \sqrt{1.75}}{3(.608)4944} = .002295$$

The flow area of a single power jet is A_1.

$A_1 = h_1 W_1$

where h_1 = power jet nozzle height

W_1 = Power jet nozzle width

It is desired to have $h_1 = 2W$

$\therefore \quad W_1 = \sqrt{A_1/2} = .03388$

The vortex outlet area is equal to 3 A_1; thus, the vortex outlet diameter $= d = \sqrt{\frac{.3A_1}{.7854}} = \underline{.09363 \text{ in}}$

The vortex chamber diameter is then required to be,

$D = 5.8\ d = .5431$ in

The vortex chamber height is then

$\underline{.4\ (D) = .2172}$

The preliminary Vortex Valve was sized as noted above.

During the design of the actual compression valve, a new force-velocity curve was required by TARADCOM (see Report No. 12226, Fig. 6-12 and 6-13) as shown in Fig. B-2.

Using 200 lbs. and 2.17 ft/sec as a design point and using the equations above, the Vortex Valve is sized as follows:

Outlet Diameter	=	.074"
Chamber Diameter	=	.4294"
Chamber Height	=	.1718"
Amplifier Power Jet Width	=	.02679"

Due to subsequent design changes (HDL amplifiers), packaging requirements, and recommendations by HDL, the final sizing was established as follows:

Vortex Outlet Diameter	=	.052"
Chamber Diameter	=	.419"
Chamber Height	=	.052"
Power Nozzle Size	=	.010 x .040 (12 amplifiers),

resulting in a nozzle area to vortex outlet area ratio of 2.260 to 1.

Maximum static compression valve performance, resistance ratio, was found using a nozzle size of .010 x .020 (12 amplifiers) for a nozzle area to vortex outlet area ratio of 1.13 to 1.

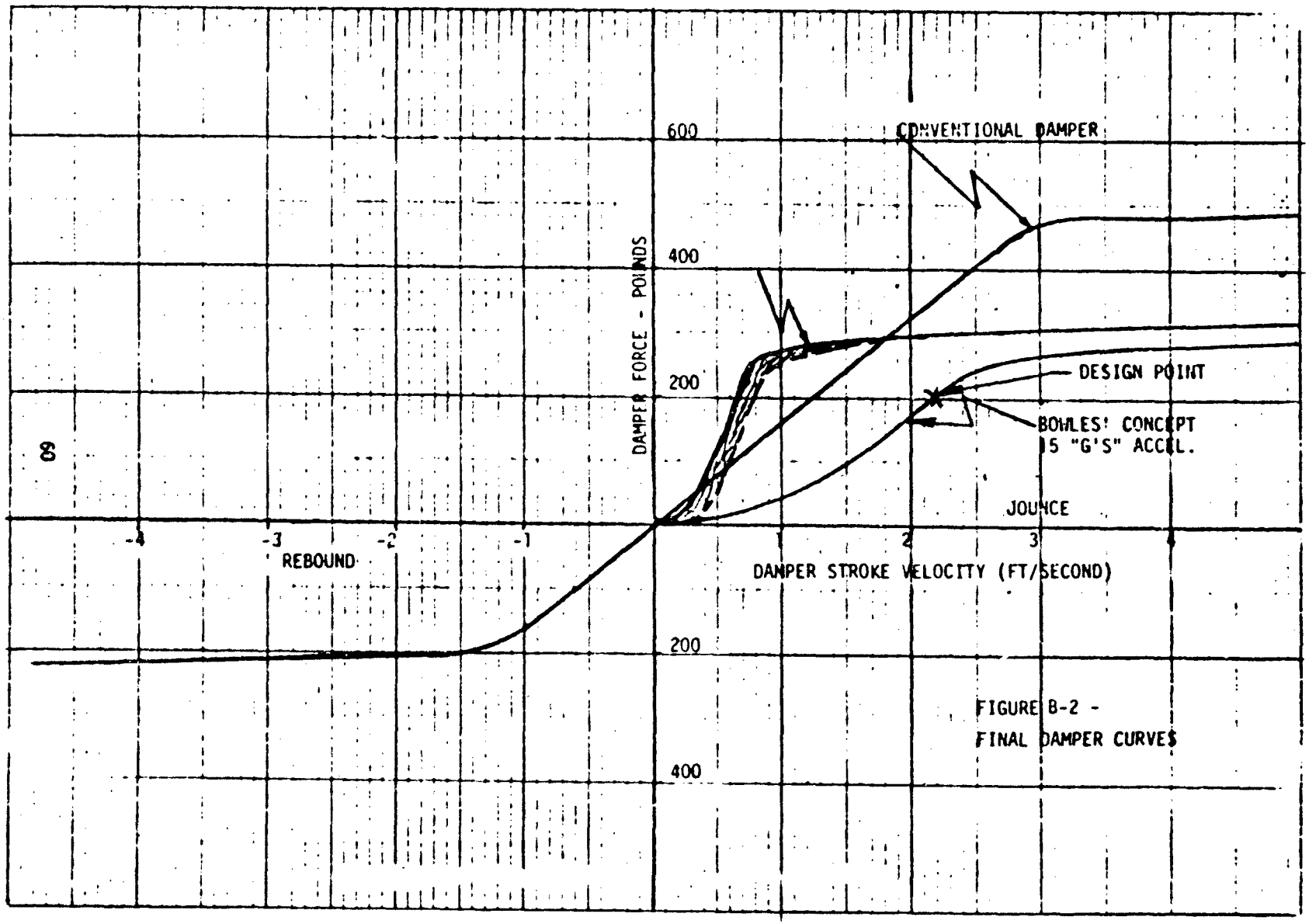


FIGURE B-2 -
FINAL DAMPER CURVES

The damping orifice size for the inertial valve to achieve a break frequency of τ = .1943 second is based on the displacement of the valve with respect to the wheel which is given by:

$$X = \frac{S \quad Vw}{S^2 + (D/M)S + K/M}$$

where X = inertial valve displacement with respect to the wheel displacement

Vw = wheel velocity

K = valve centering spring rate

M = valve mass

$M/K = 33.33 \times 10^{-6}$

The first order break time constant at τ = .19 second.

Solving for the lowest root of the above equation,

$$\frac{-D/M + \sqrt{D/M^{\,2} - 4\ K/M}}{2} = 1/.190$$

$$(D/M)^2 - 4\ K/M = (D/M)^2 - 21.06\ D/M + 110.9$$

$$21.06\ D/M = 4\ K/M + 110.9$$

$$\underline{\underline{D/M = 5704}}$$

Using $M = 7.922 \times 10^{-6}$ lb-sec^2/in.

$$D = .04462$$

The damping force is caused by the flow generated pressure drop across an orifice attached to the valve. The flow through the orifice is given by

$$q = 103\ A_o \sqrt{\Delta P}$$

The relative velocity of the valve is given by

$$V = q/A_1$$

The damping force is given by

$$F = A_1 \Delta P$$

The damping coefficient, D, is given by

$$D = F/V = A_1^2 \Delta P/q$$

Combining the above 4 equations,

$$D = \frac{A_1^2 \sqrt{\Delta P}}{103\ A_o}$$

Assuming ±3 g load,

$$\Delta P = \frac{7.822 \times 10^{-6} \times 3 \times 386}{(.1)^2\ \pi/4} = 1.153$$

$$\therefore \quad D = \frac{.6431 \times 10^{-6}}{A_o} = .0462$$

$$A_o = \frac{.6431 \times 10^{-6}}{.0462} = 13.92 \times 10^{-6}$$

The damping orifice diameter is therefore,

D_o = .00421 in.

The predicted frequency response is shown in Figures B-3 and B-4.

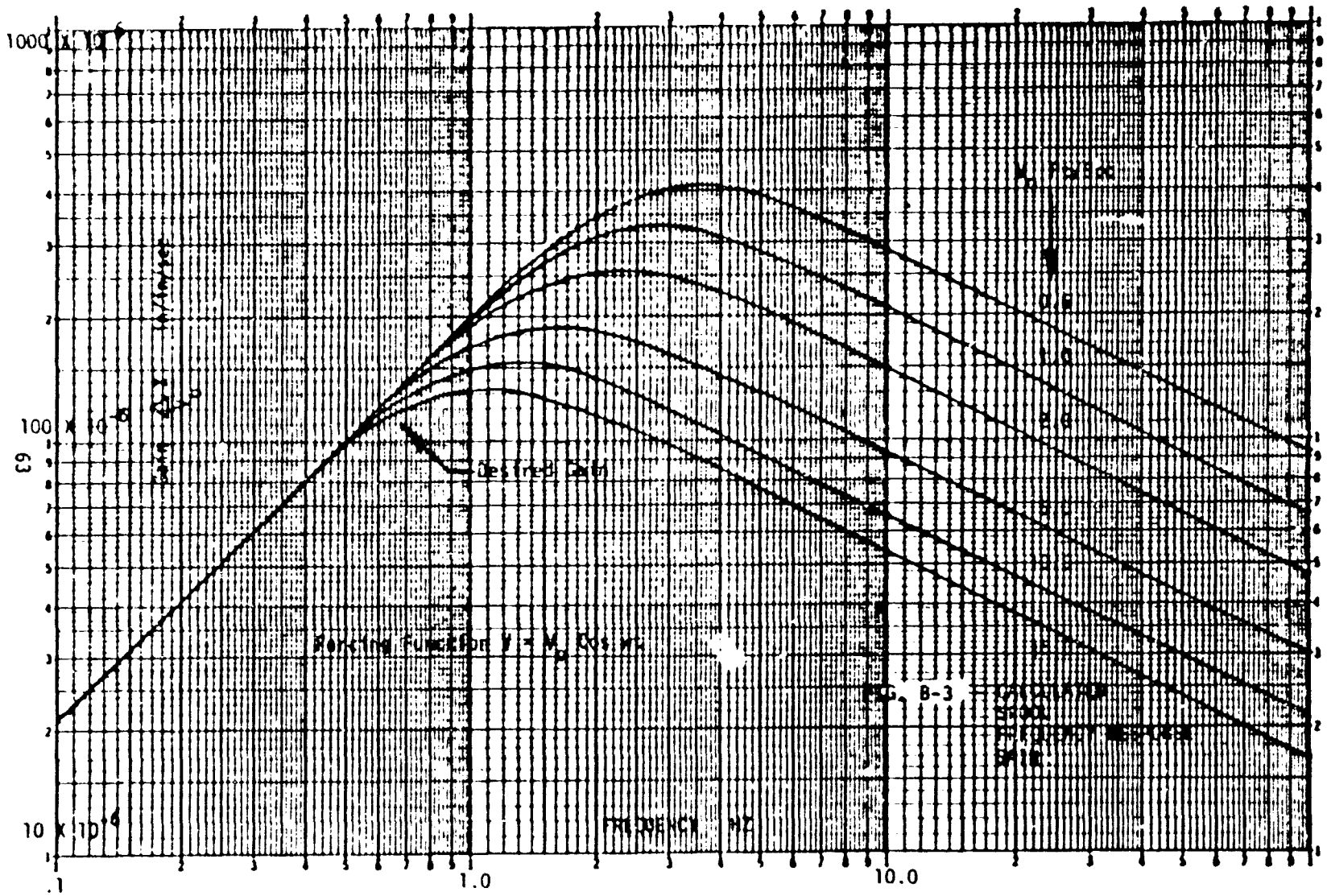

1000 x 10^-6
100 x 10^-6
10 x 10^-6
Gain Q/Y, in/inch
Desired Gain
Forcing Function Y = M_o Cos ωt
FREQUENCY, HZ
.1
1.0
10.0

FIG. B-3 — [illegible]

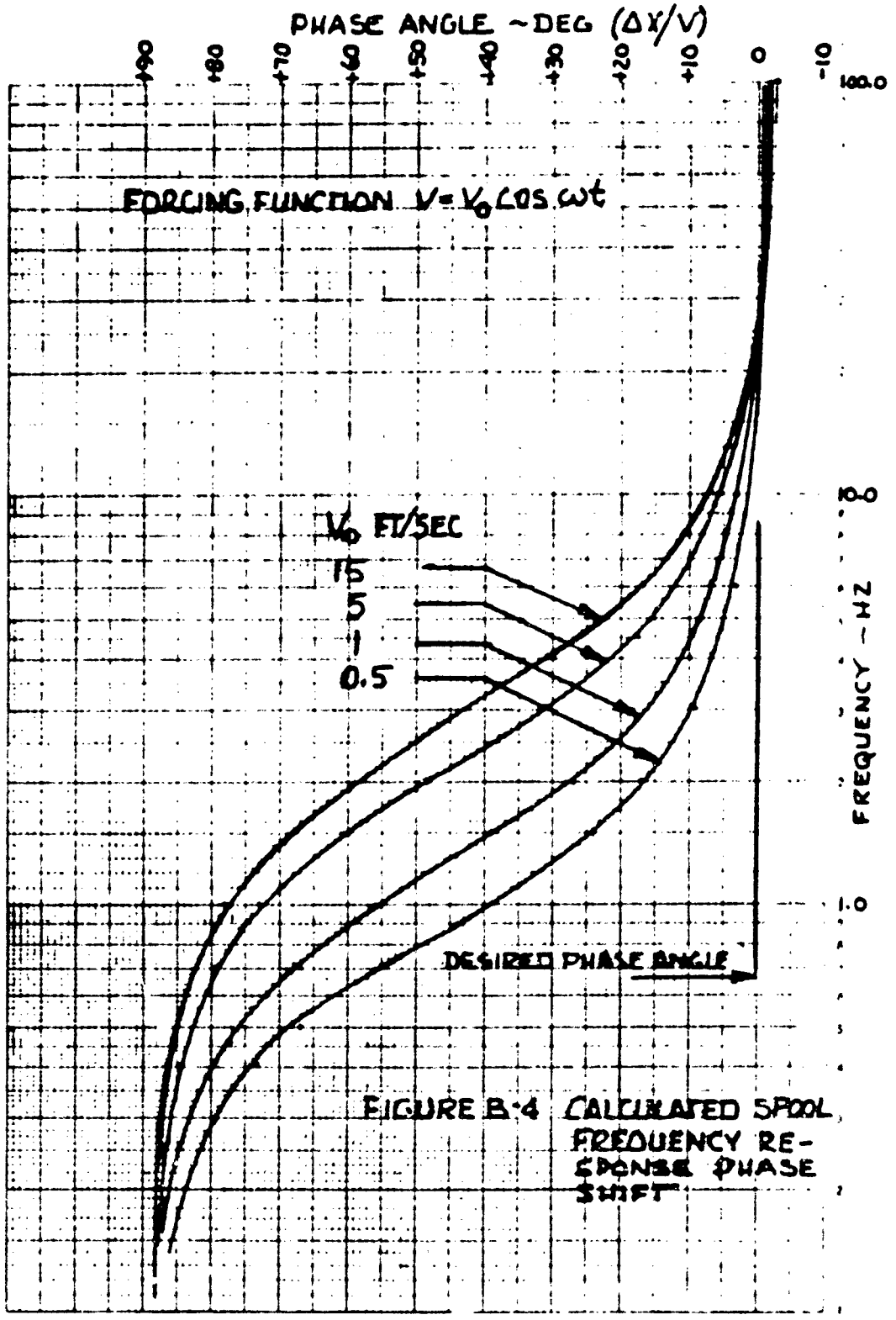


FIGURE B-4 CALCULATED SPOOL FREQUENCY RESPONSE PHASE SHIFT

APPENDIX C

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APPENDIX D

DD FORM 1473

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REPORT DOCUMENTATION PAGE

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2. GOVT ACCESSION NO.

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle): A FLUIDIC ADAPTIVE DAMPER - BREADBOARD DESIGN, FABRICATION AND TEST

5. TYPE OF REPORT & PERIOD COVERED: FINAL, JANUARY 1979

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2. R.K. VanAusdal
3. M.M. Steele
4. S.M. Tenney

8. CONTRACT OR GRANT NUMBER(s)

9. PERFORMING ORGANIZATION NAME AND ADDRESS:
1. and 2. Bendix Corp., Electrodynamics Div., North Hollywood, CA 91605
3. and 4. See reverse side

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS

11. CONTROLLING OFFICE NAME AND ADDRESS:
U.S. Army Tank Automotion Research & Development Command, Attn: DRDTA-RKT
Warren, Michigan 48090

12. REPORT DATE: JANUARY 1979

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14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)

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18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Vibration Damper	Hydro-Fluidics
Adaptive Damper	Fluidic Compression Valve
Fluidic Adaptive Vibration Damper	Fluidic Adaptive Compression Valve
Suspension Systems	Damping Devices
Adaptive Suspension System	Shock Absorbers

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The purpose of this development effort was to achieve a practical implementation of a fluidic composite valve to replace the compression valve of a conventional shock absorber to improve vehicle control and performance. The effort was to result in the delivery of two breadboard adaptive fluid vibration dampers (shock absorbers) for Government evaluation.

Program results include the sizing, design, packaging, fabrication and testing of several modifications of a compact, inertially controlled arrangement

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of three parallel fluidic beam deflection amplifiers controlling a vortex. Fabrication techniques of the fluidic valve included photo-etching of steel amplifier discs that were diffusion bonded to form a sleeve for a conventional valve spool that serves as the inertial sensor. A unique two-electrohydraulic servovalve test circuit was produced that provided constant flow control over a wide range of load variations.

Two breadboard vibration dampers, one assembled and one unassembled, were delivered to the Government for evaluation.

BLOCK 9 (Continuation)

3. U.S. Army Tank Automotive Research and Development Command, Warren, Michigan 48090
4. U.S. Army Electronics Research and Development Command, Adelphi, Maryland 20783