An Electrically Operated Hydraulic Control Valve

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The electrohydraulic transducer used in the servos that drive the control surfaces of the NIKE missile is described and its operating characteristics are discussed. Special attention is directed to the secondary dynamic forces that exist in a high-gain device of this type and to the resulting tendency to oscillate. The application of the valve to a servo system is discussed briefly.

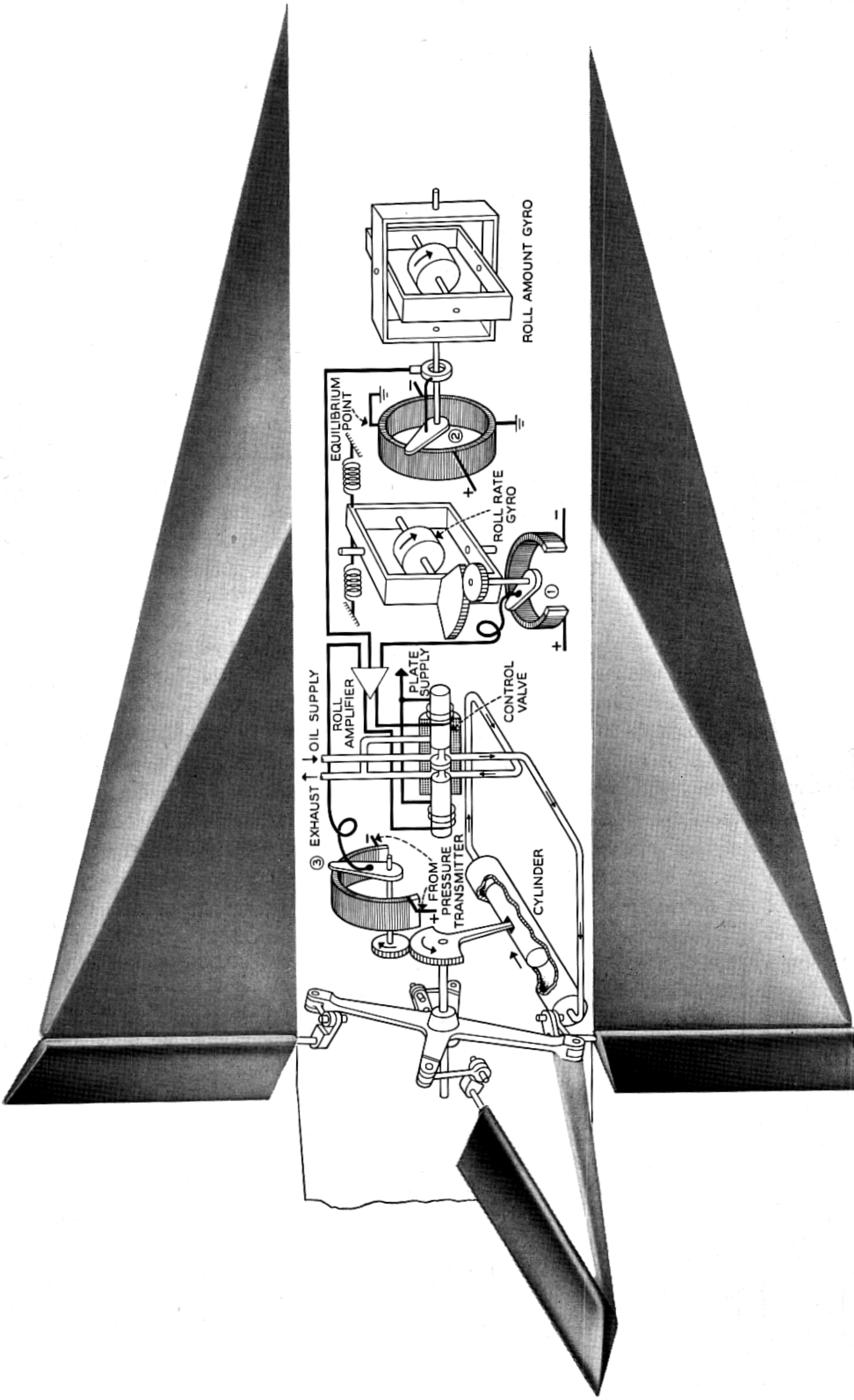
INTRODUCTION

Early in the study of the NIKE guided missile project, it became apparent that the requirements for the fin actuators could not be fulfilled by the servo-mechanisms available at that time (1945). All existing types failed to meet the combined requirements of small size, light weight, high torque, and rapid response. Further investigation showed that the development of a hydraulic servo employing an electrohydraulic transducer appeared to provide a promising solution. A control system of this type, therefore, has been developed for the NIKE missile.

The design of the transducer, or control valve, was one of the principal problems in the development of the missile control systems and is the subject of this article. The specific design of the valve that will be discussed here is known as "Model J-7", and represents the state of the development in 1950. Valves of this type, with varying degrees of modification, are used in missiles of several other projects.

APPLICATION

Fig. 1 is a simplified schematic of the roll positioning system in the NIKE missile. It is the simplest of the three applications of the valve in the missile, but will serve to illustrate the situation in which the valve operates. The purpose of the roll servo is to keep the missile in a predictable roll orientation.


ROLL AMOUNT GYRO
EQUILIBRIUM POINT
ROLL RATE GYRO
ROLL AMPLIFIER
PLATE SUPPLY
OIL SUPPLY
EXHAUST
CONTROL VALVE
FROM PRESSURE TRANSMITTER
CYLINDER

The roll system's reference is an "Amount Gyro," which is a free-free gyro oriented on the ground prior to missile launch. The brush of a four-tap potentiometer (Item 2 in Fig. 1) is connected to the outer gimbal and provides a dc signal whose sign and magnitude indicate the roll position with respect to the stable equilibrium point. This signal is the principal input to the servo amplifier that drives the valve.

A roll-position error exists in the situation illustrated in Fig. 1. The valve is driven in the direction to cause the oil flow to rotate the ailerons, which in turn will roll the missile toward the null position. As the missile rolls, the winding of the roll-amount potentiometer rotates with it. The brush stays fixed in space with the gyro gimbal.

The aerodynamic coupling between the aileron position and the missile's roll position is a complex and variable term in the feedback loop of the servo. The nature of the aerodynamic coupling is such that an otherwise simple servo problem becomes considerably more complicated. During a normal flight the aerodynamic stiffness, and hence the gain in the feedback loop, varies over a 50:1 range. A first order correction for this change is accomplished by a variable gain local loop around the valve, cylinder and amplifier. A potentiometer (Item 3 in Fig. 1) is geared to the fin in such a way that a dc signal is produced, which is proportional to fin position. The gain of this local loop is varied by supplying the potentiometer with voltages that are directly proportional to the measured aerodynamic stiffness. In this way the amount of the deflection of the aileron is made inversely proportional to the aerodynamic stiffness. This effect results in an approximately constant torque about the roll axis of the missile for a given signal. The local loop around the fin position also reduces the effect of any non-linear characteristics of the valve.

A third input to the servo amplifier is provided by a potentiometer that is driven by a spring-restrained gyroscope mounted so that the sensitive axis is aligned with the missile's roll axis. The dc signal produced is proportional to the roll rate. This signal provides some anticipation to the roll position loop. It performs the function of a tachometer in a conventional servo. It also insures that the roll rate is limited to a value that can be handled by the position loop. If very high roll rates were allowed to exist, the roll amount gyro would produce a signal changing in sign at such a rate that the ailerons would be unable to keep up or reduce the rate.

The roll servo insures that the missile's orientation is aligned with the free-free gyro. This enables the yaw and pitch servos to steer about their assigned axes in a consistent manner. The two steering servos are

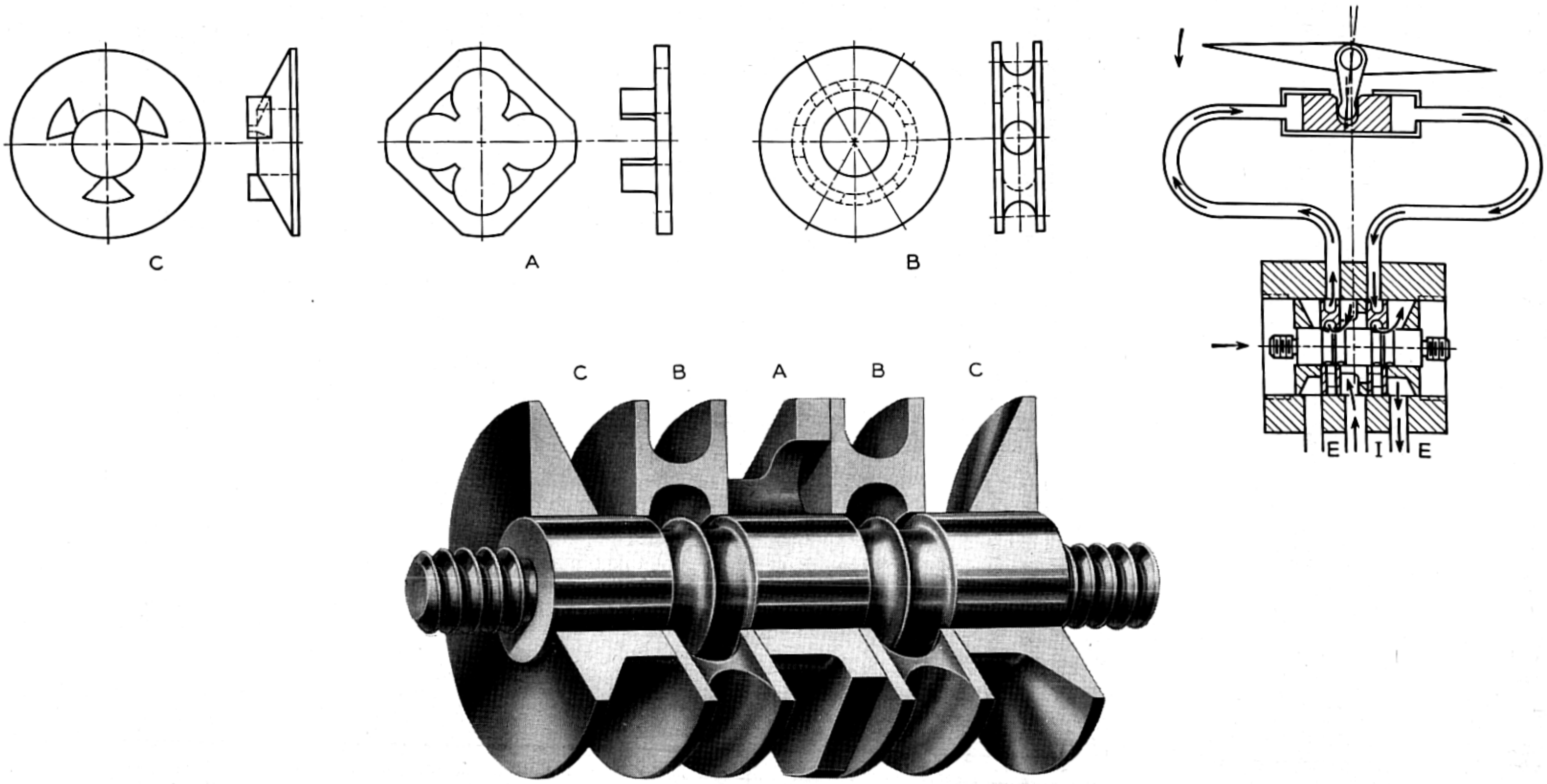


Fig. 2 — Porting arrangement, J-7 solenoid valve.

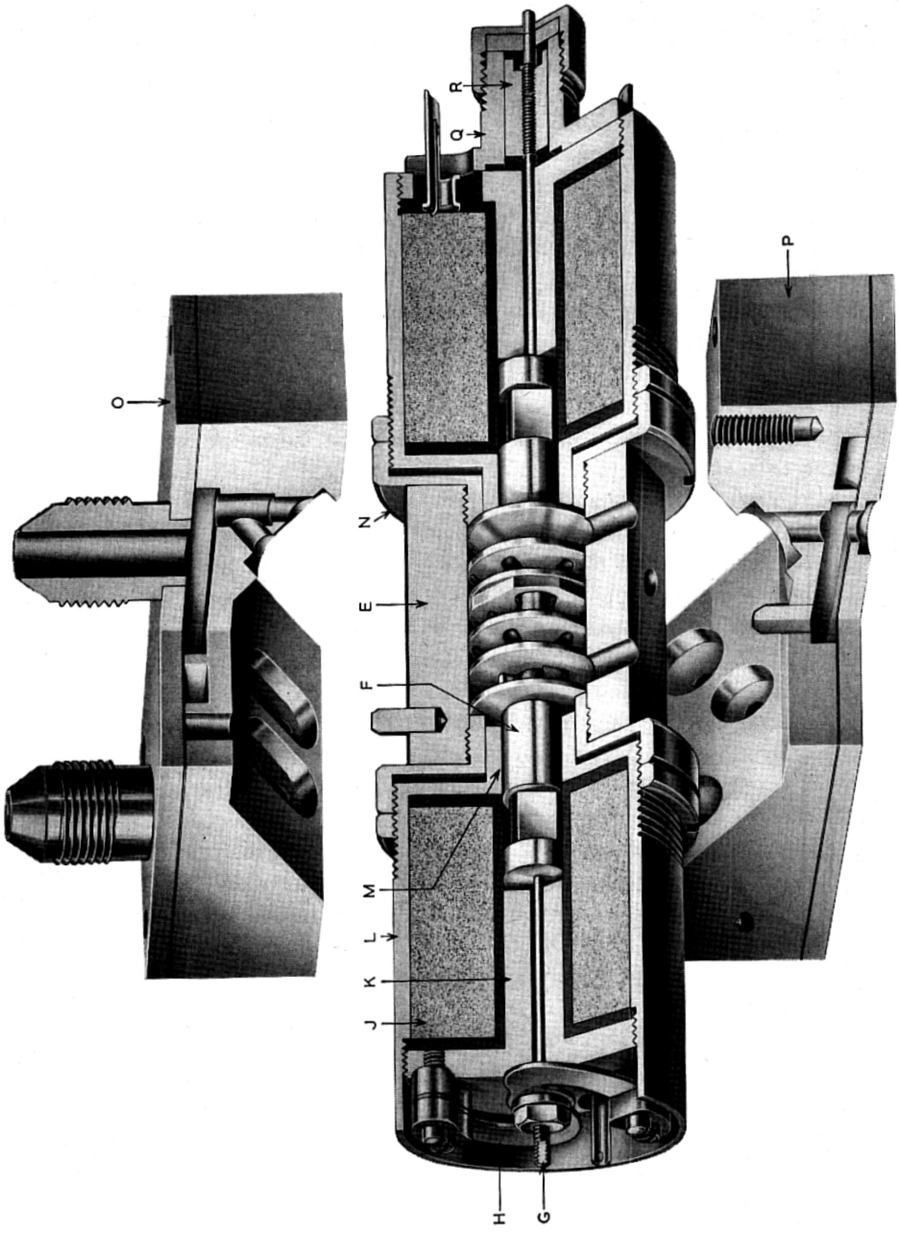
identical to each other and somewhat similar to the roll system described above. Each of the three systems employs identical valves.

GENERAL DESCRIPTION

Basically, the J-7 valve is a conventional four-way type. Fig. 2 illustrates the porting arrangement. The parts in sections A, B and C are inserts that are shrunk-fit into the valve body. The plunger accurately fits the holes in the inserts, so that oil cannot flow between the plunger and inserts except where the diameter of the plunger is reduced. The annular space around the outside of the center insert is connected to the high pressure oil supply. The radial passages in this insert (part A) carry the oil to its internal cusps. With the plunger centrally located its center land completely covers the port formed by the cusps and no oil flows. With the plunger moved to the right the oil is carried to the cylinder and back to the exhaust in the manner illustrated by the small sketch in the upper right corner. If the motion of the plunger is to the left, a similar performance occurs but the piston and fin are driven in the opposite direction. To illustrate the construction of the inserts, detail sketches are also shown at the top of Fig. 2.

The inserts and the plunger are made of hardened steel. Fig. 3 show- their location in the complete valve. The thickness of the inserts, hence the longitudinal location of the ports, is held to an extremely close tolerance by lapping their parallel faces. Their outside diameter is accurately ground so that a tight seal will occur between the various passages when they are shrunk fit into the internal bore of the body. After assembly, the internal bore formed by the holes in the various inserts is lapped to a straight and accurate cylindrical shape. This process is controlled to provide a diametral clearance of 0.0002 inch on an interchangeable basis. The plunger must slide freely in the bore in spite of the small clearances involved. The longitudinal location of the lands on the plunger must be controlled to a high degree for reasons that will become apparent.

Those parts shown in Fig. 2 are not sectioned in Fig. 3. The valve proper is clamped between two manifolds (O and P); these are moved apart in the picture to better illustrate the internal construction. The brazed laminated manifolds provide the mounting means for the valve and also serve to connect the multiple outlets of the valve body to standard hydraulic fittings for external connections. The manifolds are designed to adapt the valve to a specific application. In this way, different plumbing arrangements can be utilized without changes in the valve proper. The manifold, O, has fittings to connect to the cylinder,


R
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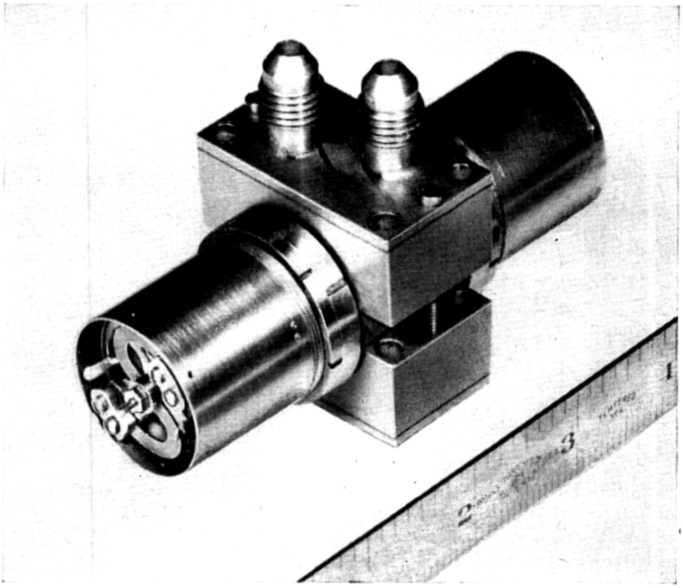

Fig. 4 — J-7 solenoid valve with manifolds.

and the lower manifold, P, serves to connect the pressure and exhaust lines to ports in the structure to which the valve is mounted. The joints between the passages in the manifolds and those in the body are sealed by rubber "O" rings that are inserted in the recesses about the holes on the inner faces of the manifolds. These recesses can be seen in Fig. 3, but the "O" rings are not illustrated. Similarly, "O" rings are used to seal the joints between the manifold, P, and the flat surface to which it mounts.

A pole piece, *F*, is screwed to each end of the plunger by means of the threads visible in Fig. 2. These parts move as an assembly and form

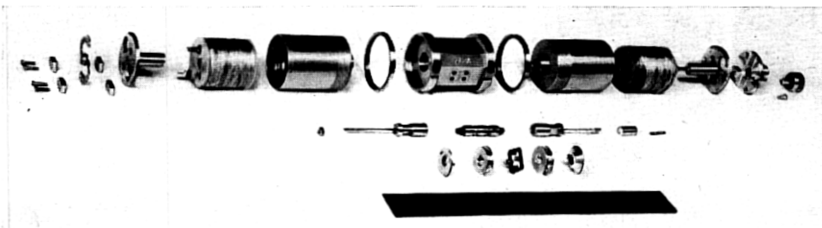

Fig. 5 — Exploded view of J-7 solenoid valve.

Fig. 6 — J-7 solenoid valve.

the armature of the valve. The push rod, G, attached to the armature, is connected to an S shaped spring, H, which tends to keep the movable assembly centered, or the valve closed.

When a current passes through the coil, J, magnetic flux passes through the fixed pole piece, K, through the coil housing, L, then across the small annular air gap, M, to the moving pole piece, F, and across the air gap between the pole faces near the center of the coil. Flux in the latter gap causes a force on the armature which tends to pull it and the valve

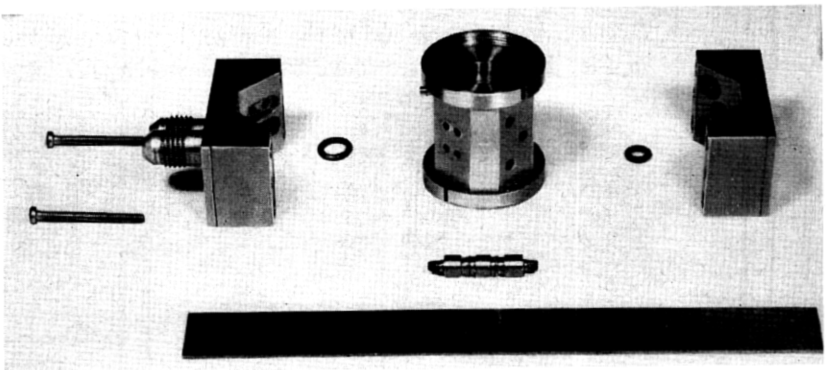

Fig. 7 — Hydraulic parts of the J-7 valve.

plunger toward that coil. If an equal current is flowing in the other coil the forces are balanced and the valve remains centered and closed. If the currents in the two coils are not equal, the valve plunger is moved until the differential magnetic force is balanced by the force in the spring. In this manner the amount of oil flow can be regulated by varying the difference of the currents in the two coils.

The coil housing is attached to the valve body by means of a non-magnetic stainless steel adapter, N. The adapter isolates the steel plunger and inserts from the magnetic flux. Because of the close fit between these parts the presence of flux would cause sticking. A push rod attached to the armature drives an aluminum piston, R, in a cylinder, Q. The small radial clearance between these parts is filled with a viscous fluid so that a damping force is produced that is proportional to armature velocity.

Fig. 4 shows the complete assembly with manifolds attached, while Fig. 5 gives an exploded view of the valve proper with all details in their correct relative positions. Other views of the valve parts are shown in Figs. 6 and 7. Fig. 8 is a view looking into the bore of the hydraulic assembly. This is the hole that is normally occupied by the plunger. The ports formed by the internal shapes of the inserts are clearly visible.

CHARACTERISTICS OF THE ACTUATING MECHANISM

The J-7 valve is designed to be driven by a push-pull dc amplifier. When the amplifier has no input signal, the output current in each side is 10 ma. When a signal is applied, the current in one side is increased and that in the other is decreased; at maximum signal, the current in one coil reaches 19 ma and zero in the other. The dissipation in each coil for quiescent current is about 0.4 watt, and the full signal power is 1.5 watts.

The requirements that the frequency response must extend to dc and that the control power consumption be held to a minimum suggest the use of a dc push-pull output stage. The quiescent dc plate current is used as the magnetizing current for the solenoids instead of providing the field by a permanent magnet or separate coil. In spite of the small output current available from the amplifier, relatively large forces and a high resonant frequency are realized. This is accomplished by an efficient magnetic circuit and low armature mass. The opposing solenoid configuration described makes these features possible.

The magnetic circuit used in the J-7 valve has very low reluctance for a sliding armature type actuator. This low reluctance is accomplished by providing ample thickness in the iron parts and employing

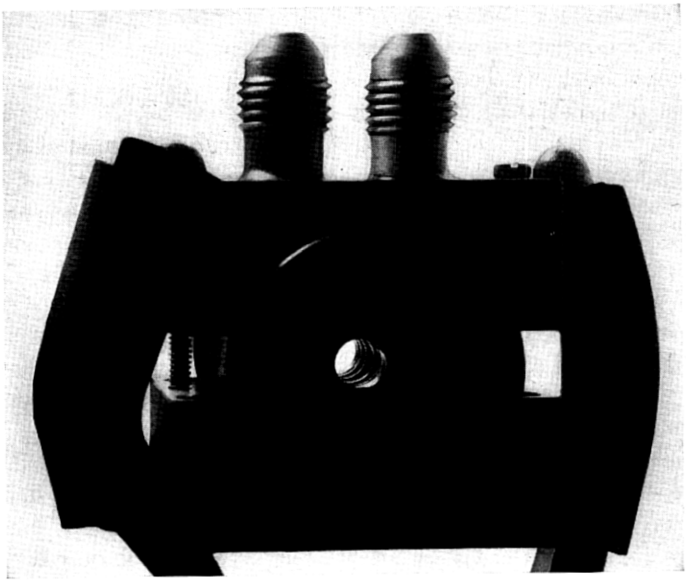

Fig. 8 — Internal view of the J-7 valve body.

very short gaps of considerable cross-sectional areas. It has been shown elsewhere[1,2] that if the distribution of reluctance of the magnetic structure is symmetrical along its longitudinal axis, minimum leakage flux and hence maximum force would be developed if the working gap were at the exact center of the coil. To a close approximation, the valve solenoids are symmetrical in this manner. The references show that the maximum pull is not sensitive to small changes from the optimum location of the working gap. The gap in the J-7 valve is displaced toward the center of the valve from the location of maximum pull. The slight reduction in magnetic force was accepted as a suitable compromise for the resulting reduction in the mass of the moving pole piece and the corresponding increase in resonant frequency.

The gap between the solenoid pole faces is 0.014 inch when there is no signal and the valve is centered. The two fixed faces have 0.004 inch non-magnetic shims attached so that the maximum motion of the armature is limited to ± 0.010 inch. The shims prevent the armature from sticking against the fixed faces by maintaining appreciable gaps. To further compensate for the inverse square law of magnetic attraction, the moving pole pieces have a reduced section that saturates under high flux or large forces. This neck can be seen plainly in Fig. 3. The saturating sections limit the flux and tend to reduce the pull for short gaps so that the centering spring can be a simple linear member and still not lose control when the armature is near the fixed pole face. The flat surfaces on the neck permit the use of wrenches for assembly. Placing the necks on the moving pole pieces further reduces the mass of the armature assembly.

To illustrate the saturation action, Fig. 9 shows the pull of one of the magnets plotted against gap length for quiescent and maximum current. The shim line and curves from a solenoid without a saturation neck also

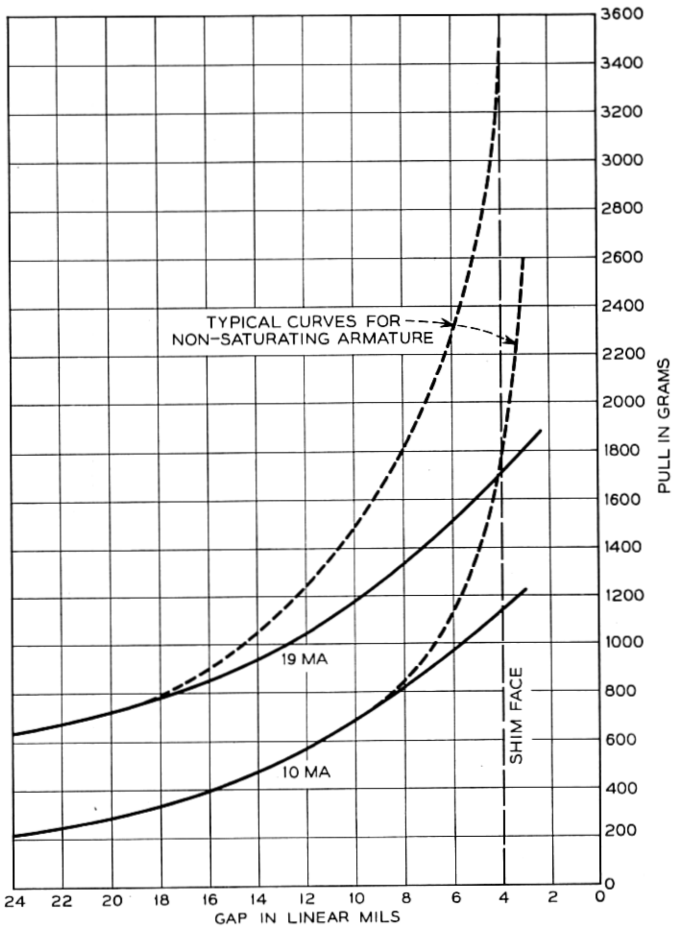

Fig. 9 — Magnetic pull curves for a coil of the J-7 valve.

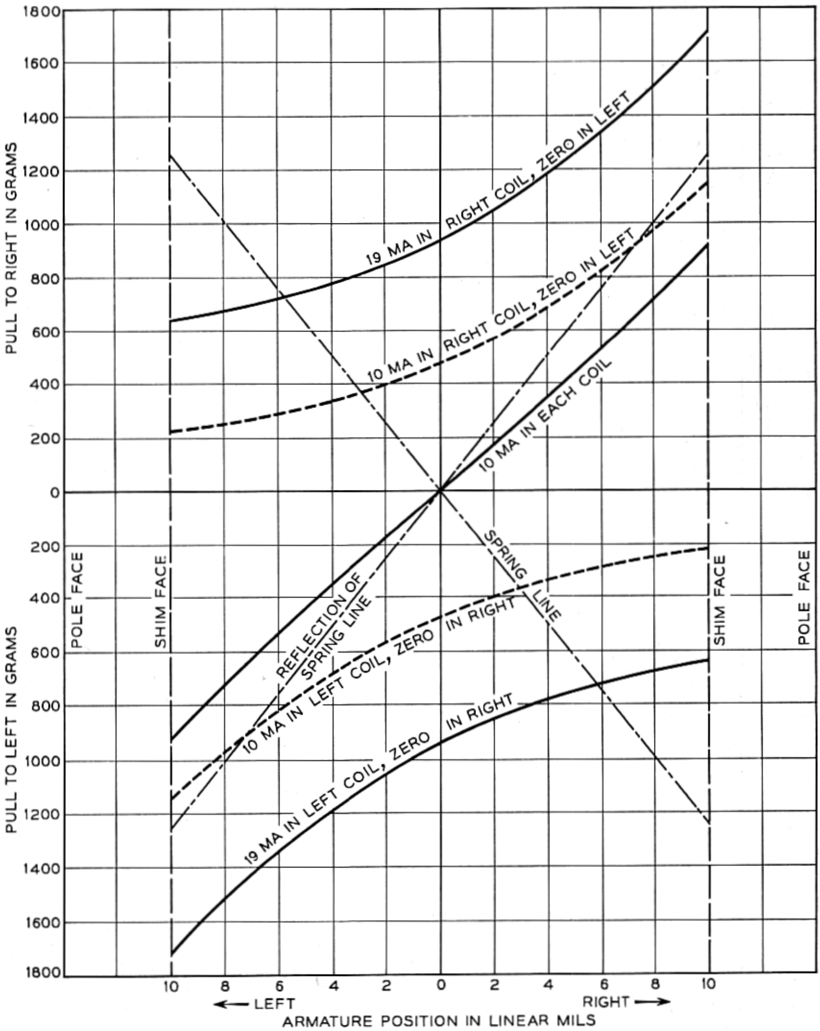

Fig. 10 — Spring and magnetic forces on armature of J-7 valve.

are shown to illustrate the need for these restrictive measures. The need is better appreciated when Fig. 10 is examined. This graph shows the differential pull of the two coils plotted against armature movement for extreme signals and for the balanced condition. There is also a line representing the spring force and one representing its reflection. The latter permits direct comparison of the magnitude of the opposing magnetic and spring forces.

It is important that the spring be able to center the valve when the coil currents are balanced. This means that the stiffness of the spring must be greater in magnitude than the negative stiffness created by the magnetic fields when 10 ma is flowing in each coil. On the other hand, the 19-ma current should be able to pull the armature against the pole piece and therefore must produce more force than the spring. If the shim and saturation limiting were not used it would be impossible to find a straight spring line that would fulfill both these requirements; i.e., its reflection would be between these curves without crossing either of them. The family of curves representing the net magnetic forces for the various intermediate values of current unbalance fall between the extreme cases shown. The intersection of one of these curves and the reflection of the spring line is the position which the armature will assume for that particular coil current. The reason for the relatively large margin of force shown for the 19-ma wide-open condition will be explained later.

Fig. 11 is plotted to show the net forces on the armature. The curves are the difference between the spring line and the two magnetic pull curves of Fig. 10. It can be seen that the forces are such as to cause the armature to move to the center in the balanced condition. In the case of maximum signal, it will move all the way to the shim stop in the direction of the coil which is carrying the current.

When there is no magnetic field present the armature resonance is about 320 cps. When measured statically, or at very low frequencies, with the coils energized, the negative stiffness of the field greatly reduces the effective stiffness of the spring, as seen in Fig. 11. However, when the valve is driven experimentally to find resonance, it occurs near 320 cps. This apparent increase in stiffness with frequency results from eddy currents that retard the change in flux to the extent that the negative stiffness virtually disappears. Eddy currents reduce the effective inductance of the coils from about 40 henries at very low frequency to less than 10 henries at 600 cps.

It is difficult to locate the resonance experimentally because of the large amount of damping provided by the extremely thin oil film between the plunger and the inserts. High resonance frequency of the valve is desired so that it is safely above any frequency encountered in the servo operation, thereby eliminating one consideration in the equalization. Also, a high resonance means that missile acceleration along the valve axis causes little displacement of the unbalanced mass of the armature.

A 250 cps differential dither voltage is superimposed on the push-pull dc signal to overcome the effects of static friction. The resulting 1 ma differential current produces a magnetic force about equal to that re-

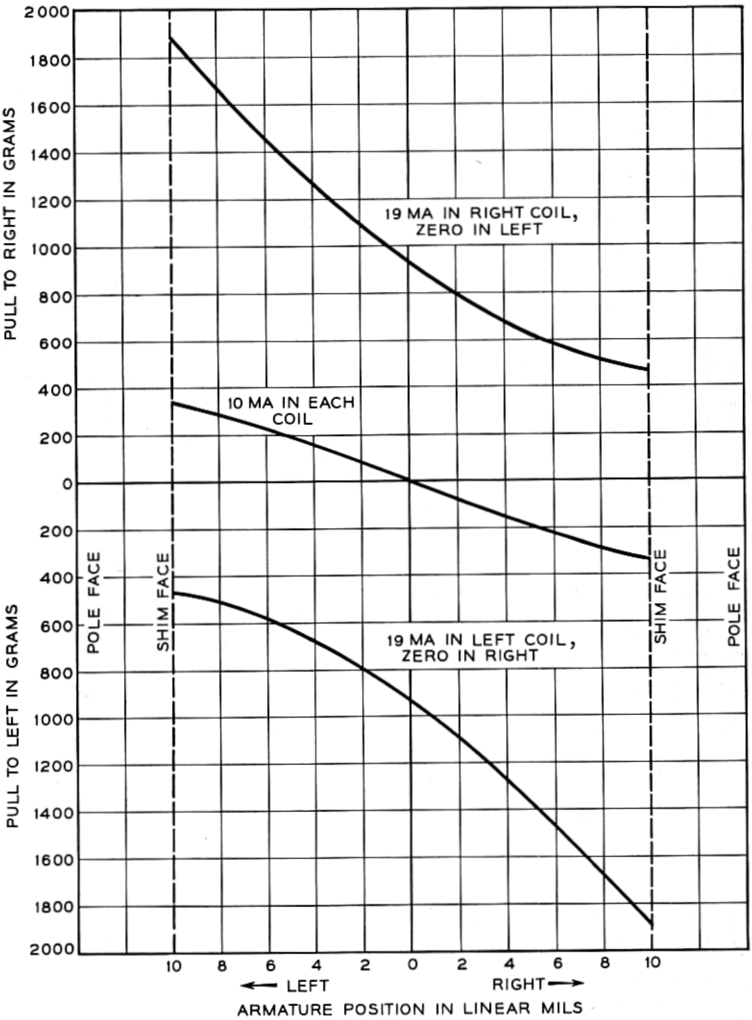

Fig. 11 — Resultant forces on armature of J-7 valve.

quired to move the armature in breaking the friction of the stationary armature assembly. Thus, the signal threshold is reduced by the amount of the dither current and the resulting increase in sensitivity to small signals greatly reduces the phase lags at low amplitude.

STEADY STATE HYDRAULIC CHARACTERISTICS

The J-7 valve is designed to operate from an oil supply having a pressure of 2,000 psi, which is somewhat higher than early valves of this type. The increase in working pressure is a great advantage for a guided missile application because it results in lower weight, higher gain, and faster response. For example, doubling the pressure permits actuating cylinders of one half the size, an oil reservoir of one half the volume, and an increase in both response and gain by a factor of about three. Such features are sufficiently attractive to be worth a great deal of development effort. However, reliable and stable operation can be achieved under these high-gain conditions only if parasitic forces are kept extremely small.

It was found that the relation between pressure drop and flow is not so simple as one might expect from a sharp-edged, orifice-type control. For large openings of the control orifice, the pressure losses in the fixed orifices and passages of the valve body become an important factor. The following law is an adequate representation of pressure-flow characteristics:

$$p = 10q + \left(2 + \frac{455}{x^2}\right) q^2 \tag{1}$$

where

p = pressure drop, psi

q = rate of flow, cu in/sec

x = valve opening, linear mils

This equation was derived from test data from a model valve. These data confirmed a computational analysis of the hydraulic circuit. Fig. 12 graphically illustrates the equation. It is a plot of flow against valve position for various pressure drops. It will be noted that there is 0.001 inch difference between valve position and valve opening because of this amount of overlap at the ports.

Equation (1) provides the information necessary to compute the maximum output power of the valve. If all the pressure drop were across the control orifices, all the pressure would be utilized to accelerate the oil at this point and only the square term of (1) would exist. If this were the case, the maximum output power would occur when the pressure drop across the valve was one-third of the supply pressure. The other two-thirds of the pressure would be used to produce work in the cylinder. If laminar flow existed throughout the valve the square term would drop out, leaving a linear equation. If this were the case, maximum power would occur with the total pressure equally divided between valve and load. When both the linear the square terms are present, maximum power will occur when the pressure drop across the valve is somewhere

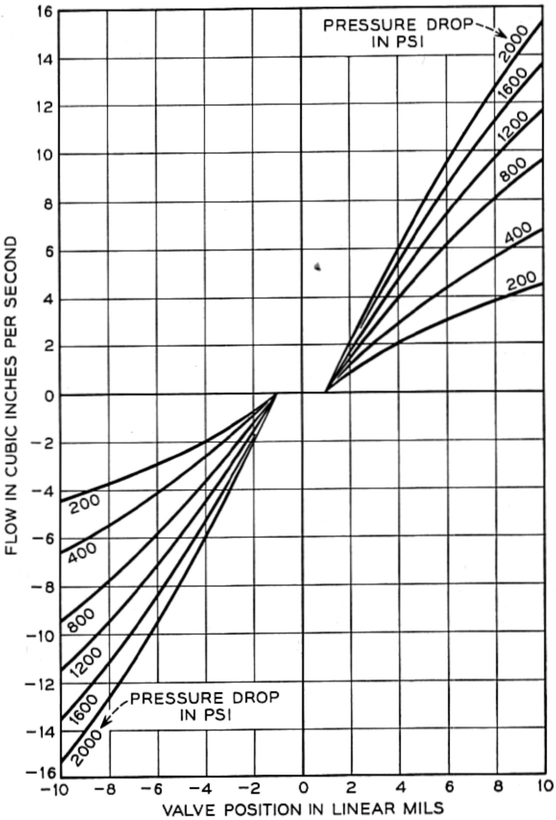

Fig. 12 — Flow characteristics of J-7 valve.

between one-third and one-half the supply pressure. The exact point is dependent on the magnitude of the supply pressure as well as the coefficients in the equation.

The valve will be wide open, an opening of 0.009 inch, when maximum power is produced. In this situation, (1) becomes:

$$p = 10q + 7.6q^2 \tag{2}$$

Useful output power at the load is the product of the flow rate and the pressure exerted on the piston.

$$\dot{W} = (P - p)q \tag{3}$$

where

$$\dot{W} = \text{Output power}$$

$$P = \text{Supply pressure, psi}$$

Therefore

$$\dot{W} = Pq - 10q^2 - 7.6q^3$$

This expression can be differentiated and equated to zero to find the point of maximum power.

$$\frac{d\dot{W}}{dq} = P - 20q - 22.8q^2 = 0 \tag{4}$$

$$q = \sqrt{0.192 + 0.0439P} - 0.439$$

Substituting for q in equation (2) we find that for maximum power

$$p = 0.333\,P + \sqrt{2.15 + 0.49P} - 1.46 \tag{5}$$

The normal supply pressure for the J-7 valve is 2,000 psi. Substituting this value for P in (4) and (5)

$$q = 8.95 \text{ cu in/sec}$$

and

$$p = 696 \text{ psi}$$

When these values are substituted in (3), we find

$$\dot{W} \max = 11{,}700 \text{ in lb/sec}$$
$$= 1.77 \text{ horsepower}$$

Examination of the above equations will show that if the valve is used with a very low supply pressure, the linear term in (1) is dominant. In this case the maximum power output occurs when the pressure drop is nearly one-half the supply pressure. In the case of a very high supply pressure, the squared term is dominant and maximum power occurs when the pressure drop across the valve approaches one-third the supply pressure.

The ratio between the electrical quiescent input power to the coils and the maximum hydraulic out power is about 1,600, or a power gain of 32 db. Based on maximum signal the gain is 29 db. All forces must be precisely balanced and tolerances on parts be carefully controlled in order to realize this amount of gain in a single stage mechanical device.

DYNAMIC HYDRAULIC EFFECTS

Examination of the illustrations of the valve will show that it is statically balanced; i.e., pressure on any of the ports does not tend to translate the plunger. However, the flow of oil through a valve of this type produces a force on the plunger which tends to close the ports or center the armature. This force creates a stiffness that adds to the effect of the mechanical centering spring. The magnitude of the force is quite nonlinear, it varies with pressure drop, and hence, with load as well as with armature displacement. Such variations tend to upset the stability of the servo loop in which the valve is used.

Fig. 13 is a simplified drawing of a valve which can be used to explain the dynamic effect. It will be noted that when the valve plunger is displaced in either direction, the fluid flow is metered by two control orifices in series. The oil flows from oil supply, through the valve body, into a groove in the plunger via the first of the two orifices. The second

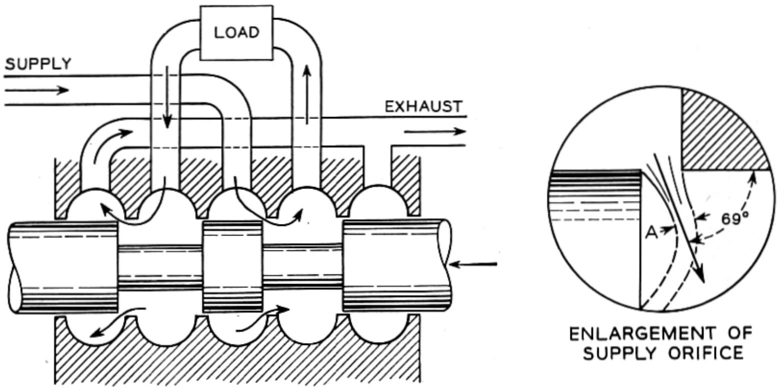

Fig. 13 — Simple valve with rectangular lands.

orifice meters the flow from the other groove in the plunger into the exhaust part of the valve body. Most of the pressure drop in the valve appears across the two orifices and is equally divided between them. The maximum fluid velocity occurs immediately downstream from the orifices at the vena contracta of the jet. This point is labeled "A" in the enlarged insert on Fig. 13. The velocity at this point can be computed by use of Bernoulli's Theorem. In the case of the J-7 valve, where the valve opening is small compared to other passage dimensions, many of the terms of the equation that formulates this theorem can be neglected. In this way the equation becomes

$$h = \frac{V^2}{2g} \tag{6}$$

$$V = \sqrt{2gh} \tag{7}$$

where

V = fluid velocity at the vena contracta, ft/sec

h = pressure drop across orifice, feet

g = acceleration of gravity, ft/sec^2

Von Mises[3] has shown that the departure angle of the jet from a small orifice, such as in the configuration shown in Fig. 13, is 69° from the longitudinal axis. Tests made with an orifice, shaped like those in a simple valve, showed that the jet continues at this angle for a short distance only. Further downstream the jet turns to hug the radial surface on the plunger. This action is depicted by the dotted lines in the insert on Fig. 13. Bernoulli's equation explains that pressure is exchanged for velocity. The low pressure within the jet stream pulls it toward the nearest wall of the cavity. The flow of oil over this surface reduces the pressure on that wall and unbalances the distribution of forces on the surface of the annular grooves in the plunger. The area of reduced pressure causes a net longitudinal force in the direction to center the plunger or close the valve.

Examination of the situation around the exhaust orifice shows that a similar action will occur, but will not result in a comparable force on the plunger. The area of high velocity lies along a surface in the valve body rather than acting on the plunger. The velocity upstream from the orifice is not localized and hence produces forces that are small with respect to those downstream. The fact that the exhaust port forces on the plunger are small compared to those of the intake was confirmed by tests.*

There is a small time lag between the opening of the ports and the dynamic centering force which is proportional to the rate of change of oil flow. This lag results from the fact that finite time is required to change the velocity of the oil mass in the system. At high frequencies, the delay results in a considerable phase lag between the plunger position and the dynamic force. This delay means that fluid velocity, and hence the force, is higher during the quarter cycle in which the valve is closing than it is during the quarter cycle in which the valve is opening.

* Considerable work has been conducted on valve theory and design elsewhere since the J-7 valve was developed. Reference 4 is an excellent example of a thorough analysis of valve dynamics with an approach to the problem from a different viewpoint. This reference reports on tests and theories which show the secondary forces from the exhaust and intake orifices to be equal. This is in direct contradiction to the experience with the J-7 valve and remains as an unresolved problem in the mind of the writer.

Therefore, more energy is exerted on accelerating the mass of the plunger during the closing operation than is absorbed in slowing the mass during the opening phase. If the net gain in momentum is larger than can be absorbed by the viscous damping of the oil film in the lapped fit, oscillation[5] will occur. In the case of the J-7 this oscillation tended to occur at slightly over 400 cps.

The Bernoulli force described above was recognized and measured early in the development of this series of valves, but was considered unimportant due to the high stiffness and large damping inherent in the design. The experimental models and the early production models showed no indication of oscillation. Later in the production program, the lapped clearances were increased and high ambient temperatures at the missile test locations were encountered. These two factors combined to reduce the damping, due to the working fluid, to the point where hydraulic oscillation occurred. An external damper was added to alleviate this problem. The damper consisted of an aluminum piston closely fitted to an aluminum cylinder. A viscous fluid (polyisobutylene) between these two parts provided sufficient damping to stabilize operation. This fluid also has the advantage of a relatively small decrease in viscosity with temperature. This type of damper is illustrated on the valve in Fig. 3. (Subsequent improvement in the internal design of the valve reduced the dynamic effect to the point where the need for the external damper has been eliminated.)

Fig. 14 shows a compensated intake orifice configuration corresponding to the insert picture on Fig. 13. It represents the first attempt to balance the dynamic or Bernoulli force. The depth of the annular groove

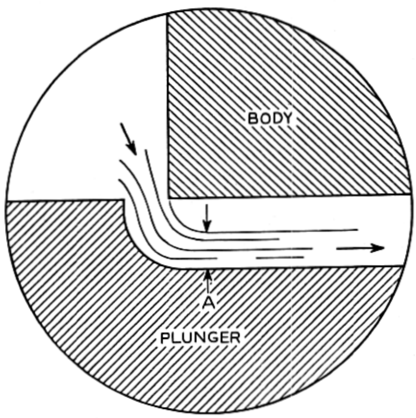

Fig. 14 — A compensated valve orifice.

in the plunger is reduced and a curved surface added to direct the flow parallel to the valve axis. This configuration reduces the dynamic force for two reasons. First, the amount of radial surface exposed to the low pressure is greatly reduced. Second, the curved surface acts much like a turbine blade in deflecting the oil stream and developing a reaction thrust that opposes the Bernoulli force. The reaction thrust increases as the longitudinal component of the high-velocity jet is increased. If the jet can be turned to become parallel with the longitudinal axis without appreciable loss in velocity, the maximum reaction thrust is obtained. In this case, the force is equal to the increase in the longitudinal component of momentum over the conditions of the free jet as shown in Fig. 13.

$$F = \frac{\rho q V}{g}(1 - \cos 69°) \tag{8}$$

where

F = force, lb

ρ = fluid density, lb/cu in

q = flow rate, cu in/sec

V = fluid velocity, ft/sec

g = acceleration of gravity, ft/sec^2

Calculations of the dynamic forces in accordance with the above reasoning yield only approximate results because the local velocities and their gradients are functions of passage shape as well as pressure drops. The contour of the plunger grooves were computed for use in the first experimental model, whose design was intended to alleviate this problem. Refinements to the initial model were made by cut-and-try methods. Since the forces involved are relatively small, and their magnitude changes so rapidly with plunger position, specialized measuring instruments had to be developed whose sensitivity was high and compliance very low.

A certain amount of contradiction was apparent in the force measurements made. To better understand the action of the oil within the valve, a transparent replica of the cross section of a valve port was used under a microscope. Figs. 15, 16, and 17 are illustrations of typical tests. Fig. 15 is two views of an early type valve with rectangular ports at different openings and pressure drops. The arrows indicate a portion of the cylindrical sliding surface separating the plunger and body. The lower left shadow is the sharp corner at the edge of the annulus in the plunger.

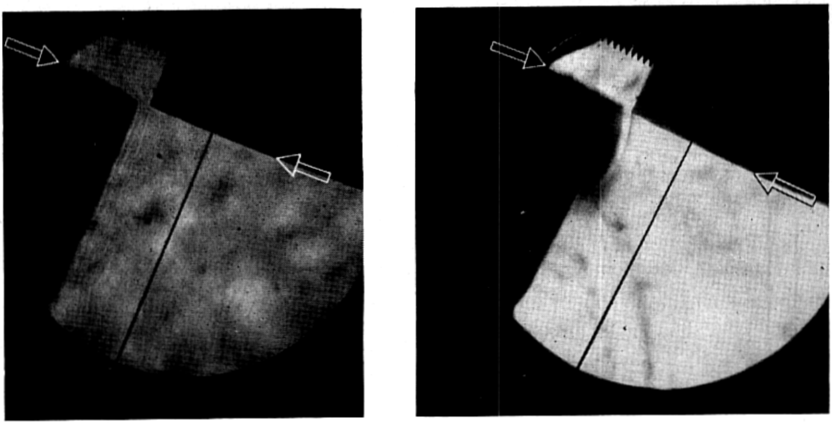

Fig. 15 — Oil flow through simple port.

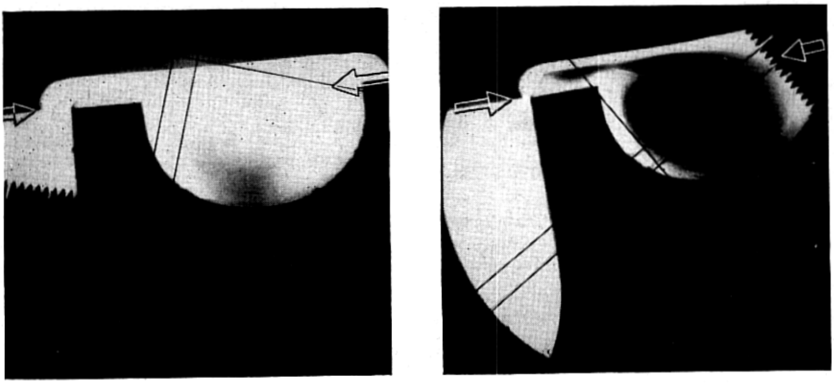

Fig. 16 — Flow through port with decreased Bernoulli effect.

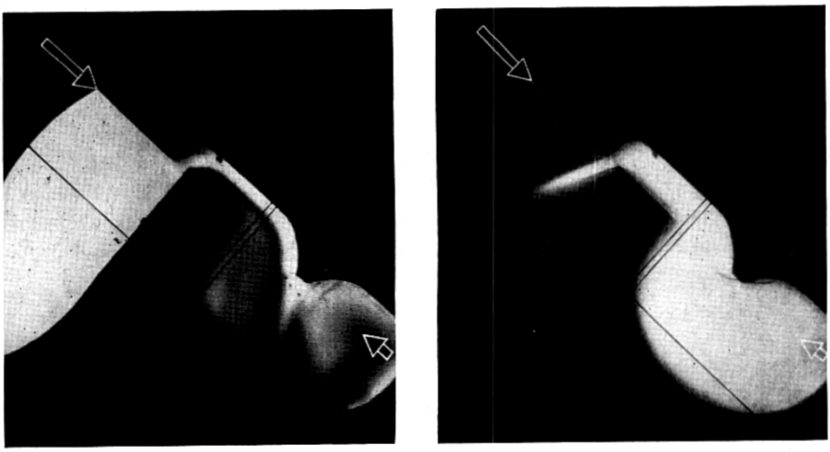

Fig. 17 — Flow through port with decreased Bernoulli effect and increased flow.

The lower light area is oil in the annular groove of the plunger. The oil is flowing from top to bottom. It will be noted that the flow on the downstream side of the orifice hugs the radial surface of the plunger, as discussed above and depicted in Fig. 13. The crosshair and the comb-like scale are a part of the microscope.

Fig. 16 shows two views of a subsequent trial model quite similar to that shown in Fig. 14. Here, the bottom shadow is the insert and the top is the plunger. The flow is from bottom left to top right. This particular design reduced the Bernoulli force but resulted in a serious reduction in flow. This reduction was caused by the large cavitation bubble visible in the right view. This bubble was the result of rapid rotation of oil in the chamber. It prevented the orderly release of the oil through the internal passage to the actuating cylinder (not visible in pictures).

Fig. 17 shows a trial model similar to the J-7. The left illustration

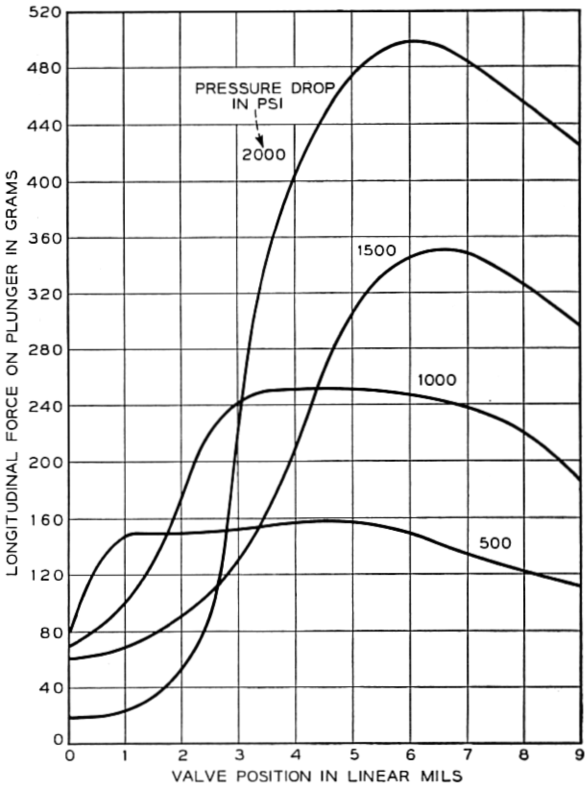

Fig. 18 — Force on plunger of J-7 valve due to Bernoulli effect.

shows the oil flowing from left to right and simulates the intake port of the valve. The extra land on the plunger directs the oil stream toward the escape passage to reduce turbulence and cavitation and increase oil flow. The right illustration illustrates the reverse flow of oil, right to left, and represents the exhaust port of the valve.

Fig. 18 is a plot of the measured Bernoulli force on the J-7 valve. It will be noted that there is little relation between the curves for various pressure drops. No simple equation has been formulated to account for the forces observed. Although Fig. 18 shows the Bernoulli force to be large, Fig. 11 shows that full signal produces enough net force on the armature to overcome this force at any valve position.

A large part of the development effort on the J-7 model was expended on the problem of reducing the forces caused by oil flow. For the same flow conditions, the J-7 has about one-fifth the Bernoulli force of earlier designs with simple rectangular grooves in the plunger.

Subsequent to the initial manufacture of the J-7 valve, the design of the annular grooves and body inserts has been improved continually. Consequently, the Bernoulli force has been further reduced to permit higher operating pressure, and hence more gain, without creating a hydraulic oscillation problem.

THE J-7 VALVE AS A SERVO ELEMENT

For any given set of operating conditions, the transfer function (expressed as cubic inches of oil flow per milliampere of control current unbalance per lb per square inch of pressure drop) can be extracted from the information presented above. However, the resulting family of curves for various load torques would be of little use to the servo designer. The pressure drop available for use by the valve is different for each curve, and they are all quite nonlinear, as is apparent from the data.

The overlap of the valve ports results in another type of nonlinearity that complicates the loop equalization problem. Examination of Fig. 12 will show that the effect of the overlap is a small dead area in the region of zero output of the valve. Small signal levels will cause the valve armature to operate in and around the vicinity of the dead zone, resulting in very little oil flow. Thus, the gain of the valve for very small signals is lower than for signals of greater magnitude.

As mentioned earlier, the effect of the nonlinearities of the valve is greatly reduced by use of the relatively fast-acting local loop which encompasses the valve, actuating cylinder, and amplifier. This inner, or secondary, loop contains sufficient gain to insure that, in spite of the nonlinearities, the fin position is controlled in strict accordance with the summation of the input signals applied to the amplifier.

The first design of the servo circuits was made by using the slopes and magnitudes of typical and extreme points of operation as obtained from Figs. 11 and 12. The design of the servo equalization networks was obtained by successive refinements made during actual tests of the complete servo systems. These tests were performed with the aid of rather elaborate simulators that subjected the systems to the conditions of actual flight.

The characteristics of the valve and the amplifier which drives it, as applied to a servomechanism, can best be illustrated by plotting gain and phase shift versus frequency in an open loop. Fig. 19 is such a graph. This information was gathered by applying an input signal to the servo amplifier from an oscillator. The valve was driven by the amplifier in the usual manner. The valve controlled the flow of oil to a piston which operated a load that was equivalent to a typical aerodynamic load as seen by the control surface. The voltage from the fin-position potentiometer was compared to the amplifier input. Fig. 19 shows the phase and amplitude comparison of these two voltages. A small amount of feedback was used to prevent the piston from drifting to one end of the cylinder.

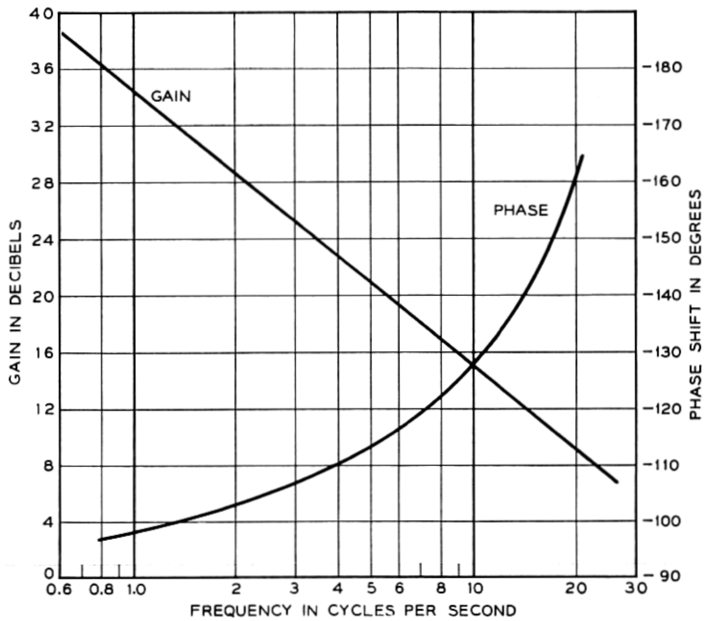

Fig. 19 — Frequency characteristics of J-7 valve.

The data were adjusted to correct for the error introduced in this manner, so that open loop conditions are represented. The equalization networks have compensating leading characteristics to prevent the oscillation suggested by the increasing phase shift at the higher frequencies.

If the servo acted in strict accordance with minimum-phase network theory, the slope of the gain curve would have started to increase at the high frequency end of Fig. 19. The effects of nonlinearities caused by such things as overlap and dither action cause this departure from classical theory. Actually, the slope does start toward 12 db per octave just above the frequency range shown.

Part of the phase shift shown on Fig. 19 is due to the inductance of the coils and the relatively low source impedance of the amplifier used. Later amplifier designs have much higher output impedance, which has greatly reduced this effect.

CONCLUSION

The series of hydraulic valves developed for use in the NIKE missile provide a light-weight, high-performance control element for positioning aerodynamic control surfaces. Although these electrohydraulic transducers provide a high power amplification, they are relatively simple single-stage devices. Their successful application in the NIKE missile has caused hydraulic servos to be considered for many other military control systems, some of which are under active development at this time. The hydraulic servo has many advantages to offer in the high power field that cannot be provided by other conventional types. It is expected that these advantages will foster a great increase in the use of hydraulic servos in high performance applications in the next few years.

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