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VARIABLE-DISPLACEMENT FUEL PUMP DESIGNED FOR ALWT.(U)

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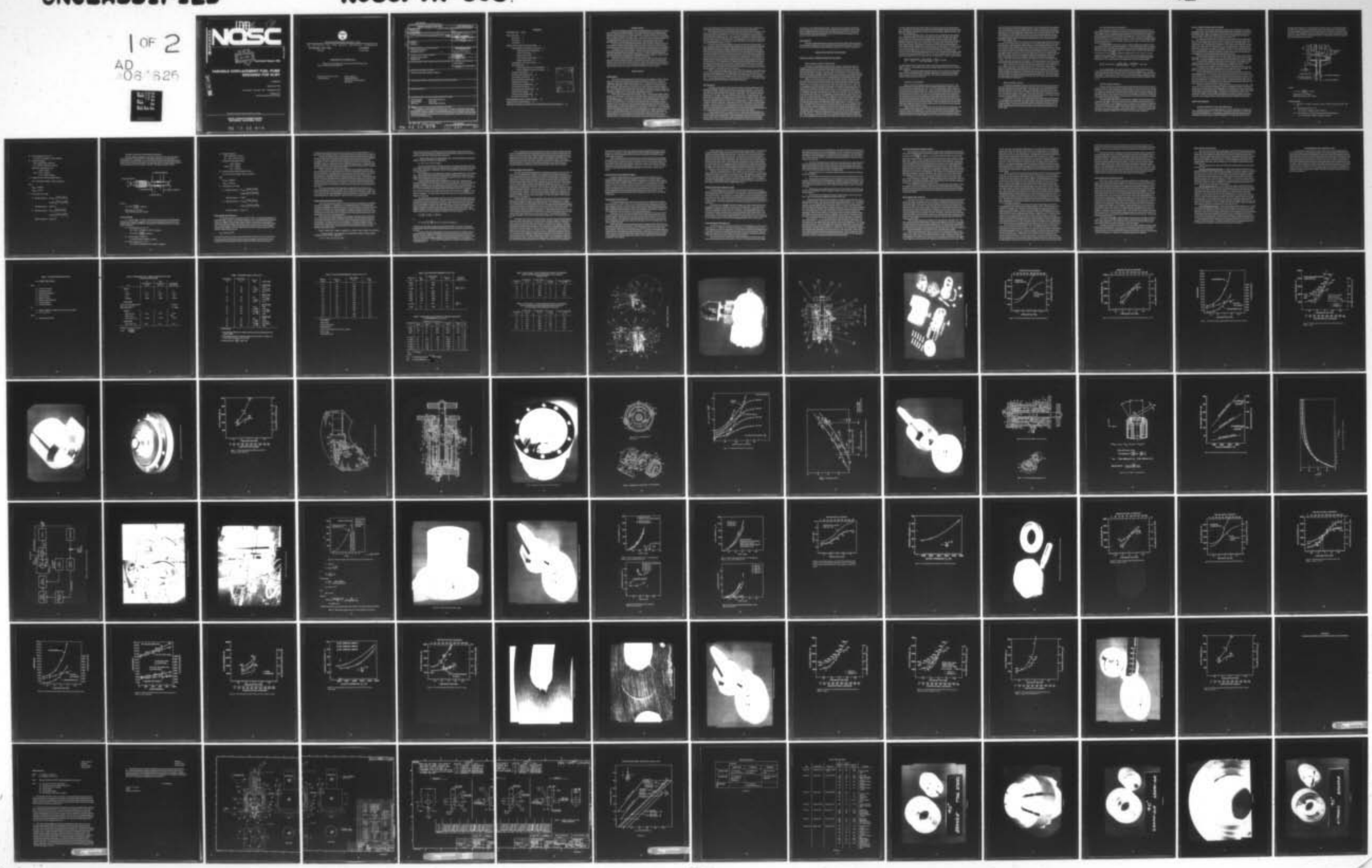

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VARIABLE-DISPLACEMENT FUEL PUMP DESIGNED FOR ALWT

LJ Martini

September 1978

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Prepared for
Naval Sea Systems Command

Approved for public release; distribution unlimited.

NAVAL OCEAN SYSTEMS CENTER
SAN DIEGO, CALIFORNIA 92152

NAVAL OCEAN SYSTEMS CENTER, SAN DIEGO, CA 92152

AN ACTIVITY OF THE NAVAL MATERIAL COMMAND

RR GAVAZZI, CAPT, USN
Commander

HL BLOOD
Technical Director

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Released by RL Matthews, Head
Torpedo Division

Under authority of
MO Heinrich, Head
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Department

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A high-pressure variable-displacement (VD) fuel pump for the Advanced Lightweight Torpedo (ALWT) has been designed and tested. It replaces the fixed-displacement pump, which was determined (from NEARTIP program data) to be noisy and inefficient. Final tests of the VD pump show that this design is capable of meeting the ALWT flow requirement. Recommendations are made regarding further development to ensure longer pump life and consistent performance.

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CONTENTS

INTRODUCTION

The idea of developing a high-pressure variable-displacement fuel pump for a torpedo occurred during the NEARTIP program. It was determined that the major high-frequency self-noise contributor was the fuel bypass mechanism of the fixed-displacement fuel pump. The fuel bypass mechanism is not only noisy, it is very inefficient, since more fuel is pumped than is actually required by the torpedo engine. The obvious solution to these problems was a variable-displacement fuel pump that did not require fuel bypass and that could be packaged within a confined space and controlled by a feedback speed-control circuit.

In the late days of the NEARTIP development stage, Gould Inc. designed and tested a hydraulically controlled variable-displacement fuel pump. The hydraulic control system had excessive hysteresis, and the pump could not develop the maximum pressure requirement.

In February 1976 NOSC designed a variable-displacement fuel pump with a positive-position electromechanical control system. The pump was specifically designed to meet the Advanced Lightweight Torpedo (ALWT) requirement: 3.2 gpm at 4800 psi at 4200 rpm, and intermediate flows and pressures at varying speeds down to 0.8 gpm at 1200 psi at 1500 rpm. The fuel pump was designated the "ALWT VD Fuel Pump."

This report presents the ALWT VD Fuel Pump effort under two main section headings; Final Status and Design Evolution. The Final Status section briefly describes pump operation, latest test results, and final conclusions of the current design and is presented first for easy review and reference. The second section, Design Evolution, describes in detail the original design analyses, test results, and design modifications as they occurred.

FINAL STATUS

DESCRIPTION

The ALWT variable-displacement (VD) fuel pump is very simple in concept. Pistons draw in and expel fuel by their reciprocating motion produced when they encounter an angled plate while their cylinder-block housing rotates. The piston stroke is varied by the pitch of the angled plate, which is determined by a cam positioned by a wheel and worm gear mechanism. This worm gear is rotated by a DC motor through a speed-reducing gear box. The cylinder block bears against the pump head via a conical-angle surface in which are located the cylinder's intake and exit ports. The inside of the pump is completely filled with incoming fuel, which acts as a coolant and minimum-grade lubricant.

The ALWT VD fuel pump is shown in Figs. 1 and 2. This prototype design is approximately 3 in. in diameter by 5 3/4 in. long. The 38-V electric control motor projects beyond this diameter by 4 1/2 in. This compact configuration takes advantage of the available space on the ALWT bulkhead. The fuel pump is mounted on the ALWT bulkhead by means of five No. 10-32 UNF screws. It is located on the starboard side of the torpedo at a radial distance of 3.835 in. from center, such that the 66-tooth pump gear will mesh with the 118-tooth drive gear. The drive gear rotates clockwise (looking towards the bow of the torpedo), while the fuel pump turns counter-clockwise at a speed ratio of 1.7879 to 1. A cross-sectional view of the fuel pump is shown in Fig. 3, and a photograph of the original hardware is shown in Fig. 4. The fuel pump consists of a nine-piston cylinder block, which rotates against a ported pump head. The nine pistons (9) are spring loaded against an angle-plate (11). The angular position of the plate determines the displacement of the pistons as the cylinder block (8) rotates. The keyed shaft (7) rotates the cylinder block, which is supported by the conical-angle bearing surface at its forward end. Thus, the matched conical surfaces of the block and pump head act as the porting seal and also as the self-centering bearing.

The fuel pump shaft (7) is surrounded by the piston-stroke control mechanism. This mechanism consists of the angle-plate (11), the cam (13) and wheel gear (14), and various thrust bearings and washers (10, 33, 12, 34, 15, 35), and is designed to operate without encountering the rotating pump shaft. The angle-plate is supported by and pivots about two screws through either side of the pump body. The angle-plate rests on the cam, which determines its angular position. The cam can be rotated 180 deg by the worm gear, changing the pitch of the angle-plate from 6 to 15 deg.

Fuel enters the fuel pump through the fitting in the sidewall of the pump body. It enters the piston cylinders via radial porting at the base of the conical seal boss of the pump head. The pistons draw fuel into their cylinders during the first 180-deg rotation of the cylinder block and expel fuel at 4800 psi during the second 180-deg rotation of the block. Fuel exits the pump head via the exhaust port in the conical seal and the exit fitting at the top of the fuel pump. The conical seal is the general term used to describe the assembly of the male portion of the conical valve, being the boss of the pump head. The female portion of the conical valve is the ported area in the top of the cylinder block. The conical seal is designed to always provide a positive sealing pressure between the pump head and the rotating cylinder block. The sealing surfaces are angled (40 deg) such that the entrance and exit pressures exert reduced vector forces that tend to separate the sealing surfaces. The ports are arranged in the conical seal such that valve timing is optimized. The result ensures that the pistons will not compress the fuel beyond the exit pressure of the pump, the flow always having someplace to go on the upstroke of the piston.

Table 1 presents the design equation of the ALWT VD fuel pump. This equation was used to optimize the geometry of the pump while meeting the ALWT design criteria. If the dimensions of the final geometry are substituted into the design equation, the volumetric flow rate will range from 1.8 to 4.8 gpm at 100% volumetric efficiency. This is plotted in Fig. 5. At a volumetric efficiency of 67%, the fuel pump will supply 1.2 to 3.2 gpm at 4200 rpm pump shaft speed, this being the ALWT design requirement.

TEST RESULTS

Although many preliminary tests proved the need for various modifications to the original pump design, the final test results were very promising. Figures 6, 7, and 8, show these final test results. Figure 6 shows that the variable-displacement fuel pump with a 40-deg conical seal meets the ALWT fuel pump flow requirement: 70% volumetric efficiency producing 1.4 to 3.3 gpm. The system efficiency shown in Fig. 7 is approximately 16% greater than an equivalent fixed-displacement fuel pump with external fuel bypass. Performance curves for pumping Otto fuel are presented in Fig. 8. The 5-min 43-sec test of 1 Dec 77 produced a volumetric efficiency of 94 to 75%, corresponding to 1.7 gpm at 1000 psi and 3.5 gpm at 4500 psi, respectively. Although this fuel test was successful, one major problem recurred: excessive wear of the conical seal surfaces of the cylinder block and pump head. Materials studies and testing were accomplished to determine compatible bearing materials. These tests concluded that the original bearing bronze cylinder block against the hardened tool steel pump head is the best interface for the speeds and loading encountered.

A final redesign involved: (1) incorporation of a large radial needle bearing to support and stabilize the rotating cylinder block; and (2) increasing the conical bearing surfaces of the cylinder block and pump head. These modifications on the 40-deg conical valve are shown in Figs. 9 and 10. The circumferential and radial grooves shown on the enlarged bearing surface of the cylinder block relieve the hydraulic forces that would otherwise be increased due to the enlarged bearing surfaces. The net result is a decrease in the bearing pressure acting upon the valve surfaces of the cylinder block and pump head. Only one test was accomplished with this hardware before the ALWT open-cycle effort was terminated. This test showed substantial improvement in wear but periodic flow and pressure fluctuations in the fuel pump output. See Fig. 11.

CONCLUSIONS

The ALWT variable-displacement fuel pump will meet the ALWT design requirement. It needs further development to ensure longer life and consistent performance. Such development may involve additional materials testing and redesign of the conical bearing surfaces.

DESIGN EVOLUTION OF VD FUEL PUMP

ORIGINAL DESIGN CONSIDERATIONS AND ANALYSES

PISTON-STROKE CONTROL MECHANISM

The object of controlling the length of piston stroke by mechanical means was to minimize control hysteresis while maximizing system sensitivity and accuracy. Previous variable-displacement fuel pumps that incorporated hydraulically actuated control mechanisms required hydraulic valves and circuitry. These valves and circuits produced excessive hysteresis, especially at the higher output pressures of the fuel pump. It was thought that the direct cam-controlled swash plate would have the required mechanical advantage for positioning the swash plate without the need of a locking mechanism or maintaining a detent force.

Figure 12 is a cross-sectional view from the top of the fuel pump. This view shows the control motor (58) rotating the worm gear (52) through a flexible coupling (60) and shaft (54). The worm gear (52) drives the wheel gear (50), which is keyed to the cam (44), shown in Fig. 13. The gear ratio between the worm and the wheel gear is 50:1. Figure 14 is a photograph showing the meshing of these gears as assembled in the pump housing. The worm shaft (54) requires a maximum torque of 32 in.-oz. Because the pitch angle of the worm gear system is 6 deg, the system is self-locking, i.e., the wheel gear can be turned only by rotating the worm gear; the wheel gear can not drive the worm gear. The cam (44) may rotate 180 deg until a boss (62) on its periphery encounters either side of the stop (63). The stop (63) is a semicircular plate, 173-deg arc length, held to the inner diameter of the pump body (12) by three sealed screws (64). The top surface of the cam (44) is angled at 4.5 deg, while the top surface of the angle-plate (28) is angled at 7.5 deg. As the cam (44) rotates through 180 deg, the pitch of the angle-plate (28) is changed from 3 to 12 deg (7.5 ± 4.5 deg). The angle-plate (28) has two pivot screws (38), shown in Fig. 15, 180 deg apart, which allow it to rotate. These pivot areas of the angle-plate (28) are beyond the diameter of the thrust washer (48) because the high side of the cam (44) must pass through the pivot line (38), shown in Fig. 13. Two hemispherical bosses on the underside of the angle-plate (28), 180 deg apart and 90 deg from the pivot screws (38), encounter the thrust washer (48) of the cam (44). See Fig. 16. The thrust washer (48) is stationary with respect to the angle-plate (28). This is because the pivot line (38) of the angle-plate (28) lies in the plane of the interface of the hemispherical bosses and the thrust washer (48), and because the cam (44) is able to rotate underneath the thrust washer (48) due to reduced friction by the thrust bearing (46). The pivot line (38) must be coincident with the boss-washer interface so that the angle-plate (28) will always be supported at both its bosses with absolute minimum clearance and, thus, zero rocking motion.

The thrust washer (42) rotates with the pistons (24), cylinder block (18), and shaft (34). This thrust washer (42) rotates on the thrust bearing (40). The ends of the pistons (24) are spherical to accommodate the changing pitch of the thrust washer (42). These spherical ends make point contact on the thrust washer (42). Each piston (24) bears against the thrust washer (42) with a force varying from 7.1 lb spring force to 365 lb due to the pump output pressure. This results in a worst-case 1855-lb axial thrust load through the thrust washers, thrust bearings, and cam into the base of the pump body (12). The net result is a torque requirement of 56.6 in.-lb to rotate the wheel gear, and because of the 50:1 gear ratio, a torque requirement of 1.13 in.-lb to rotate the worm gear. Considering a gear efficiency of 75%, this estimated torque becomes 1.51 in.-lb. Adding a safety margin of 30%, the motor requirement was specified at 32 in.-oz. To shift from the extremes of the ALWT engine speed requirement in 2 sec will require the worm-gear shaft to rotate at approximately:

$$\frac{3}{4}\left(\frac{180^\circ \text{ cam rotation}}{2 \text{ sec}}\right) \frac{50 \text{ rev. worm}}{360^\circ \text{ cam rotation}} \times \frac{60 \text{ sec}}{\text{min}} = 560 \text{ rpm}$$

A modified MK 46-1 DC actuator motor was used for initial experiments and is shown in Figs. 1, 2, and 12. The modifications involved a change in speed ratio (18.78 to 1) and drive shaft dimensions.

Figure 17 presents the original estimated fuel flow rate as a function of cam angular position and fuel pump shaft speed. The engine requirement at minimum and maximum depth is drawn over these fuel pump estimates. The engine requirement was taken from limited empirical data shown in Fig. 18.

CONICAL VALVE DESIGN

Figure 19 shows the original hardware of the unique conical valve interface. This interface between the fuel pump head and the rotating cylinder block is unique because it serves three functions simultaneously. First, it acts as a radial bearing surface to center the forward end of the rotating cylinder block. There is sufficient clearance between the pump shaft and the forward portion of the bore in the cylinder block such that the forward end of the cylinder block can seek its center upon the conical bearing surface of the pump head. The aft portion of the bore in the cylinder block fits tightly around the pump shaft, while a key interfaces the pump shaft along the entire length of the bore. Second, the interface between the pump head and the rotating cylinder block acts as a dynamic valve seal to properly sequence fuel into and out of the piston-cylinders. Third, the interface acts as an axial bearing surface, angled to maintain a net force that pushes the cylinder block towards the pump head, while providing a large wear surface.

Figure 20 and 21 show a cross-sectional view of the fuel pump and a perspective view of the pump head and cylinder block. The face of the dynamic valve seal consists of the front face (38) of the cylinder block (16) and the rear face (40) of the pump head. This rear sealing face (40) has an arcuate intake port (42), which is in communication with the fuel within the housing (12) through a passageway (44). Further, the rear sealing face (40) has an arcuate outlet port (46), which communicates with a fuel outlet fitting (48) via a passageway (50). The sealing face (38) of the cylinder block has a series of ports (54) that alternately communicate the piston cylinders (20) with the arcuate inlet and outlet ports (42) and (46), respectively, of the pump head as the cylinder block (16) is rotated by the shaft (18). With this arrangement, as the cylinder block rotates, fuel will be sucked through the inlet port (42) as the pistons move around and down the pitched thrust washer (30), and fuel will be discharged through the outlet port (46) when the same pistons move around and up the pitched thrust washer (30).

The sealing face (38) of the cylinder block (16) and the sealing face (40) of the pump head also act as bearing surfaces to support the net forces pushing the cylinder block against the pump head. These bearing faces (38 and 40) are at an acute angle with respect to the axis of rotation of the cylinder block (16) in order to reduce the axial forces that act on these faces and tend to separate the cylinder block from the pump head. Therefore, the sum of the forces inside the piston cylinders (20) acting to push the cylinder block (16) toward the pump head is greater than the sum of the axial components of the forces caused by the inlet and outlet pressures acting on and tending to separate the bearing surfaces (38 and 40). Figure 22 describes the net force pushing the cylinder block towards the pump head, $(F_{net})_y$. Note that the forces $(F_i)_1$ and $(F_o)_2$ are reduced by the sine of the angle θ (the conical angle) when calculating the net axial force. The bearing pressure on the conical bearing surfaces (38 and 40) is this net force divided by the axial projection of the bearing area.

Figure 23 shows the net bearing force and bearing pressure for conical angles 45, 40, and 35 deg as a function of fuel pump output pressure. As the conical angle becomes steeper, θ becoming smaller, and the bearing force and pressure increase. As the bearing pressure increases, the sealing effectiveness across the conical surfaces (38 and 40) increases and less leakage occurs between the cylinder block and the pump head. But as the bearing pressure increases, the conical bearing surfaces experience greater wear. Notice that beyond an output pressure of 4000 psi, the bearing pressure upon the 35-deg conical angle is greater than that allowed for bearing bronze.

SPEED CONTROL CIRCUIT

The advanced Lightweight Torpedo (ALWT) speed control system is to consist of a solid state feedback circuit that controls the output fuel flow of the variable-displacement fuel pump as required by the ALWT engine. The total speed of the engine must be controlled to within ±87 rpm at both the fast speed of 4800 rpm and the slow speed of 1745 rpm. This represents a speed deviation of ±2% and ±5%, respectively. The speed control circuit must have the capability of two additional intermediate, discrete speed settings and the potential of being modified for an infinite number of speed settings.

It is desirable to shift from one extreme engine speed to the other in approximately 2 sec, although 5 sec is acceptable. The speed control circuit must be capable of being conveniently adjusted to meet the ±87 rpm speed tolerance requirement should first tests fall outside this range. The speed control circuit must control the fuel flow produced by the ALWT variable displacement fuel pump according to the propulsion characteristics of the ALWT engine.

VD FUEL PUMP CHARACTERISTICS

The performance characteristics of the ALWT VD fuel pump are shown in Fig. 17, which shows the fuel flow produced as a function of the fuel pump speed and the cam position in degrees. Since the fuel pump is geared directly to the outer shaft of the engine, the speed of the fuel pump is 0.8939 times the total engine speed. Thus, when the engine is running at 4800 rpm, the fuel pump speed will be 4290 rpm. The present fuel pump assembly is shown in Fig. 13. The cam referred to in Fig. 17 is item 44 in the fuel pump assembly. The cam is positioned by a 50:1 worm gear, which requires a 32-in.-oz torque at the worm shaft (item 54 in Fig. 12).

A modified MK46-1 DC actuator motor is presently being used to rotate the worm-gear shaft, as shown in the assembly drawing Fig. 1. The modifications involved a change in speed ratio (18.78:1) and drive shaft dimensions. The speed ratio modification involved replacing the original Globe Industries gear box 5A2685 to that of 5A2352. The drive shaft was reduced to $0.2494^{+0.0003}_{-0.0000}$ in. diameter and shortened to 1.035 ± 0.015 in. Actual, preliminary data indicates a requirement of 3.28 W at the motor to rotate the worm-gear shaft at 137.73 rpm. See table 2. To shift from the extremes of the ALWT engine speed requirement in 2 sec would require the worm-gear shaft to rotate at approximately:

$$\frac{3}{4}(180^\circ \text{ cam rotation}) \times \frac{50 \text{ rev. worm}}{360^\circ \text{ cam rotation}} \times \frac{60 \text{ sec/min}}{2 \text{ sec}} = 562.5 \text{ rpm}$$

The modified MK 46-1 DC motor is preferred over a stepper motor because of its smaller size and weight and because it draws less current than a stepper motor. A stepper motor would provide a detent torque, which is not required because the worm-gear system is self-locking. If a stepper motor is chosen for the control circuit design, a means of turning off the detent power will be necessary. The maximum steady-state power available is ±40 V at 2 A.

ENGINE CHARACTERISTICS

The speed control circuit must provide the engine with the required fuel flow rate as specified in Fig. 18. Given a specified total engine speed, the fuel flow rate must be matched by the output of the fuel pump and governed by the speed control circuit to within ±87 rpm. The engine time response for an instantaneous step increase may be estimated using the data presented in Fig. 24. This is actual sea run data for the H-engine (very similar to ALWT) and thus, contains all the inertia effects. Alternator frequency from these tests can be multiplied by 2.228 to obtain total engine RPM.

The total engine speed will be instrumented by a magnetic sensor embedded in the crankcase of the engine. The sensor system can be mechanically decreased to produce between 1 and 60 pulses per total engine revolution. Therefore, the total engine fast speed can be devised to produce an 80- to 4800-Hz pulse signal, while the total engine slow speed can provide a 29- to 1745-Hz pulse signal. The optimum design of the speed control circuit will specify the required ratio of pulses per total engine revolution. In its present configuration, the engine will produce 4800-Hz and 1745-Hz, fast and slow speeds, respectively.

ACTUAL SPEED CONTROL CIRCUIT DESIGN

The NOSC torpedo speed is regulated by a closed loop control system. The major components of the control system are the variable-displacement fuel pump, a DC motor for pump displacement control, an amplifier to drive the DC motor, and low-level electronic circuitry that provides engine speed monitoring and servo-loop compensation networks. A schematic outline of the speed control circuit is shown in Fig. 25.

The engine speed is measured by magnetic pickups mounted on the propeller drive shaft. The total engine speed, the sum of the outer plus the inner shaft speed is read out in the form of low-voltage pulses whose frequency is proportional to the engine speed. The voltage pulse train is conditioned and converted to a DC voltage whose magnitude is proportional to the engine speed. As indicated in Fig. 25, the engine speed is compared with the desired speed, forming an error signal. This signal is filtered to remove any high-frequency noise. A compensation circuit then operates on the error signal. The purpose of the circuit is to provide servo-loop compensation needed to optimize the speed control performance. An integrated circuit operational amplifier implements this circuit.

Following the compensation network, a pulse width modulator converts the DC signal voltage into a 3-kHz voltage pulse train. The pulse width is proportional to the voltage signal level. A separate logic signal is generated that determines the sense of the error signal.

The pulse width modulator then controls a power amplifier. The amplifier consists of power transistors operated in the switching mode. This design minimizes the power dissipation and results in an amplifier with high efficiency (greater than 85%).

The voltage to the pump control motor is therefore a voltage pulse train whose frequency is 3 kHz and whose width is proportional to the magnitude of the error signal. The polarity of the voltage pulses is determined by the sense of the error signal.

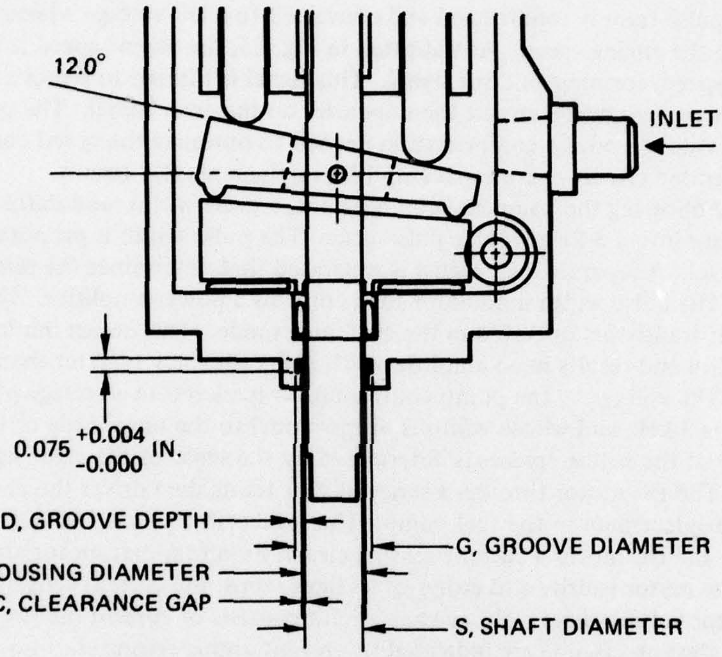
The DC motor through a series of gear trains then drives the cam that controls the piston-displacement in the fuel pump. The total cam travel is 180 deg. To provide protection for the DC motor a current-sensing circuit monitors that motor armature current. In the event the motor is driven to either of its limit stops, the current-sensing circuit disconnects the motor voltage drive. The sensing circuit consists of current monitors and logic circuits. Special start-up circuits are indicated to prevent undue engine start-up speed transients. The logic control circuits would be tailored to meet this need.

With the exception of the servo compensation networks and the start-up logic, all the electronic circuits associated with the speed control were designed and tested. Selection of system gains and compensation network parameters are dependent on the static and dynamic characteristics of the pump and engine. With this data, it was planned to simulate the control system using a digital computer program to determine the values of the various control parameters. These static and dynamic characteristics of the pump operating on the engine were about to be obtained when the ALWT open-cycle effort at NOSC was terminated.

SHAFT SEAL DESIGNS

ROTARY O-RING SEAL FOR FUEL PUMP SHAFT

The decision to use a standard O-ring seal for the fuel pump shaft was based on the following design considerations: (1) The "PV" value that the seal will incur is within the limits of O-ring capability, i.e., 180 psi differential pressure across the O-ring cross section multipled by the shaft velocity of 2625 fpm, giving 4.7×10^5 psi, fpm; (2) The ease of machining the female groove in the pump body; (3) The economy of a standard O-ring seal over other more elaborate seal configurations; and (4) The incorporation of a design that maintains peripheral compression on the O-ring and prevents seal failure caused by the Gow-Joule effect. The peripheral compression design functions by using an O-ring whose outside diameter is slightly larger than the maximum diameter of the groove in which it is installed. The design calculations are as follows:



GIVEN:

$S = 0.6251 \begin{array}{l} +0.0000 \\ -0.0003 \end{array}$ in. diameter

Shaft speed, $N \approx 4200$ rpm

Differential Pressure, $\Delta P = 180$ psi.

CALCULATIONS:

1). Clearance, $C = 0.005$ in. nominal, to prevent O-ring extrusion under $\Delta P = 180$ psi.

2). Housing diameter, $H = S + 2C$

$$H = 0.6251 + 2(0.005) = 0.635 \pm 0.002 \text{ in.}$$

3). Groove depth, $D \approx (0.95)$ (W, O-ring cross-sectional diameter):

$$D = (0.95)(0.070 \pm 0.003) = 0.0665 \text{ in. nominal}$$

4). Groove diameter, $G = S + 2D$

$$G = 0.6251 + 2(0.0665) = 0.758 \pm 0.001 \text{ in.}$$

5). O-ring size required:

$$OD = 1.08\,(0.7581) = 0.8187 \text{ in.}$$

$$ID = 0.8187 - 2(0.070) = 0.6787 \text{ in.}$$

therefore use nearest size O-ring:

$$2\text{-}017 \begin{cases} OD = 0.816 \text{ in.} \\ ID = 0.676 \text{ in.} \\ W = 0.070 \pm 0.003 \text{ in.} \end{cases}$$

6). O-ring cross section when installed in groove:

$$W' = 1.022\,(0.070 \pm 0.003) = 0.0715 \pm 0.0031 \text{ in.}$$

then,

$$W'_{min} = 0.0684$$

$$W'_{max} = 0.0746$$

7). Actual squeeze of O-ring:

a). Minimum squeeze $= W'_{min} - \left(\dfrac{G_{max} - S_{min}}{2}\right)$

$$= 0.0684 - \left(\frac{0.759 - 0.6248}{2}\right)$$

Minimum squeeze $= 0.0013$ in.

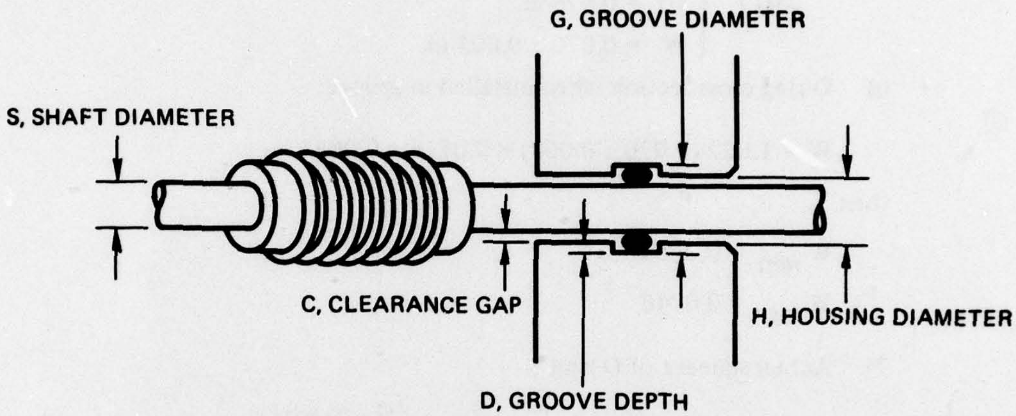
b). Maximum squeeze $= W'_{max} - \left(\dfrac{G_{min} - S_{max}}{2}\right)$

$$= 0.0746 - \left(\frac{0.757 - 0.6251}{2}\right)$$

Maximum squeeze $= 0.0087$ in.

ROTARY O-RING SEAL FOR WORM GEAR SHAFT

The same design considerations were made for the seal for the worm gear shaft as were made for the fuel pump shaft. An additional consideration was made relative to supporting the worm gear shaft. Since it was decided to allow the seal housing (fuel pump body) to also act as a journal bearing for the worm gear shaft, the clearance gap between the housing and the shaft must be kept to a minimum. The design calculations follow:



GIVEN:

$S = 0.1869 \begin{smallmatrix} +0.0000 \\ -0.0004 \end{smallmatrix}$ in. diameter

Shaft speed, N = 560 rpm
Differential Pressure, ΔP = 200 psi

CALCULATIONS:

1). Clearance gap, C = 0.005 in. will prevent O-ring extrusion for 200 psi differential pressure, but since the housing must act as a journal bearing for the shaft, reduce the clearance gap to 0.00035 to 0.0007 in., this being the recommended clearance for running fits RC-3 and RC-4.

2). Housing diameter, H = S + 2C

$H = 0.1867 + 2(0.0004) = 0.1875$ in. nominal

$\therefore H = 0.1873 \begin{smallmatrix} +0.0004 \\ -0.0000 \end{smallmatrix}$ in. diameter

3). Groove depth, D = (0.92) W

$D = (0.92)(0.070) = 0.0644$ in. nominal

4). Groove diameter, G = S + 2D

$G = 0.1869 + 2(0.0644) = 0.315 \pm 0.001$ in. diameter

5). O-ring size required:

$$OD = 1.08\,(0.315) = 0.341 \text{ in.}$$

$$ID = .341 - 2(.070) = 0.201 \text{ in.}$$

therefore use nearest size O-ring:

$$2\text{-}009 \begin{cases} OD = 0.348 \text{ in.} \\ ID = 0.208 \text{ in.} \\ W = 0.070 \text{ in.} \end{cases}$$

6). O-ring cross section when installed in groove:

$$W' = 1.022\,(0.070 \pm 0.003) = 0.0715 \pm 0.0031 \text{ in.}$$

then,

$$W'_{min} = 0.0684 \text{ in.}$$

$$W'_{max} = 0.0746 \text{ in.}$$

7). Actual squeeze of O-ring:

a). Minimum squeeze $= W'_{min} - \left(\dfrac{G_{max} - S_{min}}{2}\right)$

$$= 0.0684 - \left(\frac{0.316 - 0.1865}{2}\right)$$

Minimum squeeze $= 0.0037$

b). Maximum squeeze $= W'_{max} \left(\dfrac{G_{min} - S_{max}}{2}\right)$

$$= 0.0746 - \left(\frac{0.314 - 0.1869}{2}\right)$$

Maximum squeeze $= 0.0111$ in.

FIRST SERIES TEST RESULTS

The VD fuel pump was first tested on March 9, 1977. The test stand configuration is presented in Figs. 26 and 27. A volumetric efficiency of 79% was obtained with a maximum output of 3.0 gpm at 110 psi without orifice restriction at the pump exit. The pressure-volume data was then obtained with a 0.052-in. orifice to simulate the ALWT engine requirement. This data appears in Fig. 28 and indicates the maximum output as being 2.5 gpm at 2450 psi output pressure. This curve approximately follows the expected theoretical curve and was actually determined to be:

$$P_{f_o} = 360.9\,(\omega_f)^{2.09} \quad ,$$

the output pressure, P_{f_o}, being an exponential function of the flow rate, $\dot{\omega}_f$. Inspection of the fuel pump showed that what had previously been thought to be the maximum output setting by the cam was actually in error by 50 deg. The boss on the cam that acts as a mechanical stop was therefore removed so the maximum flow could be obtained, i.e., 3.5 gpm with 92.9% volumetric efficiency without orifice restriction at the pump exit (table 3).

A theoretical analysis of the data determined that even with the cam rotated for the maximum output flow, the maximum output pressure that could be expected was 3400 psi at 2.93 gpm. This is the extrapolated point "Expected with maximum cam rotation" presented in Fig. 28. In order to meet the desired output criteria for the ALWT, this curve was again extrapolated, indicating 3.4 gpm at 5000 psi output pressure. Assuming the same volumetric efficiency for the pump working against the 0.052-in.-diameter orifice, the angle-plate assembly would have to be pitched at 15 deg.

Table 3 presents the results of internal diagnostic tests consisting of pressure and flow measurements at two pump speeds and two cam settings without a restriction orifice at the exit of the pump. Two phenomena were experienced as the inlet pressure to the pump was increased: (1) the slight decrease in output flow with an increase in inlet pressure indicated valving overlap, the cylinder port being instantaneously open to both inlet and exit slots of the valve at Top Dead Center and Bottom Dead Center positions of the pistons; and (2) the increase in output flow to a maximum with an increase in inlet pressure indicated choking or restriction at the inlets to the cylinders. Calculations in Fig. 29 verified the fact that choking would occur through the 0.100-in.-diameter valve ports for inlet pressures less than 29 psi.

As the maximum pitch of the angle-plate was approached, excess vibration was noticed emanating from the VD fuel pump. Post run inspection revealed that the spherical ends of the pistons were wearing grooves into the bearing surface of the thrust washer. The thrust washer was also found to be slightly warped in the direction of piston pressure. There was also indication of "piston bounce" on the worn surface of the thrust washer.

FIRST-GENERATION MODIFICATIONS

Due to the first series test results of the VD fuel pump, the following three modifications were accomplished: 1.) Replace the 7.5-deg angle-plate with a 10.5-deg angle-plate such that the angle-plate assembly (piston stroke control mechanism) could attain a maximum 15-deg pitch at maximum cam setting resulting in the required 3.4 gpm at 5000 psi at the pump exit; 2.) Eliminate the valving overlap by increasing the web size between the exit slot and the inlet slot of the valve; and 3.) Increase the size of the cylinder inlet ports and valve porting slots to minimize the choking problem.

The object of increasing the angle on the angle-plate was to increase the fuel output of the VD fuel pump by simply increasing the piston stroke. Another method considered was to increase the angle on the cam by 3 deg, but this modification would reduce the control sensitivity of the piston stroke control mechanism. It would also produce a shorter minimum piston stroke when the cam was positioned at its minimum setting, $\alpha = 0$ deg. The results of these considerations can readily be seen from the equation for the pitch of the angle-plate appearing in Table 1:

(Pitch of Angle-Plate) = (Angle on Angle-Plate) – (Angle on Cam) (Cosine of Cam rotation)

1. Using the original 7.5-deg angle-plate and replacing the original 4.5-deg cam with a 7.5-deg cam, the pitch of the angle-plate is

$$\phi = 7.5^\circ - (4.5^\circ + 3^\circ)\cos(0 \text{ to } 180^\circ).$$

As the cam rotates from 0 to 180 deg, the pitch of the angle-plate will increase from 0 to 15 deg. The desired 15-deg pitch is obtained by this modification method, but the minimum pitch of zero will result in no piston stroke whatsoever.

2. Replacing the original 7.5-deg angle-plate with a 10.5-deg angle-plate and using the original 4.5-deg cam, the pitch of the angle-plate is

$$\phi = 10.5 - (4.5°) \cos (0° \text{ to } 180°).$$

As the cam rotates from 0 to 180 deg, the pitch of the angle-plate will increase from 6 to 15 deg. The desired 15-deg pitch is obtained while the original 9-deg range for the piston stroke control mechanism is maintained. Therefore, this modification method was chosen over that of the cam.

The increased pitch of the angle-plate necessitated a slight modification to the cylinder block. The aft edge of the cylinder block was beveled at 10 deg to allow clearance for the edge of the piston thrust washer when rotated at its maximum pitch of 15 deg. This modification to the cylinder block can be seen in Fig. 30.

Valving overlap was eliminated by designing a new pump head. This pump head incorporated wider webs between the inlet and exit ports of the valve. These webs were wide enough to completely cover the cylinder ports in the cylinder block. The cylinder ports, themselves, were widened to minimize choking and decrease pressure losses at the valve interface. The new pump head and modified cylinder block appear in Fig. 31. The cylinder ports consist of nine slots, each $24\,^{+1/2}_{-0}$ deg circumferentially long by $0.094\,^{+0.008}_{-0.000}$ in. wide, spaced 16 deg apart. The webs between the inlet and exit ports of the valve are each an arc length of $25\tfrac{1}{4}\,^{+0}_{-1/2}$ deg.

The valve timing is arranged such that the exit port is being closed while the piston is 2 deg beyond Top Dead Center (TDC). This means the piston is on its way back down when the exit port closes, and assures the piston will not compress fluid beyond the exit pressure; the flow is guaranteed exit during the compression stroke. The exit port closes when the piston moves 0.0001 to 0.0002 in. past TDC. Towards Bottom Dead Center (BDC), the port entrance closes when the piston is still moving down. This means the piston has to move 0.0001 to 0.0003 in. with its entrance port closed and before the exit port opens. The total length of piston stroke in which neither the entrance or exit port is open is:

$$(25\tfrac{1}{2}\,^{+0}_{-1/2}) - (24\,^{+1/2}_{-0}) = 1\tfrac{1}{2}\,^{+0}_{-1} \text{ deg}$$

or

$$(1 - \cos 1\tfrac{1}{2}\,^{+0}_{-1})\,\frac{1.390}{2}\,\tan 15 = 7 \times 10^{-6} \text{ in. to } 0.0001 \text{ in.}$$

and this occurs only when the piston is moving down to BDC or from TDC. This assures excess pressure will not be produced within the cylinder block, tending to unseat the seal at the valve.

In order to minimize choking of the flow into the cylinders, the cylinder ports were widened into slots. Each port was widened to approximate the flow area of those in the MK 46-1 fuel pump, 0.0233 in.^{2}. The inlet ports in the pump head (ports 44 shown in Figs. 20 and 21) were correspondingly enlarged to accept the increased flow. This results in a choking prevention pressure of 2.75 psi at the pump inlet, compared to the original 28.5 psi calculated in Fig. 29.

Three modifications were incorporated into the VD fuel pump to eliminate vibration and wear occurring between the pistons and the thrust washer. Flattened bearing balls were added to the ends of the pistons. These flattened bearing balls spread the piston forces over a greater area on the thrust washer, while rotating to accommodate the variable pitch of the angle-plate and reduce the wear between the pistons and the thrust washer. The thrust washer was made of 8620 carbon steel, instead of the leaded B113 steel material, to improve its bending strength. The piston return springs were changed to the stiffer music wire material, instead of the stainless steel wire, to eliminate the "piston bounce" problem.

SECOND SERIES TEST RESULTS

The VD fuel pump with first-generation modifications was tested on the 20 and 21 of June 1977. The modifications consisted of a new pump head with larger porting and improved timing, larger ports in the cylinder block, ball-and-socket pistons to decrease the bearing pressure between the pistons and the thrust washer, a material change to strengthen the thrust washer, a steeper angled angle-plate to increase the piston stroke, and stiffer piston springs to eliminate piston bounce. These modifications were introduced into this test series in a consecutively additive method to determine their significance. Tests No. 1 and 2 incorporated the new pump head, large ported cylinder block, new 8620 piston washer, and original pistons and angle-plate. Test No. 1 was run without pump back pressure (no orifice), while all following tests were run with a 0.052-in. orifice at the pump exit. Test No. 3 used the new hardware of tests No. 1 and 2 plus the new pistons and balls, new angle-plate, and new piston springs. Tests No. 4 and 5 also had the beveled cylinder block so that the new angle-plate could be rotated to its maximum angle of 15 deg. The results of these tests are shown in Figs. 32 through 35.

Figure 32 shows that the new pump head and bored-out ports in the cylinder block result in an increased output pressure of 600 psi for the same maximum flow of 2.6 gpm, both using the original 7.5-deg-included-angle angle-plate. The increased output pressure is due to the improved valve timing design of the new pump head, which assures against valving-overlap (the cylinder port being instantaneously open to both inlet and exit slots of the valve). Notice that the conical valve sealing surface of the new head did not match that of the cylinder block as well as the original pump hardware. This is apparent between the 76% and 92.9% volumetric efficiencies in the no-orifice tests. Evidently, the leakage across the conical valve decreases with wear-in-time, as can be seen in Fig. 33. The volumetric efficiency increased throughout this test series until the failure during Test No. 4 and especially Test No. 5.

Post inspection of Test No. 4 showed excessive wearing of the piston bearing balls on the thrust washer. Metal flakes were noticed welded to the thrust washer and inside the fuel pump. The conical valve had minor scratches but was considered operational. The piston bearing balls were polished, except for one which was replaced with a tungsten carbide ball bearing and the thrust washer was replaced. The pump was then assembled with the angle-plate set at its maximum angle of 15 deg, and run at standard, simulated ALWT speed. The result was the one point of Test No. 5 shown in Fig. 34. Although the maximum output of 3960 psi and 3.1 gpm is quite an improvement over the original pump design shown in Fig. 32, it is still short of our 4800-psi, 3.2-gpm ALWT design goal. Post inspection of Test No. 5 showed increased wear of the piston bearing balls, except for the one tungsten carbide ball, and extensive scratches on the conical valve sealing surface of the cylinder block. Because of the wear problem the input horsepower was very great, as shown in Fig. 35, and because of the scratches on the conical valve, the volumetric efficiency drastically decreased, as shown in Fig. 33. In conclusion, it was assumed that the required ALWT fuel pump performance could be met if the wear problem between the piston bearing balls and the piston thrust washer could be eliminated.

Table 2 presents the results of the actuator motor characteristics before and during the testing of the VD fuel pump. From the first two columns in the table, the instrumentation potentiometer and gear can be seen to contribute 2.91 in.-oz of frictional torque (23.37 minus 20.46). Then, the torque required to rotate the actuation motor without the instrumentation accessories can be calculated as: 32.16 in.-oz – 2.91 in.-oz = 29.25 in.-oz. This is very close to the originally approximated requirement for the actuator or speed control motor.

SECOND-GENERATION MODIFICATIONS

Tungsten carbide bearing balls and a Graph-Air piston thrust washer were two material changes to improve the wear capabilities of these parts. Graph-Air material is a trade name for graphite-impregnated tool steel. Its application for a thrust washer combines the hardenable surface of heat-treatable tool steel with the very good lubricity properties of graphite. In order to investigate "wear-in" characteristics of the conical valve, an instrumentated run-in period was recommended for the next series of fuel pump tests. In this manner, the mating surfaces of the conical valve would be allowed to form an efficient seal before the high pressure (and, therefore, high load) performance data is taken. It was thought that this run-in period would prevent scratching and excessive wear of the conical valve surfaces.

THIRD SERIES TEST RESULTS

The Variable-Displacement Fuel Pump with first- and second-generation modifications was tested on 11 and 12 July 1977. The second-generation modifications involved the use of Graph-Air material for the piston thrust washer and tungsten carbide piston bearing balls to eliminate the excessive wear problem at their interface. These material changes completely eliminated the wear problem. In fact, after running the fuel pump for a total time of over one-half hour, there was virtually no wear between these parts.

Allowing a "run-in" period, as recommended in the previous test series, showed that the output pressure increases by 25% (flow rate by 10%) within the first minute of run-in time (see table 4). Notice that after 6 min, the output pressure has fallen by 10%, to 2650 psi from 2930 psi. This indicates that the sealing surfaces tend to wear out soon after wearing in. Changing the cylinder-block material to naval brass, in place of bearing bronze, may extend the life of the seal.

Figure 36 shows how the fuel pump performance compares with the theoretical output based on 100% volumetric efficiency. The data matches the theoretical form, except the actual volumetric efficiency decreases from 81 to 73% with increase in cam rotation and output pressure. This decrease in volumetric efficiency is a function of the leakage across the conical valve seal, the leakage increasing with increase in output pressure. It is not a function of seal "wear-out" time because the test was made from the maximum 180-deg cam setting to the 0-deg cam setting (from high output pressure to low output pressure) and well after the allowed "run-in" period.

Table 5 shows that "run-in" is a function of pump output pressure. Notice that the no-orifice volumetric efficiency improved by 5% after the 0.038-in. orifice was used to obtain the maximum 5000 psi output pressure. Post inspection of the conical seal surface indicated increased bearing pressure and wear. Evidently, the cylinder block is forced against the pump head in direct proportion to the output pressure, but the leakage across the conical valve seal increases exponentially with output pressure as can be seen in Fig. 37, plotted from the data of table 6. This means that the leakage across the conical valve seal can be decreased if the force tending to push the cylinder block towards the pump head is increased. This can be accomplished by either increasing the size of the pistons, decreasing the angle of the conical valve, or decreasing the cross-sectional bearing area of the conical seal. The latter is the least effective, but will be tried since it is more easily accomplished.

While the overall performance of the VD pump was successful, there were two failures found after one-half hour of run time. One of the pistons broke in two pieces right behind the O-ring groove, probably due to fatigue failure. The angle-plate cracked at one of its pivot holes, and the opposite pivot pin broke during disassembly. An interesting fact is that even with these failures, the pump continued to function adequately. In conclusion, the ALWT Variable-Displacement Fuel Pump was functioning adequately, except for excessive leakage across the conical valve at high output pressure.

THIRD-GENERATION MODIFICATIONS

The fuel leakage at the conical valve can be reduced by increasing the net forces acting to push the cylinder block towards the fuel pump head. This can be accomplished by decreasing the angle of the conical valve, θ, or decreasing the cross-sectional bearing area of the conical valve seal. It was decided to try decreasing the cross-sectional bearing area of the conical valve seal first, before trying the more expensive method of decreasing the angle of the conical valve. The width of the conical valve seal of the cylinder block was reduced from 0.260 to 0.236 in., effectively reducing the cross-sectional bearing area of the conical valve seal by 10%.

The pistons were redesigned with the O-ring grooves moved towards the top end of the pistons to reduce the possibility of failure due to bending fatigue. The angle-plate was redesigned to strengthen the pivot pin area of the plate. The latest configuration of the pistons and angle-plate is shown in Fig. 38. Because the new angle-plate was shaped to increase the pivot bosses, the detent pin in the cam (Drawing No. NUC 03684) had to be decreased in size, and the notch in the stop (Drawing No. NUC 03692) had to be increased to allow clearance. The only other modification was to strengthen the pivot pins by increasing the 10–32 thread section to 5/16 -24 and rethreading the pump body to accept them.

FOURTH SERIES TEST RESULTS

The VD fuel pump was retested on soluble oil on 8–9 September and on Otto fuel on 20 October 1977. These tests incorporated new pistons, a new angle-plate and pivot pins, and a modified cam, stop and pump body. The new pistons incorporated O-ring grooves close to the head of the piston to eliminate bending failure at the center of the piston. The new angle-plate and pivot pins were designed for increased strength, necessitating modifications to the cam, stop and pump body to provide required clearances.

After initial tests, the cylinder block conical sealing surface area was reduced 10% (conical mating length reduced from 0.260 to 0.236 in.) to effectively increase the sealing force across the porting area of the cylinder block. The object of this test was to show that the proposed reduction in the conical sealing angle (from 45 to 40 deg) is a viable solution in reducing internal leakage between the cylinder ports and the pump head. The advantage in reducing the conical sealing angle over reducing the conical sealing area is that the wear surface is not also reduced. Thus, the cylinder block will have greater life. This series of tests indicated the following:

a. The size of the ball sockets in the pistons is very critical for the correct function of the piston balls against the thrust washer. Only three of the new pistons could be used in these tests because the ball sockets of the others were machined undersize and would not allow the bearing balls to rotate. The three new pistons were substituted in the test series as each of three of the old pistons broke due to bending fatigue.

b. A broken piston is not a catastrophic failure unless the fuel pump exit pressure is greater than 4000 psi.

c. Teflon O-rings on the pistons increase output pressure by approximately 100 psi.

d. Reducing the conical sealing area on the cylinder block by decreasing the conical length of the mating surface proved insignificant, probably because the fluid pressure across the leakage surface is minimal at its extreme edge, where the modification was made. Compare volumetric efficiencies of 74 and 72% in Fig. 39 to those in Fig. 36; they are basically equivalent.

e. Excessive wear was experienced between the I.D. of the piston thrust washer and the O.D. of the angle-plate. The use of grease to increase lubricity between these two parts proved ineffective.

f. The set screw in the worm gear loosened, causing the gear to slip on its control shaft. This occurred during the high-pressure portion of the test.

Figure 39 presents the data from a typical test using the 45-deg conical valve pump head and cylinder block. The data was obtained for cam shaft settings of −7 to +7 instead of the full −12 to +12 range (this represents the full 180-deg rotation of the cam). This was because the worm gear slipped on the control shaft of the piston-stroke control mechanism. The flow data from this run indicates a volumetric efficiency of 74 to 72%, while the pressure rises from 1000 psi to 3700 psi. Figure 40 shows the VD fuel pump theoretical performance for a volumetric efficiency of 100%. Notice that for a flow of 3.2 gpm, the fuel pump operating at 100% volumetric efficiency will produce 4200 psi through a 0.0465-in. orifice. The ALWT requirement specifies 3.2 gpm at 4800 psi output pressure. This indicated the necessity of a smaller diameter orifice to simulate the fuel flow and pressure requirements.

On October 20, 1977 the 45-deg-conical valve VD-fuel pump was run on Otto fuel. The varidrive that drives the fuel pump was started at 60% of maximum ALWT speed and the cam was set at its mid-point to ensure moderate startup flows and pressures. The start-up was smooth and quiet. The varidrive speed was increased to within 94% of maximum ALWT speed and the output pressure was 4500 psi at 2.4 gpm flow when the fuel pump exploded. The explosion occurred at approximately 12 sec after startup. The cylinder block was driven into the pump head, splitting the pump body wide open. The pump body, cylinder block, thrust washer and thrust bearing were damaged beyond repair. Post-run inspection revealed the explosion initiated at the thrust bearing between the thrust washer and the angle-plate, apparently caused by overloading and heating of the thrust bearing.

FOURTH-GENERATION MODIFICATIONS

The fourth series of tests proved that the pistons with O-ring grooves at the top end will not fail in bending fatigue, they should use teflon O-rings, and the ball sockets should be machined to $0.189^{+0.002}_{-0.000}$ in. diameter. It was theorized that reducing the conical sealing angle would improve loading at the seal by redirecting the pressure forces instead of trying to reduce them. Therefore, three cylinder blocks with conical valve angles of θ = 40, 35, and 30 deg were designed and manufactured for the next series of tests. Another modification was to incorporate a bronze journal bearing sleeve in the center of the piston thrust washer to eliminate the wear between the angle-plate and the thrust washer. This modification appears in Fig. 38. The last modification was to incorporate the next larger size thrust bearing between the piston thrust washer and the angle-plate to redistribute and reduce the bearing pressure on the surface of the angle-plate. This would prevent overheating of the thrust bearing and angle-plate, which caused the Otto fuel explosion. To prevent the worm gear from slipping on the control shaft, Lock-Tite will be used on the set screw of the worm.

In order to more accurately simulate the predicted fuel flow and pressure requirement for the ALWT open cycle engine, a 0.038-in.-diameter orifice was selected for the next series of tests. The 0.0465-in.-diameter orifice would continue to be used when comparing the effects of changing the angle of the conical valve.

Due to the distruction of the hardware, a second pump body was modified to accept the larger pivot pins of the angle-plate and the 40- and 35-deg cylinder blocks and pump heads were used for the next series of tests. The next series of tests would also incorporate thermocouple instrumentation at various places within the fuel pump so that any overheating could be avoided.

FIFTH SERIES TEST RESULTS

The fifth series of tests was run to investigate fuel pump performance as a function of conical valve angle. According to theory, the net force pushing the cylinder block towards the fuel pump head should increase, and the leakage at their interface should decrease as the conical valve angle, θ, decreases. Fuel pump output flow and pressure will increase as the leakage across the conical valve decreases. The penalty for decreasing the leakage across the conical valve is that the bearing pressure and, therefore, the wear at the conical valve increases. The relationships between net bearing force, bearing pressure, fuel pump output pressure, and conical angle were shown in Figs. 22 and 23. The object of these tests was to verify the theoretical prediction that the 40-deg conical valve is the optimum angle, resulting in a good balance between leakage and wear at the valve as the VD fuel pump meets the ALWT performance requirements.

This test series started on 4 November 1977 and ended on 1 December 1977. The tests incorporated all the modifications through and including the forth series: pump heads with larger porting and improved timing, cylinder blocks with larger porting and chamfered aft edge, ball-and-socket pistons with forward O-ring grooves, stronger piston thrust washer with bronze journal bearing, steep-pitched angle-plate with large thrust bearing, and large pivot pins, pump body modified to accept large pivot pins, and stiffer piston springs.

Figures 41 through 46 and 50 and 51 present the fuel pump performance for this series of tests. Figure 41 indicates a slight improvement in output pressure and flow for the 40-deg conical valve design over the 45-deg conical valve design. (Compare with Fig. 39.) There appeared to be a little more fluctuation in the output pressure and flow at the higher cam settings for the 40-deg conical valve design than the 45-deg conical valve design. There was also more wear on the 40-deg conical surface of the cylinder block, but acceptable to warrant a test with actual Otto fuel. Before this test, an energy analysis was accomplished. This is shown in Fig. 42. This data indicates an increase in efficiency with cam setting and therefore increase in flow and output pressure. Although the maximum system efficiency is only 48%, this is approximately 16% greater than the system efficiency for a fixed-displacement fuel pump biased to the equivalent flow and pressure output. Performance for such an equivalent fixed-displacement fuel pump is shown in Fig. 43, ALWT backup fuel pump No. 4. Notice that the data is basically the same for pumping either soluble oil or Otto fuel. The important advantage of the VD fuel pump is obvious at the lower flow and pressure requirements. The horsepower required into the VD fuel pump drops off exponentially, while that for the equivalent fixed-displacement fuel pump drops off linearly, the net result being a two-thirds savings in energy required to drive the VD fuel pump at the lower torpedo speed.

The test results for the 35-deg conical valve design of the VD fuel pump are presented in Fig. 44. Two sets of data are shown, the lower flow and pressure curves occuring first. By the end of this test sequence the volumetric efficiency had improved from an average 69 to 88% and the output pressure from 1200 to 2200 psi. The disappointing factor was excessive wear at the 35-deg conical valve, especially of the bronze surface of the cylinder block. In fact, the wear became so excessive that large fluctuations in flow and output pressure, together with audible noise and visual vibration of the hardware and test stand, precluded any further testing. Decreasing the angle of the conical valve from 40 to 35 deg conclusively verified the predicted relationships between leakage and wear at the conical valve. The 35-deg conical valve design had increased the forces tending to push the cylinder block against the pump head, causing more wear and a better seal at the valve and, therefore, reducing the amount of leakage across the valve. Figure 45 shows that there was about a 5% reduction in leakage for the 40-deg conical valve over the 45-deg conical valve, but a 25% fluctuation in percent leakage for the 35-deg conical valve, this being due to the excessive wear. Therefore, for the materials incorporated (tool steel pump head and bronze cylinder block comprising the male and female portions of the conical valve, respectively) the 40-deg conical valve design is optimum.

An attempt was then made to improve the sealing of the 40-deg conical valve by increasing pump output pressure using a 0.038-in.-diameter orifice. Figure 46 presents the results of this attempt. The conical valve suffered damage at the edge of its ports due to ingestion of a metal fragment. Evidently, the metal fragment was dislodged from the test stand plumbing left over from previous and different test hardware. The damage incurred is shown in Fig. 47, and the machining rework is shown in Figs. 48 and 49. The rework shown in Fig. 48 is at the opposite edge of the nicked intake slot shown in Fig. 49. Using the 0.038-in. orifice increased the output pressure as expected and also improved the sealing and volumetric efficiency of the 40-deg conical valve when the 0.0465-in. orifice was again used. This is apparent from the 94 to 75% volumetric efficiency performance of the reworked fuel pump shown in Fig. 50. The original volumetric efficiency was from 76 to 70% (see Fig. 41).

The results of the reworked 40-deg conical valve design were good enough that the pump was run on Otto fuel on December 1, 1977. The results are shown as an overlay to the soluble oil run made the same day (Fig. 51, showing the Otto fuel results over the soluble oil results that appeared in Fig. 50). This test indicated a reduction in pump performance when pumping Otto fuel: an approximate 13% reduction in volumetric efficiency and 27% reduction in putput pressure. The total run time was 5 min 43 sec. Pump startup and running temperatures were safe, and the pump seemed to run quietly and without vibration. The cam setting could not be increased beyond its mid-point because the worm gear set screw had again slipped. Post-run inspection revealed excessive wear of the conical valve area of the cylinder block and radial cracks on the hardened steel surface of the male portion of the conical valve. Wear was so extensive that bronze particles were found throughout the pump. Evidently, Otto fuel is a poor lubricant at the pressures induced at the 40-deg conical valve surfaces.

The overall conclusion of this test series was that the 40-deg conical valve appeared to function better than the 45- and 35-deg conical valve angles but was still slightly short of meeting the 4800 psi at 3.2 gpm ALWT high-torpedo-speed requirement without pressure and flow fluctuations. These fluctuations were either causing or being caused by wear at the conical valve interface. The VD fuel pump is approximately 20% more efficient than an equivalently biased fixed-displacement fuel pump. It was also discovered that the leakage across the 40-deg conical valve could be decreased by temporarily increasing the output pressure of the pump by using a smaller orifice (0.038 in. diameter). The 5-min 43-sec Otto fuel run proved the feasibility of the VD fuel pump design, but showed the fuel to be a poor lubricant. The final conclusion was that the 35-deg conical valve could be made to function at high volumetric efficiency (88%) if better wear-resistant materials could be found and utilized and if the output pressure and flow fluctuations could be eliminated.

FIFTH-GENERATION MODIFICATIONS

Based on the results of this test series three modifications were developed: (1) incorporate a large radial needle bearing to support the cylinder block in order to eliminate the pump's output pressure and flow fluctuations; (2) increase the bearing surface of the conical valve without increasing its sealing area; (3) redesign the pump head to strengthen the area between the fuel entrance ports and the intake slot of the conical valve to prevent cracking; and (4) screw the set screw of the worm gear into its shaft to prevent slippage. The fuel pump body was machined to accept a B-328 torrington radial needle bearing at the aft edge of the cylinder block. The 40-, 35-, and original 30-deg conical angle cylinder blocks were each modified with a tool steel band to act as the inner race for the supporting bearing. A bronze insert was pressed into the 40-deg conical angle cylinder block and then machined to continue the same 40-deg angle, doubling the bearing surface of the conical valve. A circumferential groove and four radial grooves were then machined into this surface such that the original net force pushing the cylinder block against the pump head would be maintained. A new pump head was designed to accommodate a larger bearing surface at the conical valve and strengthen the entrance port area. This new pump head was heat-treated to Rockwell hardness 55 (instead of the previous 62) to alleviate surface embrittlement and reduce the tendency of cracking on the conical valve surface. The formerly flat, aft surface of the pump head was machined in a dish-like fashion towards the base of the conical valve protrusion so that the four entrance ports could be spaced farther away from the intake slot. (These modifications are shown in Fig. 53.)

The 35-deg conical angle cylinder block and pump head were remachined to produce a valve in the form of a 0.7500-in. spherical radius. It was hoped that this spherical-radius valve would compensate for any inaccuracies in concentricity or alignment that may have caused the fluctuations experienced with the conical valve designs.

A program was initiated in December 1977 to test various candidate materials for the piston block and pump head. A test fixture was designed and manufactured to simulate the actual fuel pump environment, bearing pressures, and surface speeds at the conical valve. Test pieces made of various materials were dimensioned the same as the block and pump head at their valve interfaces. A summary of this program appears in the Appendix.

SIXTH SERIES TEST RESULTS

The sixth series of VD fuel pump tests began on 28 March 1978 and ended on 5 May 1978. These tests incorporated a large radial needle bearing to help support and stabilize the cylinder block and eliminate the output pressure and flow fluctuations previously experienced. These tests also include the set screw of the worm gear being screwed into its control shaft to prevent slippage. Three different valve designs (composed of a matching cylinder block and pump head) were tried in this series: (1) a 35-deg-spherical-radius valve, (2) an original 30-deg conical valve, and (3) a 40-deg conical valve with increased bearing surfaces at the valve area.

While running each of the three valve designs, the following observations and conclusions were made:

For the 35-deg spherical radius valve: (1) there was no significant improvement in pump performance over the 35-deg conical valve design; (2) there was a reduction in vibration and wear at the valve surfaces when the large radial needle bearing was installed to support the cylinder block; (3) the wear on the valve surfaces was more evenly distributed when a thin metal shim was installed between the cylinder block and its drive shaft, reducing the diametral clearance from a nominal 0.0063 in. to 0.0003 in. The final conclusion was that the spherical-radius valve does not compensate for any inaccuracies in concentricity or alignment, and it is better to support and stabilize the cylinder block instead of allowing it to "float" and seek its own center upon the pump head.

For the 30-deg conical valve: (1) the 30-deg conical angle cylinder block showed even wear, appearing to be better centered on the pump head than the 35-deg conical angle cylinder block; (2) there was less pressure and flow fluctuation than with the 35-deg conical valve design; (3) output pressures reached 5000 psi at 2.4 gpm flow with a volumetric efficiency of 76%, but the volumetric efficiency dropped off to 59% at the low output pressures. The data is plotted in Fig. 52, and the pump head and cylinder block are shown in Fig. 53. Notice that the hardware is shown in the pre-run condition, before the bearing race modification to the cylinder block; (4) the wear was reflected in the large amount of torque (66 ft-lb) required to rotate the pump's driveshaft at the higher output pressures. See table 7 for the actual data. The final conclusion was that the 30-deg conical valve design was not satisfactory from either a performance or wear point of view.

For the 40-deg conical valve: (1) there were good volumetric efficiencies, 77 to 76%, and the maximum pressure requirement was reached, 5000 psi. See Fig. 54; (2) for the first time there was acceptable wear on the conical valve area of the bronze cylinder block and pump head. This conical valve design incorporated twice the bearing area of previous designs. The cylinder block and pump head are shown after 4 min of running in Figs. 9 and 10; (3) there were still pressure and flow fluctuations, especially at the higher output pressures. As can be seen in the original data, Table 8, there was a 200-psi pressure fluctuation at the −7.2 cam setting (40 deg cam rotation) and a 145-psi pressure fluctuation at the +0.3 cam setting (117 deg cam rotation). The final conclusion was that the increased bearing area of the conical valve is necessary to prevent excess wear of the valve surfaces.

RECOMMENDED FINAL MODIFICATIONS

In view of the last series of tests on the VD fuel pump, several recommendations have been proposed in a pump redesign program. Such a pump redesign would include the large radial needle bearing for stabilizing the cylinder block without restricting the fuel flow around the block, and reduced clearance between the pump shaft and the forward end of the bore in the block in order to prevent the cylinder block from "floating" on the pump head. This redesign program would determine the cause of the pressure and flow fluctuations by using more sophisticated instrumentation, such as differential pressure transducers across the fuel pump. Other modifications could include slightly larger diameter pistons or the addition of another piston if volumetric efficiencies could not be increased, thereby creating greater output pressures and flows at lower cam settings. Higher output pressures at the lower cam settings may improve cylinder block stability and reduce the pressure and flow fluctuations previously experienced.

TABLE 1. VD FUEL PUMP DESIGN EQUATION

$$Q = K\,S\,N\left(\frac{\pi}{4}d^2\right)\left(D\,\mathrm{TAN}\,\phi\right)E$$

where:

Q = Volumetric Flow Rate
K = Dimensional Constant
S = Speed of Fuel Pump Shaft
N = Number of Pistons
d = Diameter of Pistons
D = Diameter of Piston Bolt Circle
ϕ = Pitch of Angle-Plate
E = Volumetric Efficiency

and

ϕ = (Angle on Angle-Plate) - (Angle on Cam) (cosine of cam rotation)
$\phi = 10.5^\circ - 4.5^\circ \mathrm{Cos}\,\alpha$

where:

α = Cam Rotation (0 to 180°)

TABLE 2. PRELIMINARY DATA: ENERGY ANALYSIS OF ACTUATOR MOTOR FOR VD FUEL PUMP

Item	No Instrumentation Or Gear	With Instrumentation And Gear	Full Assembly While Running
<u>At Motor</u>			
volts	13.1	13.1	13.1
amperes	0.160	0.180	0.250
watts	2.10	2.36	3.28
horsepower	0.0028	0.0032	0.0044
<u>At Worm Shaft</u>			
Number of turns of worm gear shaft			25.25 turns
Time to rotate cam 180°			11 sec
RPM of worm shaft			137.73 RPM
<u>At Motor</u>			
Motor gear ratio	18.78:1	18.78:1	18.78:1
RPM of motor			2586.5
Motor torque, in.-lb	0.0682	0.0779	0.1072
<u>At Worm Shaft</u>			
Shaft torque, in.-oz	20.46	23.37	32.16

horsepower = (1.341×10^{-3}) watts

torque $= \dfrac{63{,}00\ (hp)}{N\ (RPM)}$

TABLE 3. DIAGNOSTIC TESTS (16 MARCH 1977)

Input Pressure, P_{f_i}, psi	Output Pressure, P_{f_o}, psi	Flow, ω_f, gkm		Varidrive RPM	
				Mid Cam angle	
30	80	1.56	1	1500	Slow Speed
50	100	1.52		1500	
80	100	1.45		1500	
				Mid Cam angle	
30	145	2.05		1963	High Speed
50	145	2.05	1	1963	
78	170	1.98		1963	
				Max Cam angle	
20	150	2.83	2	1963	High Speed
30	200	3.32		1963	
50	225	*3.50	1	1963	
65	225	3.48		1963	
				Max Cam angle	
20	145	2.66	2	1500	Low Speed
30	150	2.70		1500	
50	175	2.70	1	1500	
80	200	2.68		1500	

CONCLUSIONS:

1 Flow reduction probably due to valving overlap (entrance and exit simultaneously open at TDC and BDC).

2 Flow increase probably due to starvation caused by restriction at inlet to cylinders. As the inlet pressure increased, the flow increased.

*Volumetric efficiency $= \frac{3.50}{3.8} \times 100\% = 92\%$

TABLE 4. RUN-IN TEST PERFORMANCE VS TIME (12 JULY 1977)

Duration, min, sec	Flow, ωf_o, gkm	Output Pressure, P_{f_o}, psi	Torque, ft-lb
0, 0	2.6	2325	22.0
0, 15	2.59	2360	20.2
0, 30	2.59	2370	20.0
0, 45	2.6	2300	20.0
1, 0	2.9	2930	34.5
1, 15	2.88	2850	33.0
1, 45	2.8	2780	31.5
2, 0	2.8	2725	31.0
2, 15	2.8	2700	30.7
2, 30	2.45	2060	17.5
2, 45	2.45	2050	17.1
3, 30	2.49	2080	17.1
4, 35	1.44	725	5.9
5, 15	2.5	2075	17.6
6, 0	2.74	2650	31.0

Test Conditions

Teflon O-rings

Polished cylinder block

Polished head slots

One new piston bearing ball (to replace lost original)

0.052-in. orifice

Varidrive RPM: 1963

TABLE 5. RUN-IN VS OUTPUT PRESSURE (12 JULY 1977)

Orifice Size, in.	Flow, gkm	Output Pressure, P_{f_o}, psi	Torque, T_ϕ, ft lb	Volumetric Efficiency, %
0.0465	3.2	4200		
0.0465	3.2	4200	40+	
no orifice	4.3	≈100	3.6	$\frac{4.3}{4.76} \times 100 = 90$
0.038	1.95	4900	40+	
0.038	2.00	5000	40+	
0.038	1.68	3550	27.5	
0.038	2.00	5000	40+	
0.038	2.00	5100	40+	
no orifice	4.5	≈100	3.5	$\frac{4.5}{4.76} = 95$
0.0465	3.2	4200	40+	

TABLE 6. 45-DEG CONICAL VALVE TEST DATA, INTERNAL LEAKAGE VS OUTPUT PRESSURE (12 JULY 1977)

Orifice Size, in.	Flow, ω_f, gkm	Theoretical Flow, ω_f,* gkm	Output Pressure, P_{f_o}, psi	Cam Setting volts	Torque, T_ϕ, ft-lb	η_L,** %
0.038	2.0	4.3	5270	+0.611	40+	53
0.0465	2.99	4.3	3510	+0.611	38.8	30
0.052	3.1	4.3	3075	+0.611	33.0	28
						η_V†
no orifice	4.3	4.3	100	+0.611	3.0	99
no orifice	4.4	4.76	105	+1.250	3.2	92
no orifice	4.3	4.52	100	+0.800	3.0	95
no orifice	3.2	3.29	75	-0.005	1.9	97
no orifice	1.9	2.09	75	-0.799	1.3	91
no orifice	1.8	1.86	75	-1.249	1.1	96

*ω_f theory = 17.7514 tan θ

**η_L = percent leakage across seal: $\frac{4.3 - \omega_{fo}}{4.3} \times 100\% =$

†η_V = volumetric efficiency, no orifice

TABLE 7. ACTUAL DATA – VD FUEL PUMP 30-DEG CONICAL VALVE DESIGN; 30-DEG CYLINDER BLOCK AND PUMP HEAD, 0.043-IN. ORIFICE (28 MARCH 1978)

Cam Setting, volts	Inlet Pressure, P_{f_i}, psi	Outlet Pressure, P_{f_o}, psi	Flow Rate, ω_f, gkm	Torque To Pump Shaft, T_ϕ, ft-lb
-11.5	50	850	1.1	7.0
- 8.8	47	1350	1.3	9.0
- 3.5	45	2050	1.6	18.3
- 1.5	37	4400	2.3	48.0
+ 0.4	36	5000	2.4	66.0

TABLE 8. ACTUAL DATA – VD FUEL PUMP 40-DEG CONICAL VALVE DESIGN; 40-DEG CYLINDER BLOCK WITH INCREASED BEARING SURFACES AT VALVE AREA; 40-DEG PUMP HEAD (5 MAY 1978)

Cam Setting, volts	Inlet Pressure, P_{f_i}, psi	Outlet Pressure, P_{f_o}, psi	Flow Rate, ω_f, gpm	Torque To Pump Shaft, T_ϕ, ft-lb
-7.1	52.0	2275	1.70	24.0
-7.2	52.0	2275	1.72	16.0
-7.2	54.0	2075	1.66	14.9
+1.5	43.0	4850	2.48	45.0
+0.3	39.0	4855	2.48	48.5
+0.3	37.0	5000	2.52	46.5

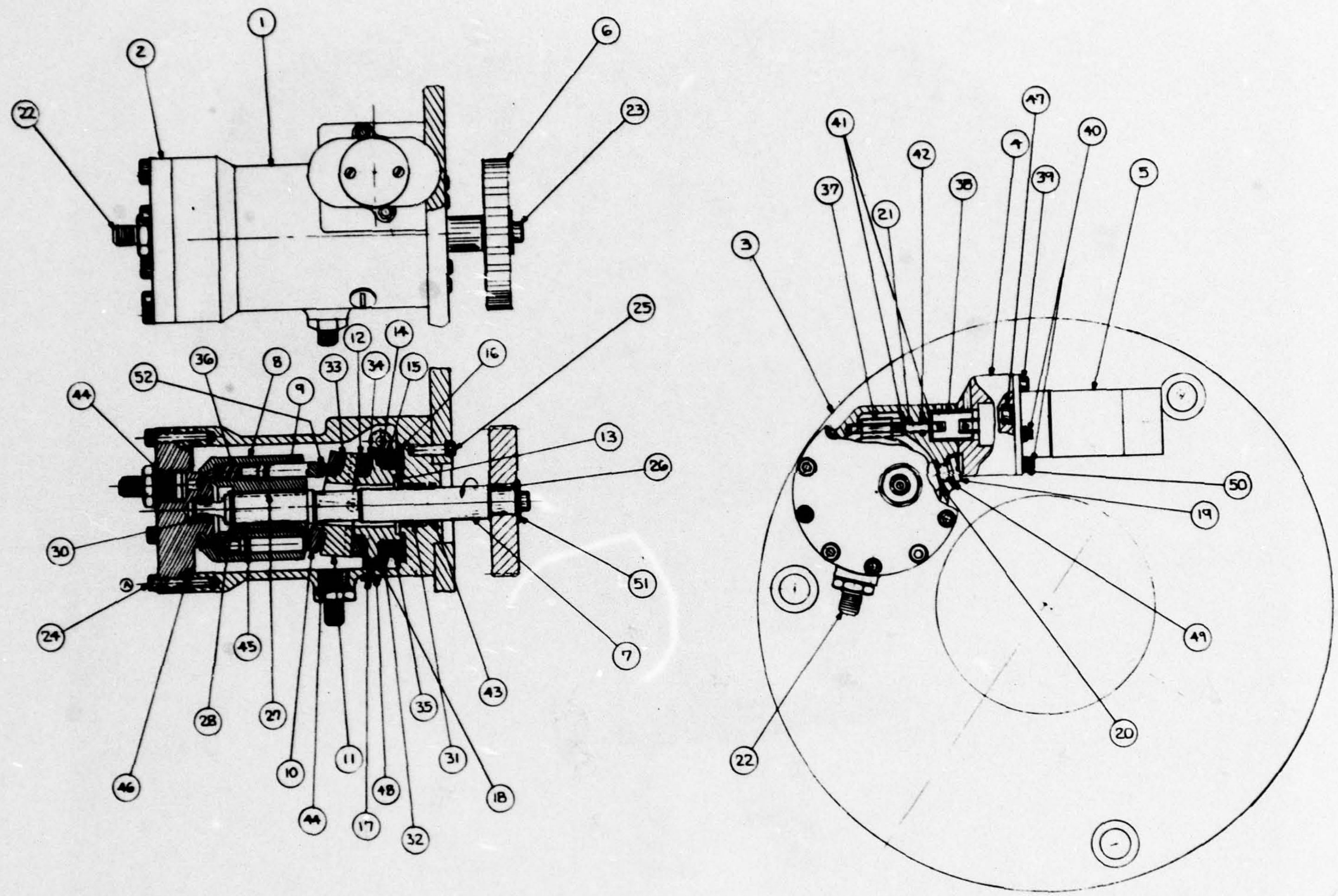


Figure 1. Assembly, VD fuel pump.

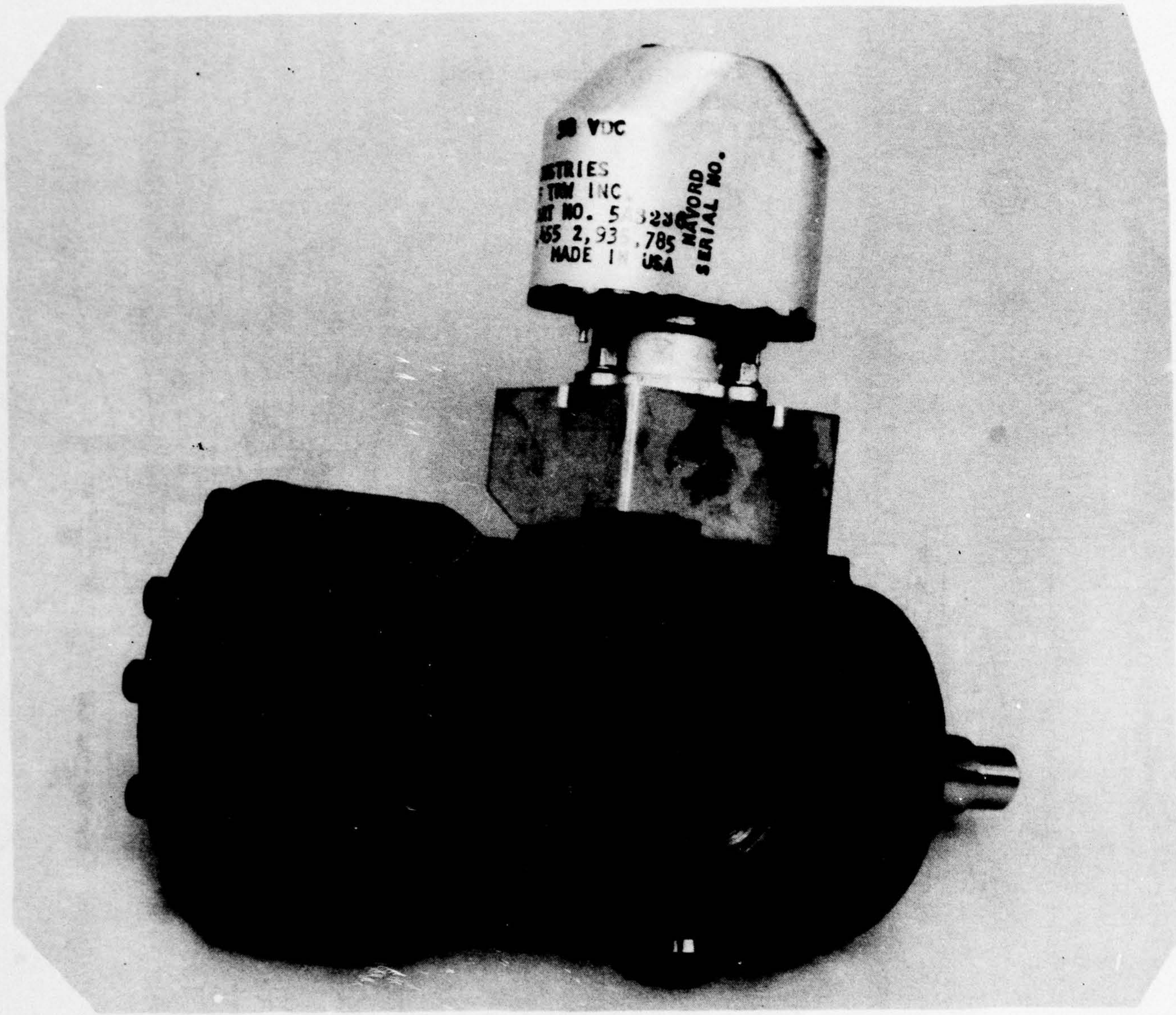


Figure 2. Assembled VD fuel pump.

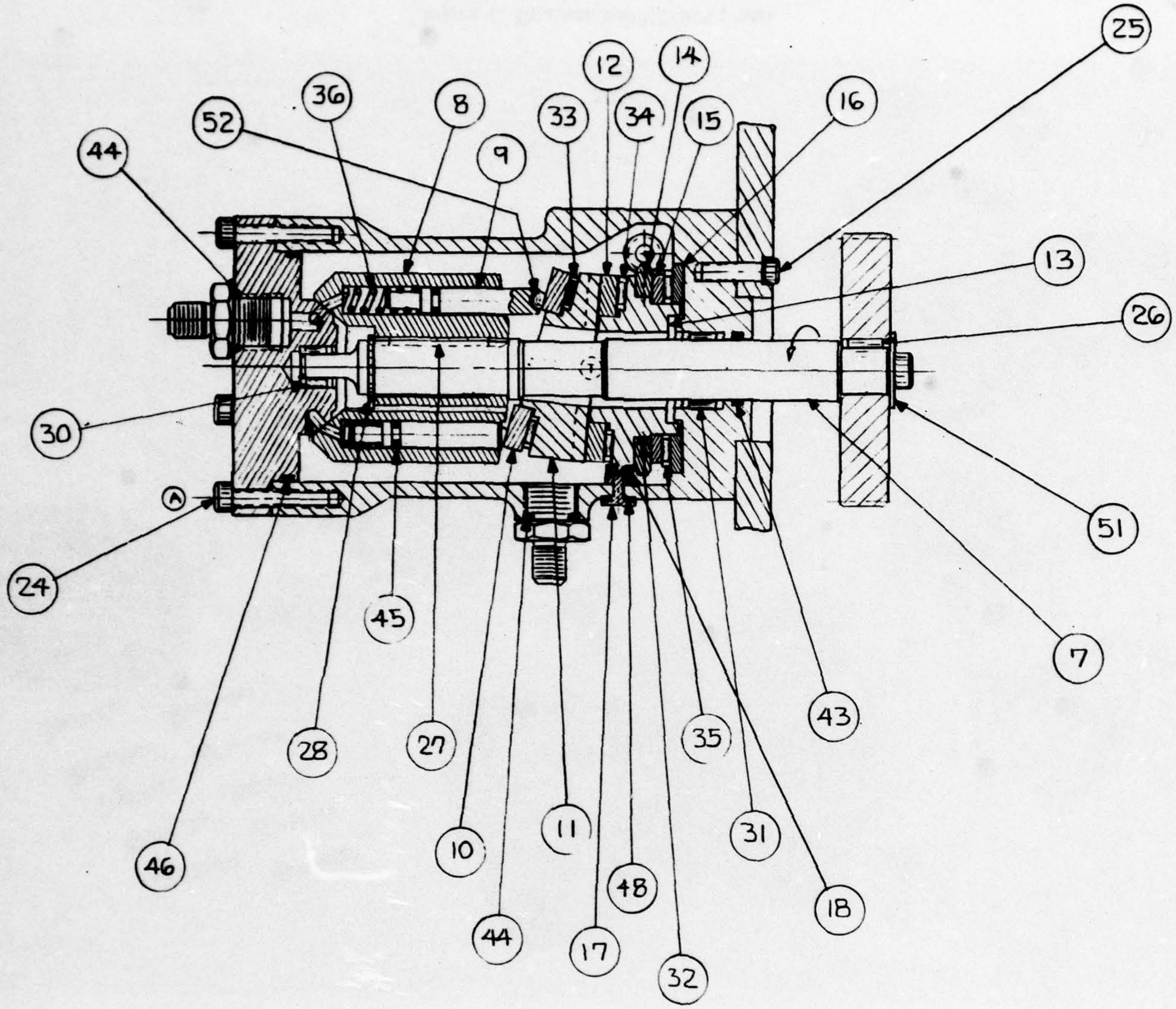


Figure 3. Assembly, VD fuel pump.

Figure 4. Disassembled VD fuel pump.

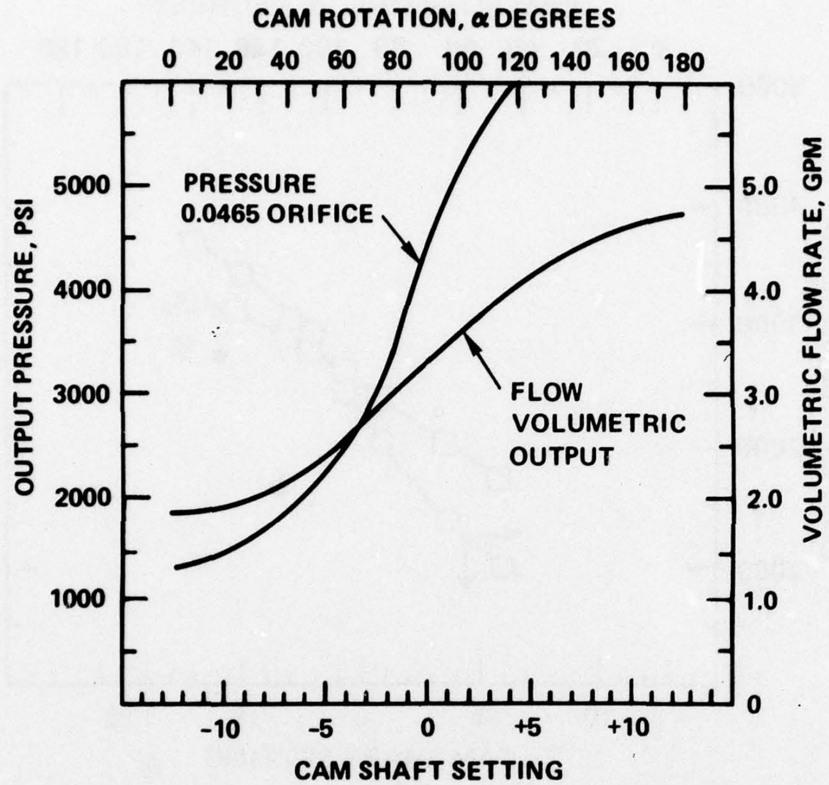


Figure 5. VD fuel pump performance (theoretical 100% efficiency).

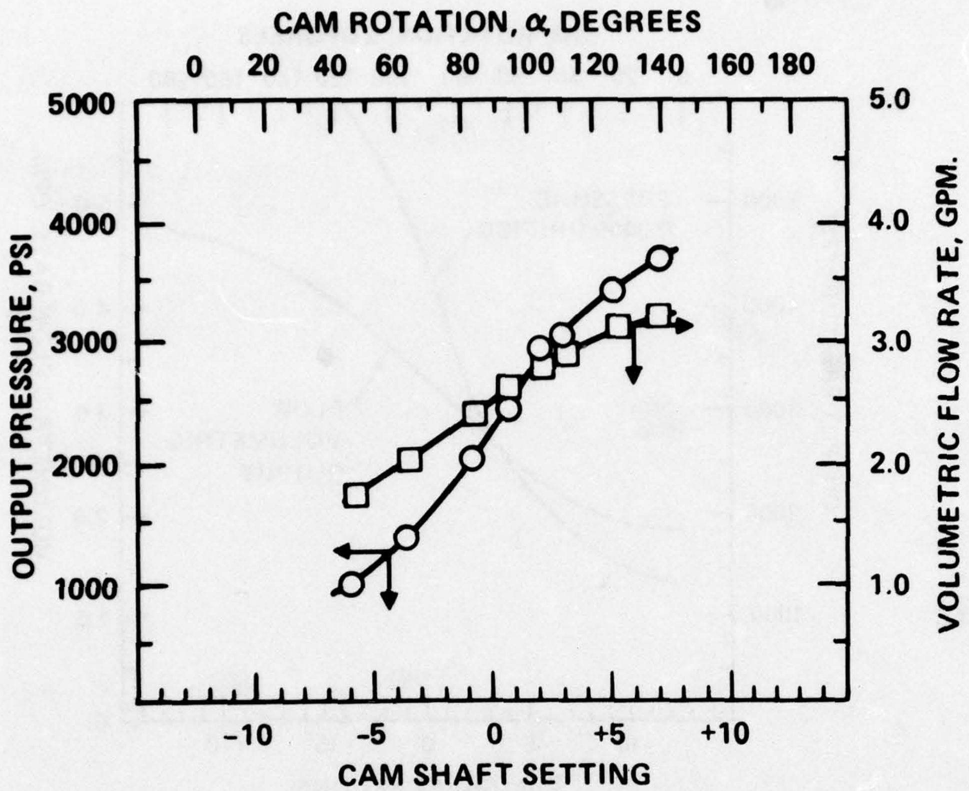


Figure 6. VD fuel pump performance (40-deg conical valve, 0.0465-in. orifice, C = 0.79).

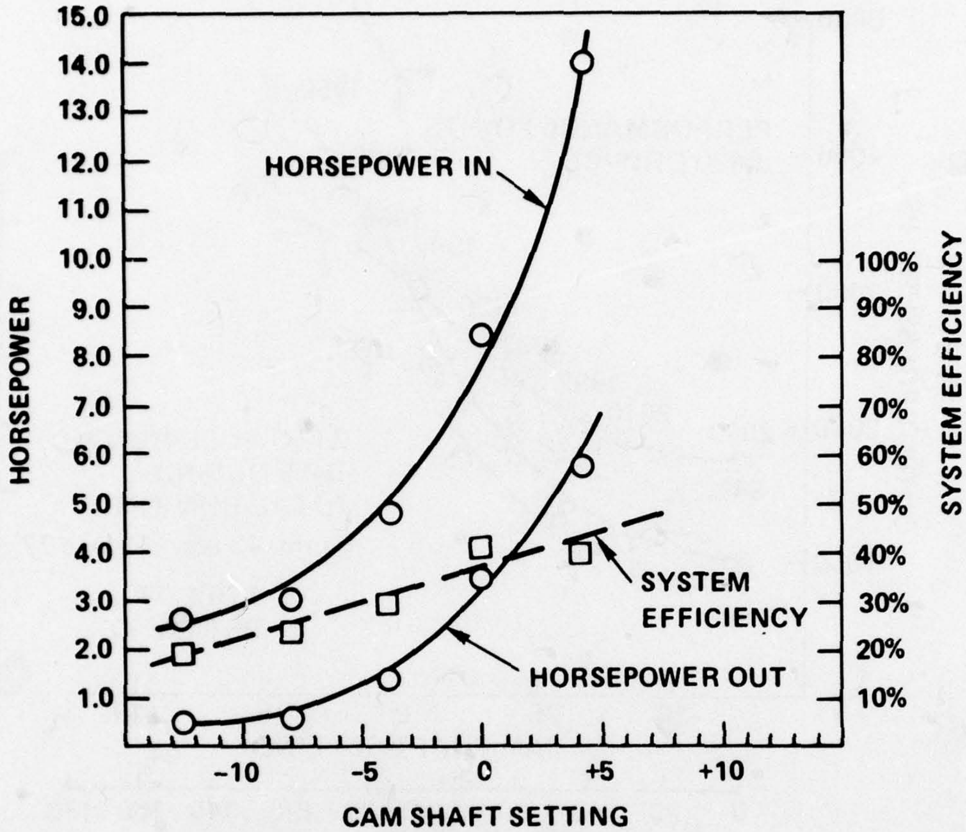


Figure 7. VD fuel pump energy analysis (40-deg conical valve, 0.0465-in. orifice).

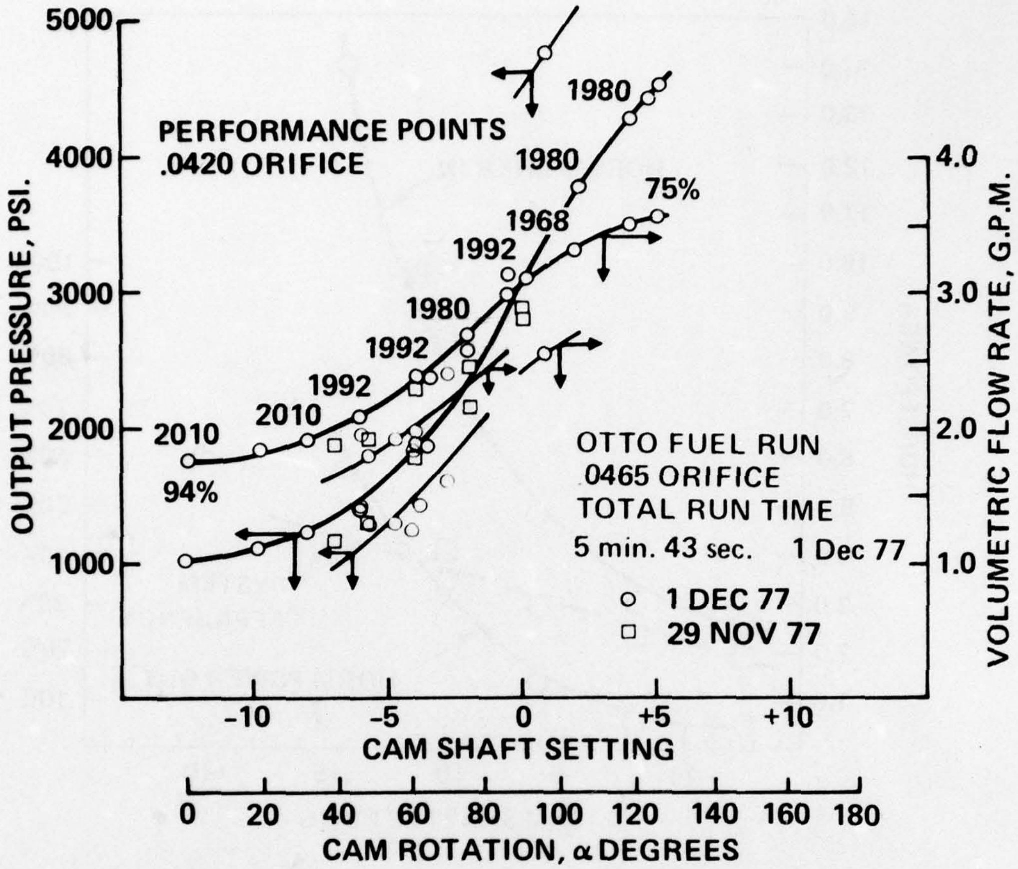


Figure 8. VD fuel pump (reworked) performance (40-deg conical valve, 0.0465-in. orifice).

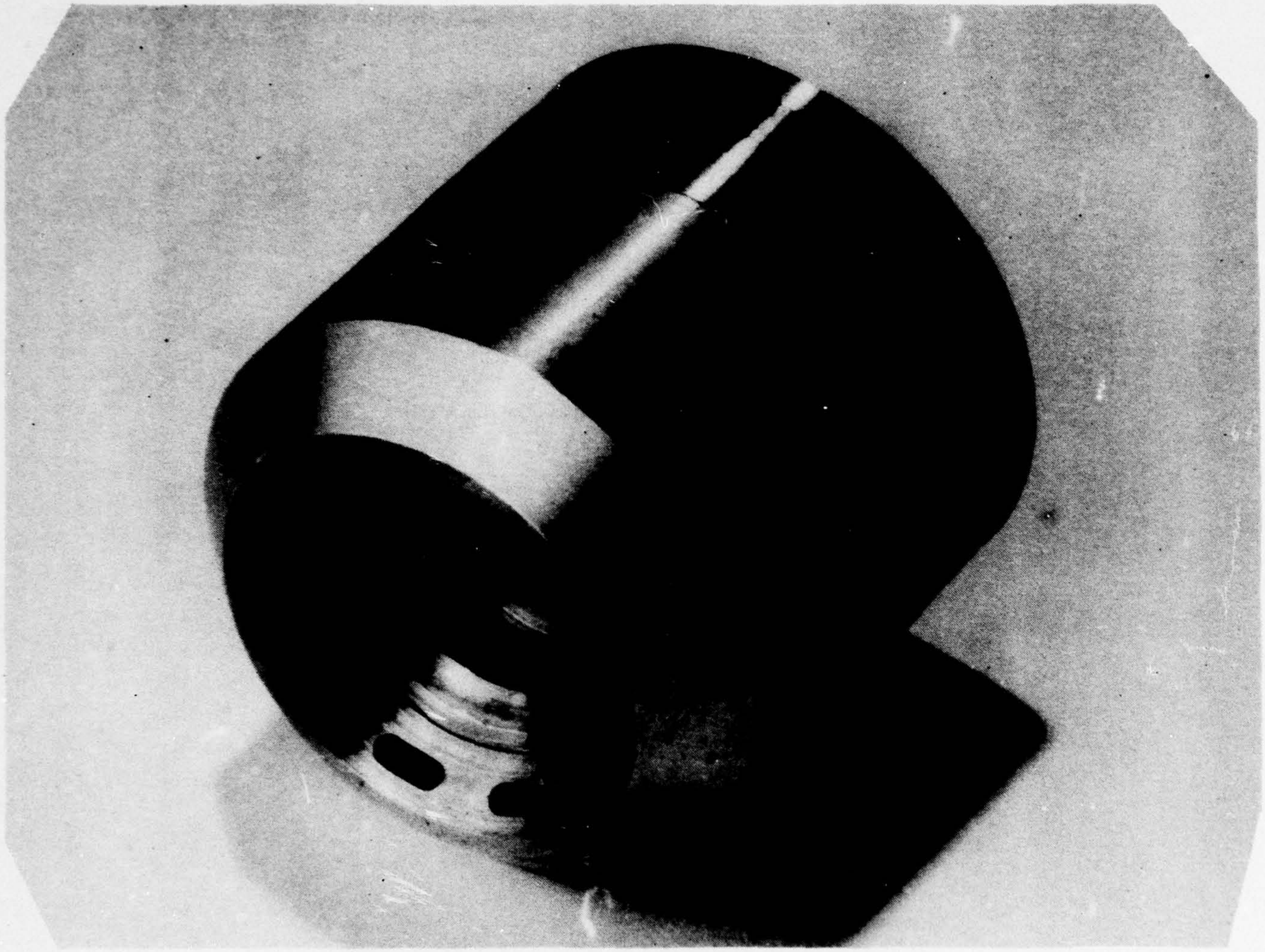


Figure 9. Cylinder block with radial needle bearing race and enlarged conical bearing surface.

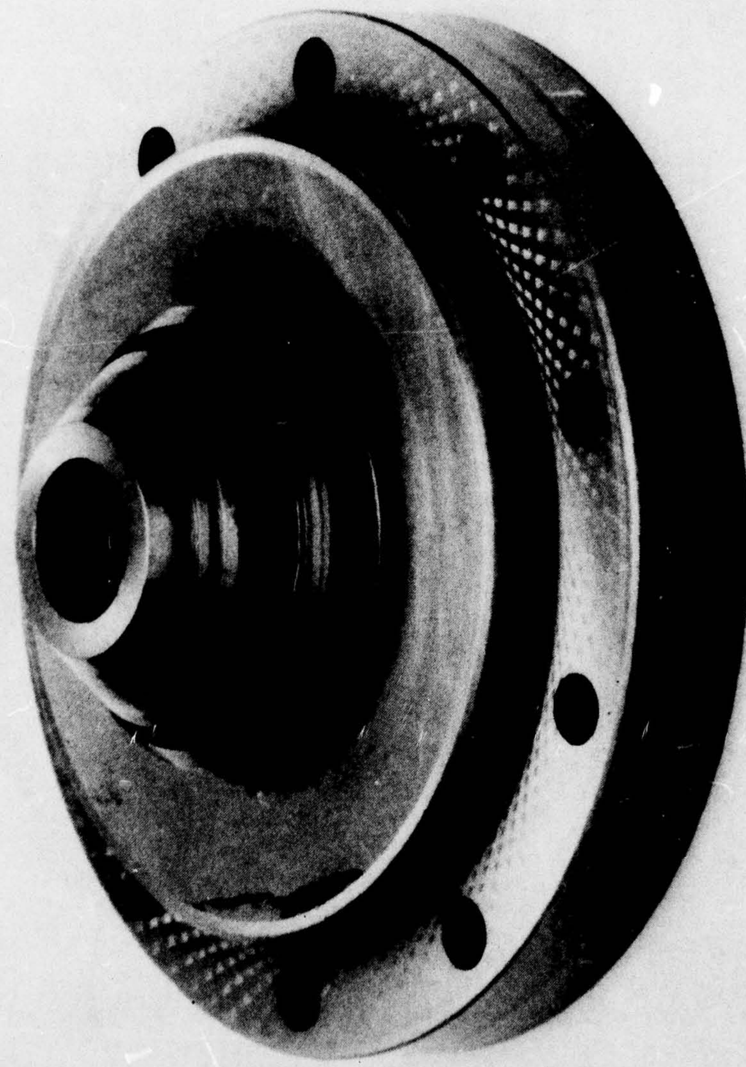


Figure 10. Pump head with enlarged conical bearing surface

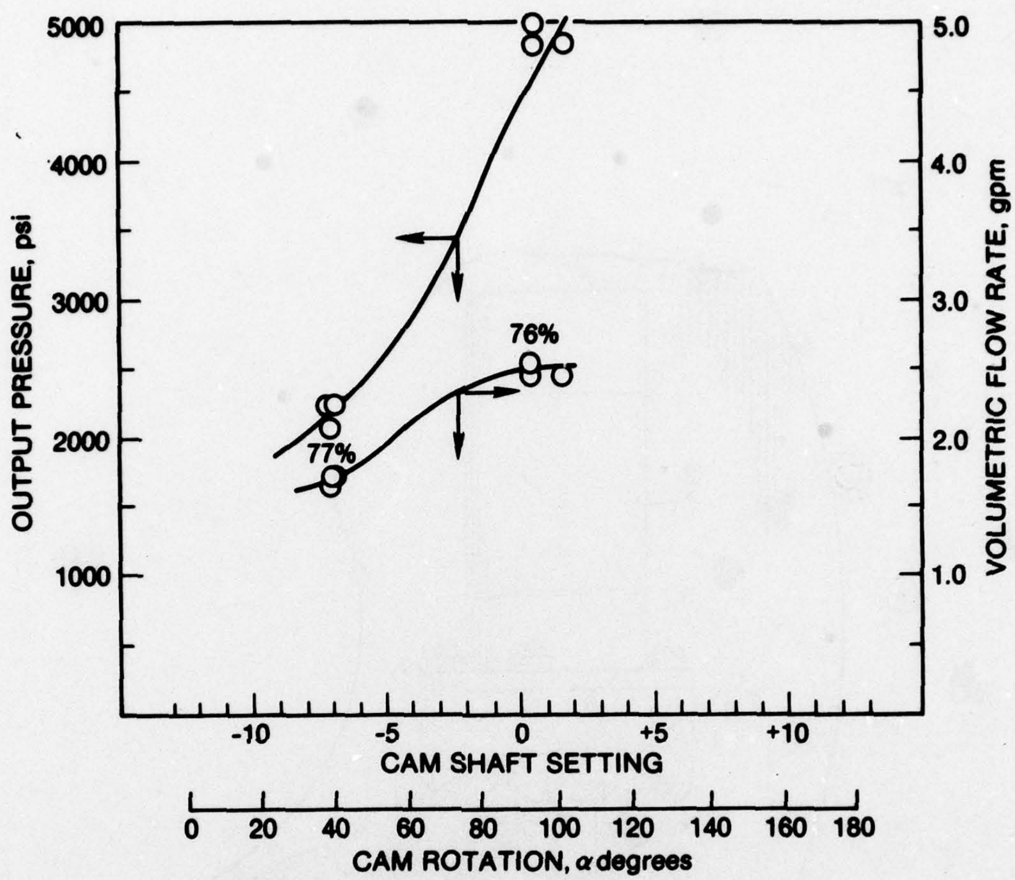


Figure 11. VD fuel pump performance (40-deg conical valve, 0.043-in. orifice), 5 May 1978.

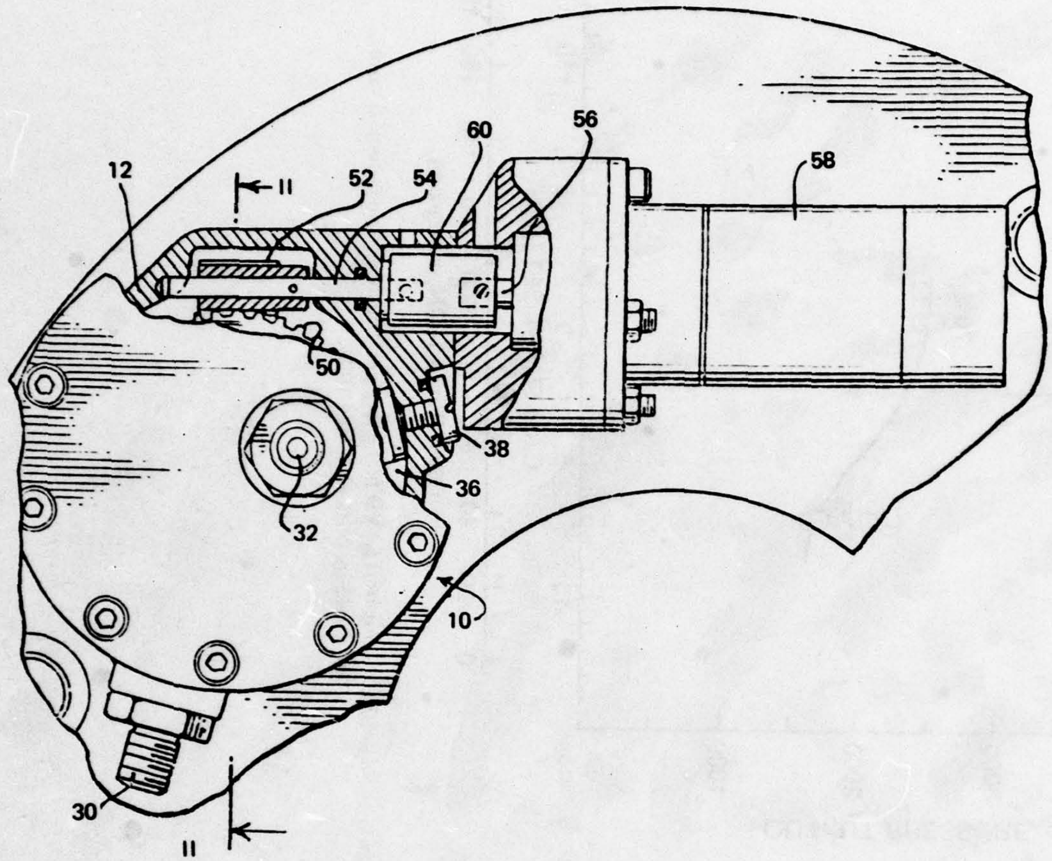


Figure 12. Piston-stroke control mechanism, top view.

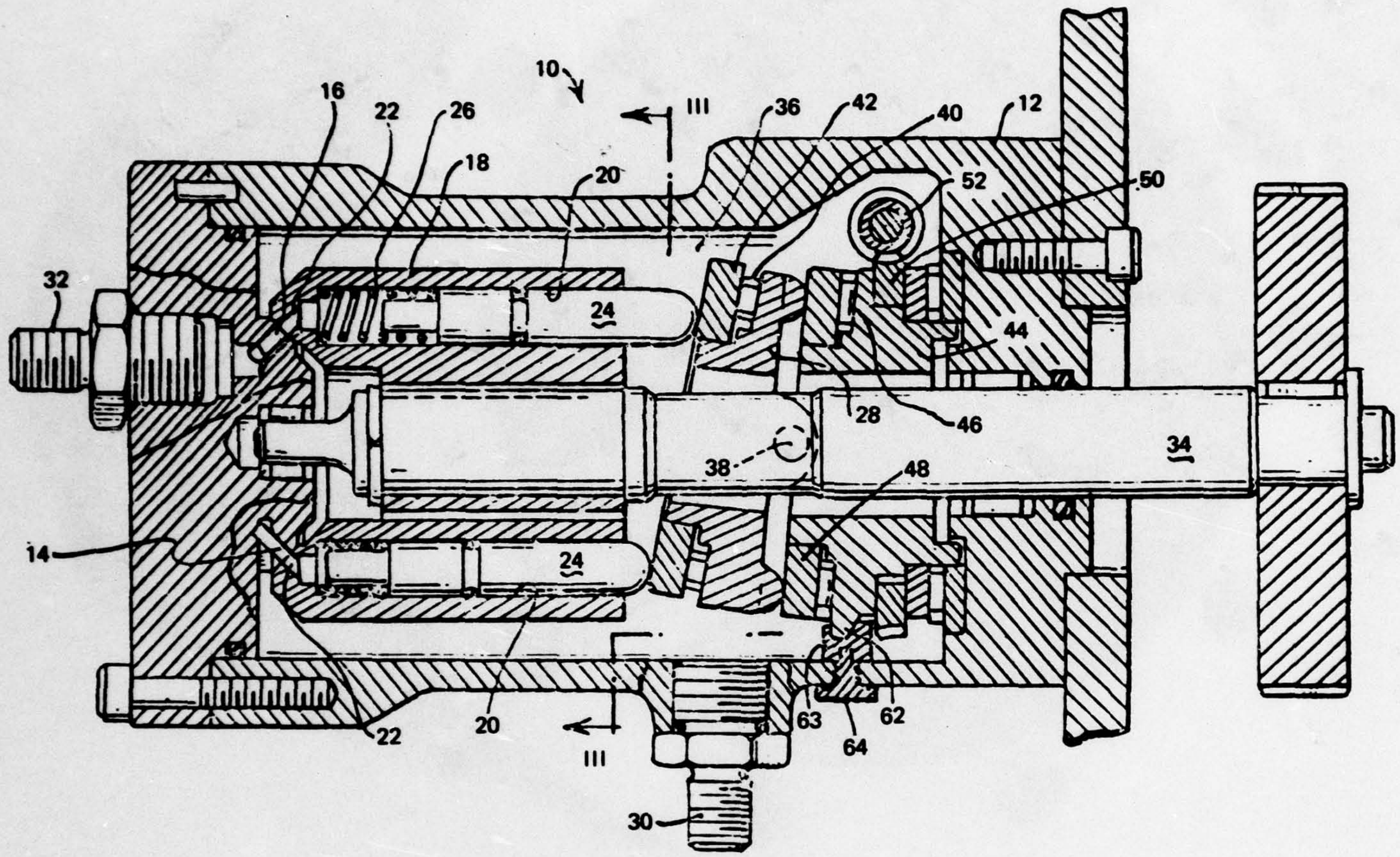


Figure 13. Cross-sectional-view of VD fuel pump.

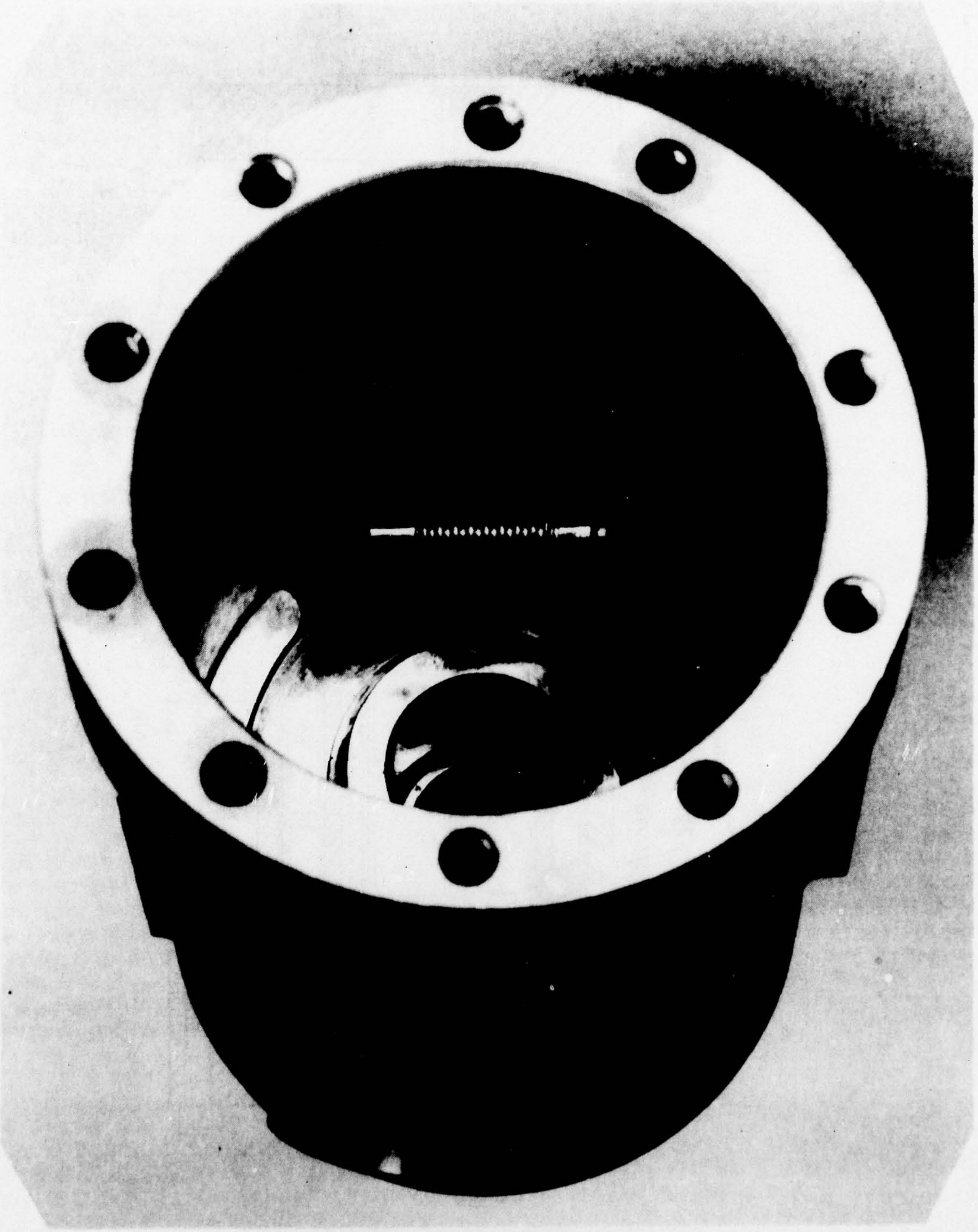


Figure 14. Meshing control gears assembled in pump body.

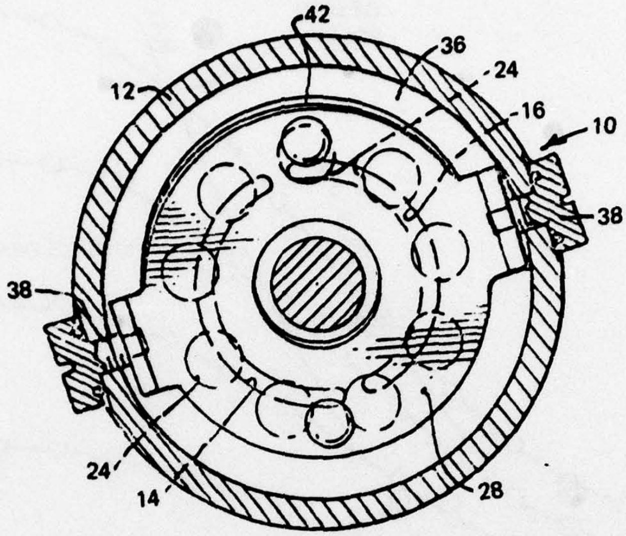


Figure 15. View of angle-plate and pivot screws.

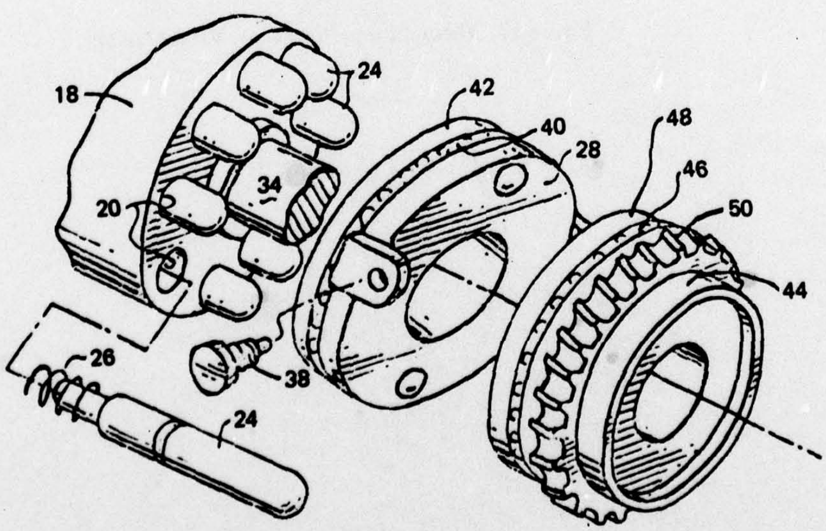


Figure 16. Exploded view of piston-stroke control mechanism.

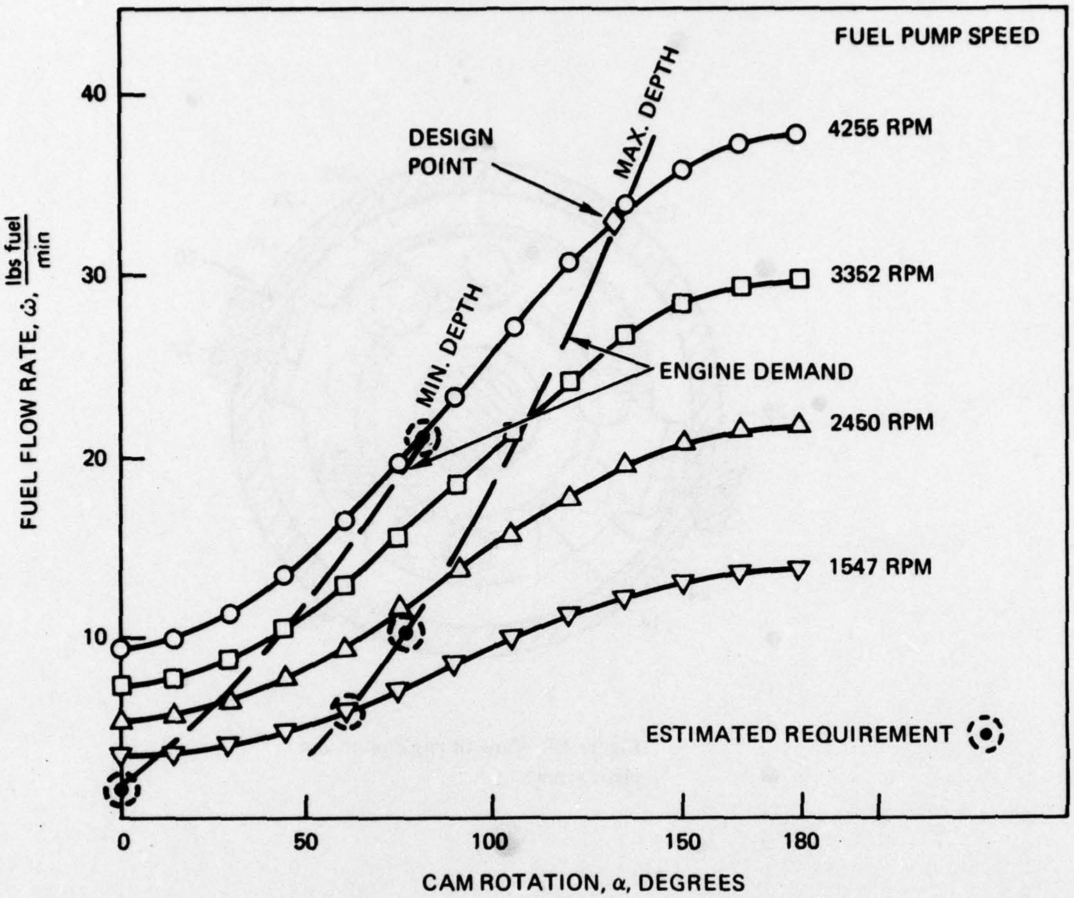


Figure 17. Theoretical performance, VD fuel pump.

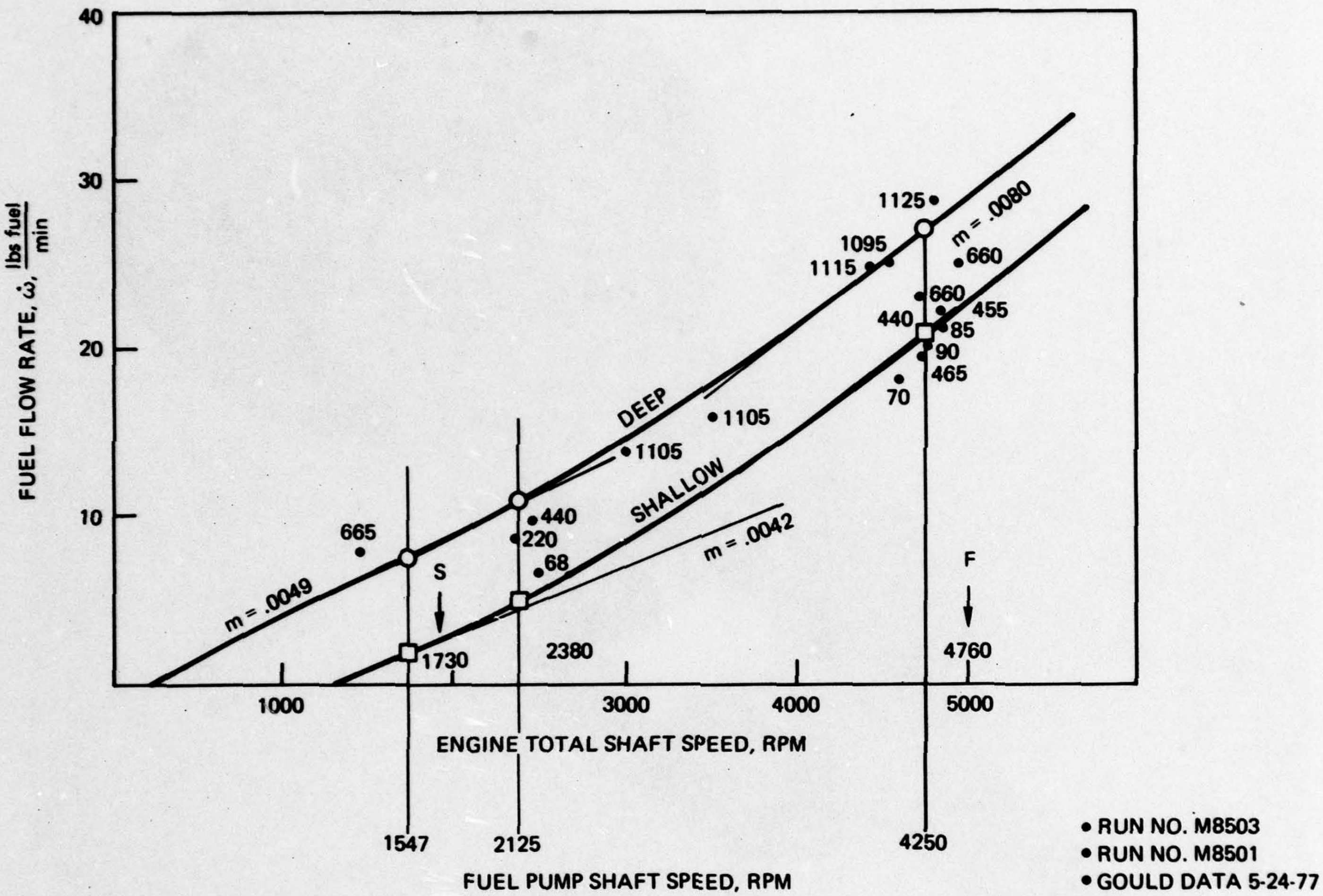


Figure 18. Estimated engine requirement (limited empirical data).

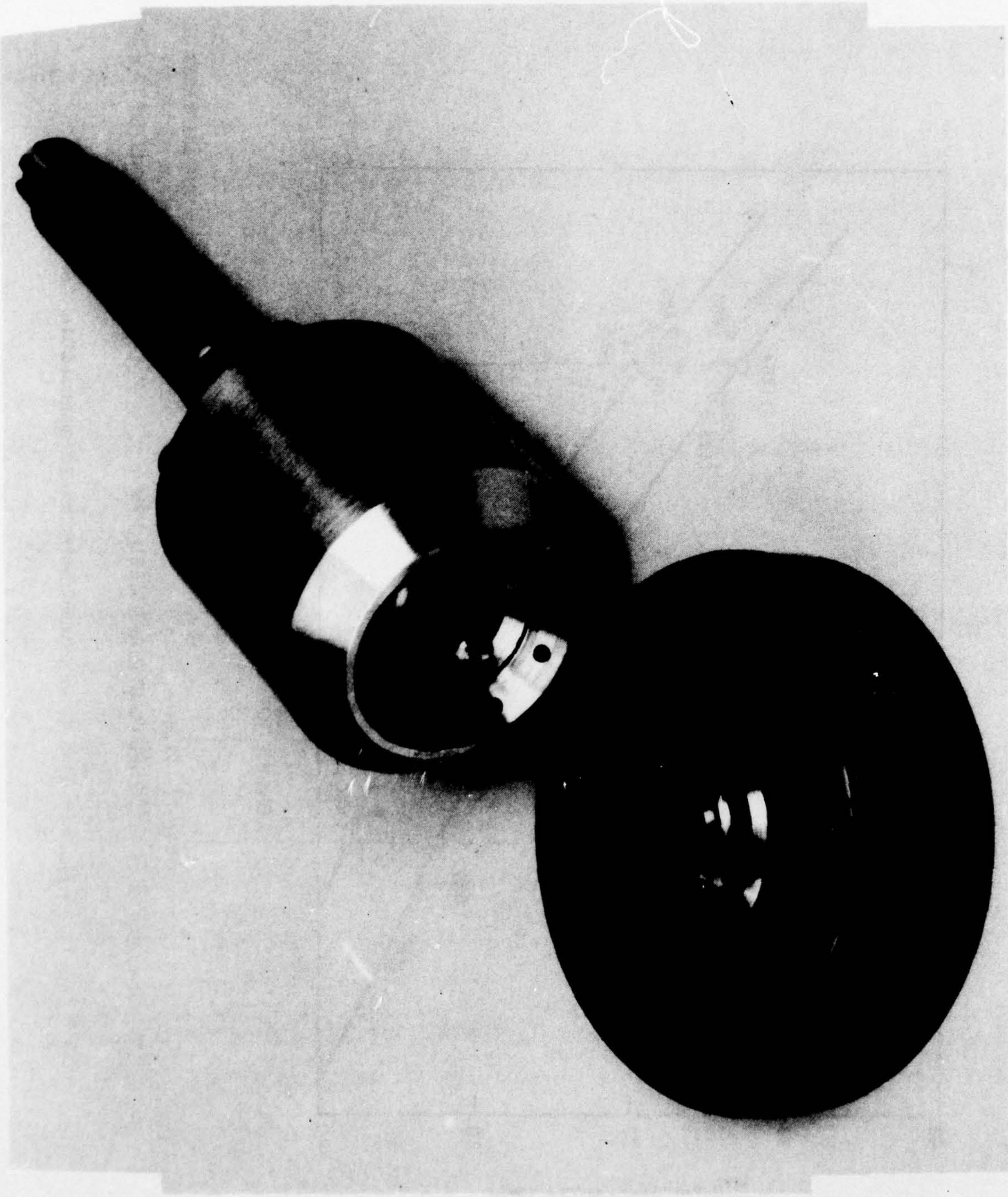


Figure 19. Original conical valve design.

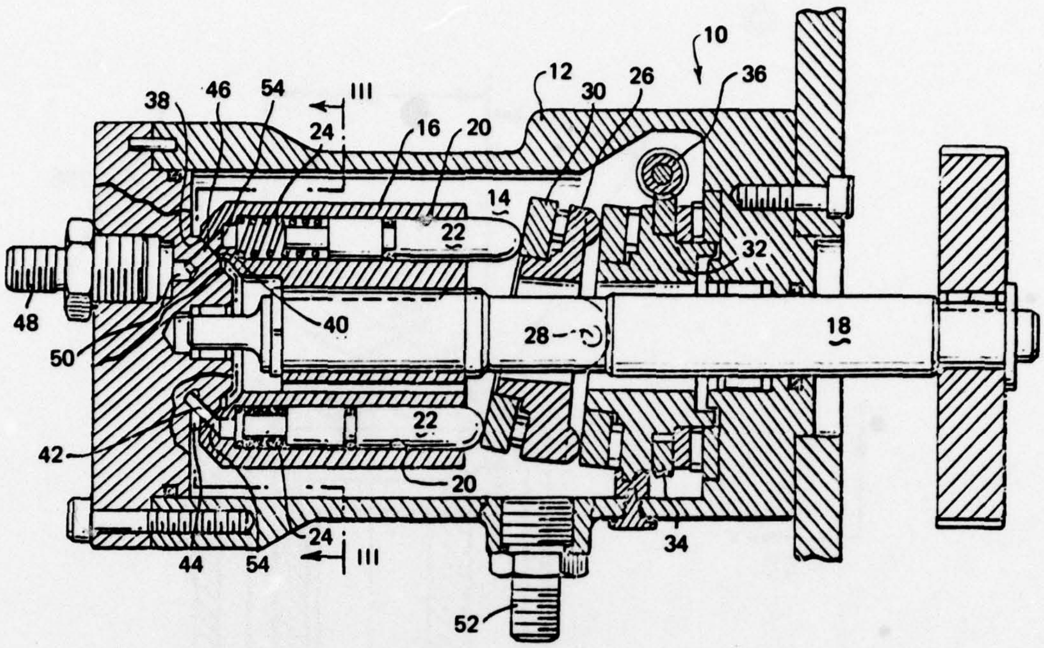


Figure 20. Conical valve design, cross-sectional view.

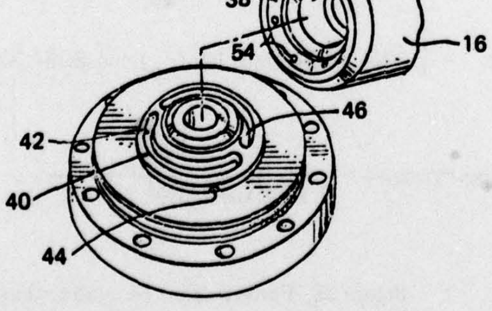
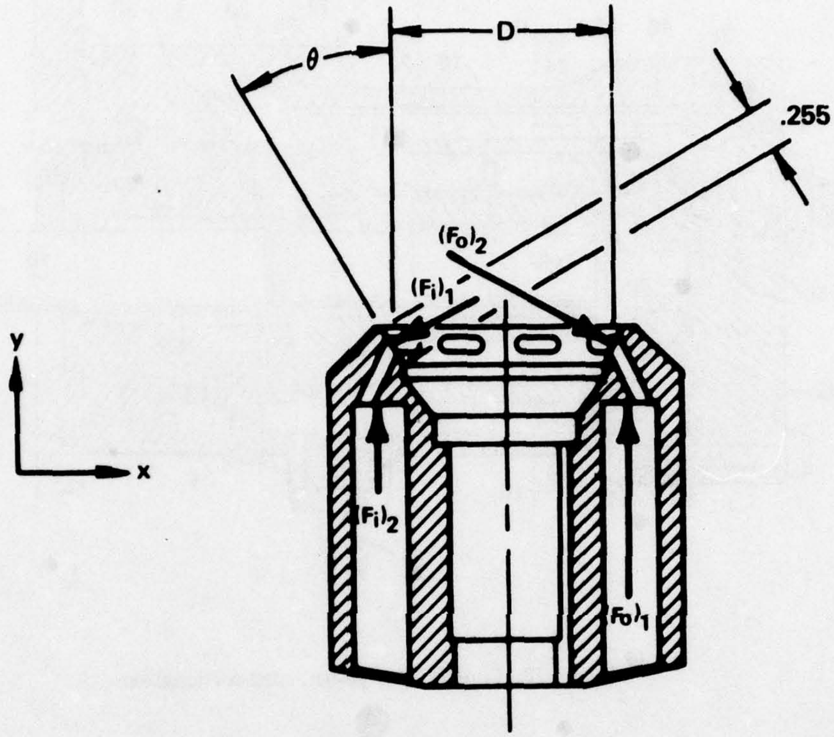


Figure 21. Conical valve design, perspective view.



$$(F_{net})_y = (F_o)_1 + (F_i)_2 - (F_i)_1 \sin\theta - (F_o)_2 \sin\theta$$

$$= \frac{\pi}{4}(.313)^2\ [4(P_o) + 4(P_i)]$$
$$- \ \pi(D)(.255)(\sin\theta^\circ)\left[\frac{160^\circ}{360^\circ}(P_o) + \frac{150^\circ}{360^\circ}(P_i)\right]$$

$$F_{net} = [.308 - .356(D)\sin\theta]\ (P_o) + [.308 - .334(D)\sin\theta]\ (P_i)$$

$$\text{Bearing Pressure} = \frac{F_{net}}{\pi(D)\sin\theta(.255-.1)(.6)}$$

Figure 22. Force balance on cylinder block.

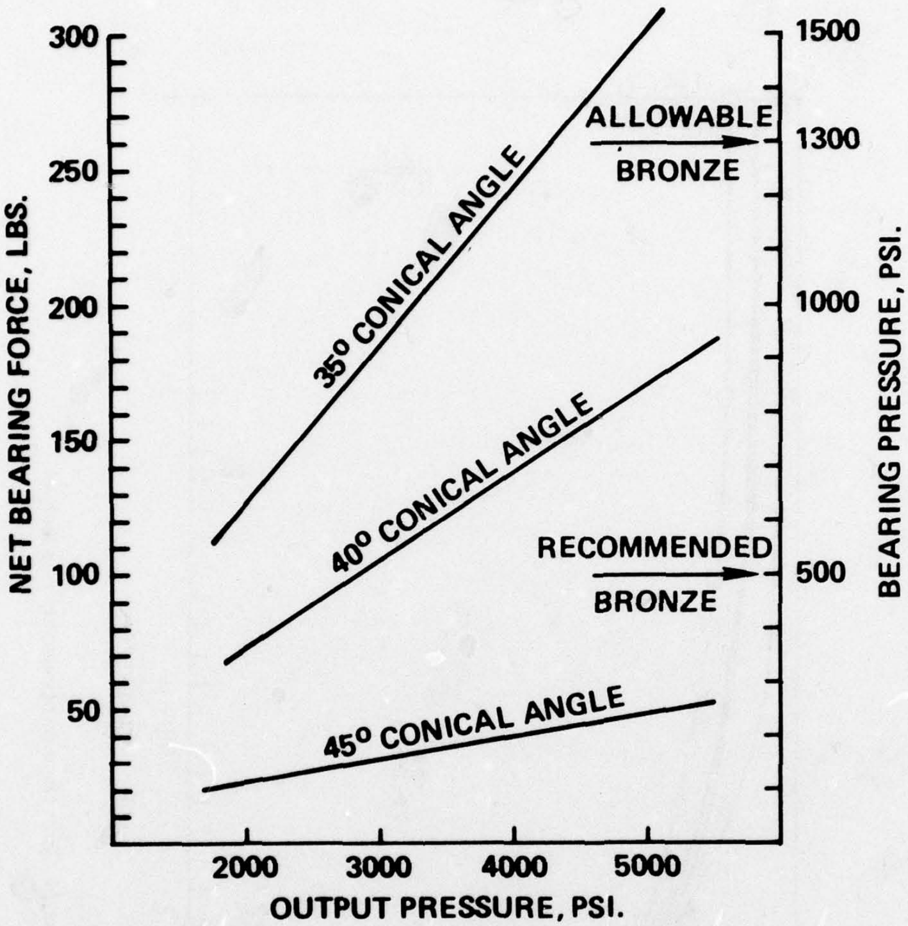


Figure 23. Bearing force and pressure vs output pressure and conical angle.

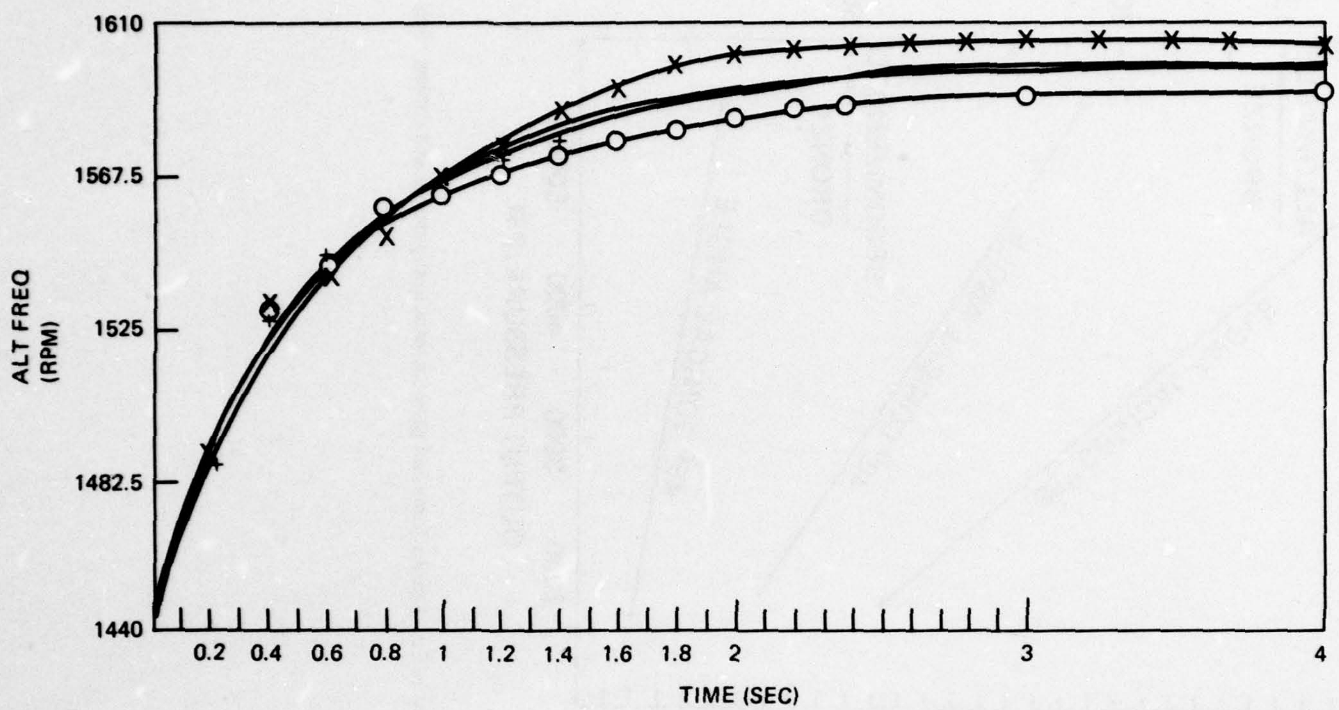


Figure 24. Time response of engine to step increase.

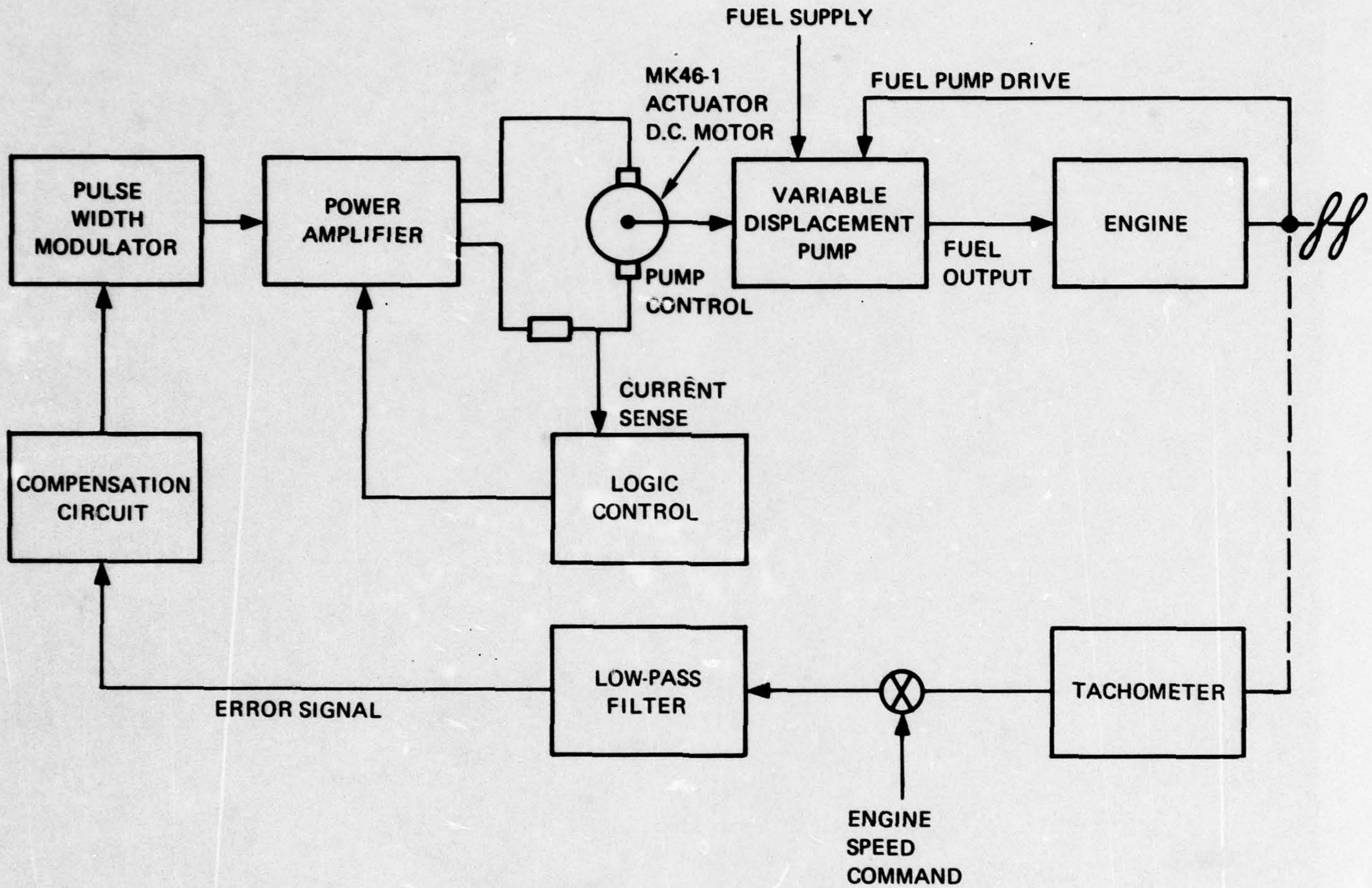


Figure 25. ALWT speed control circuit schematic.

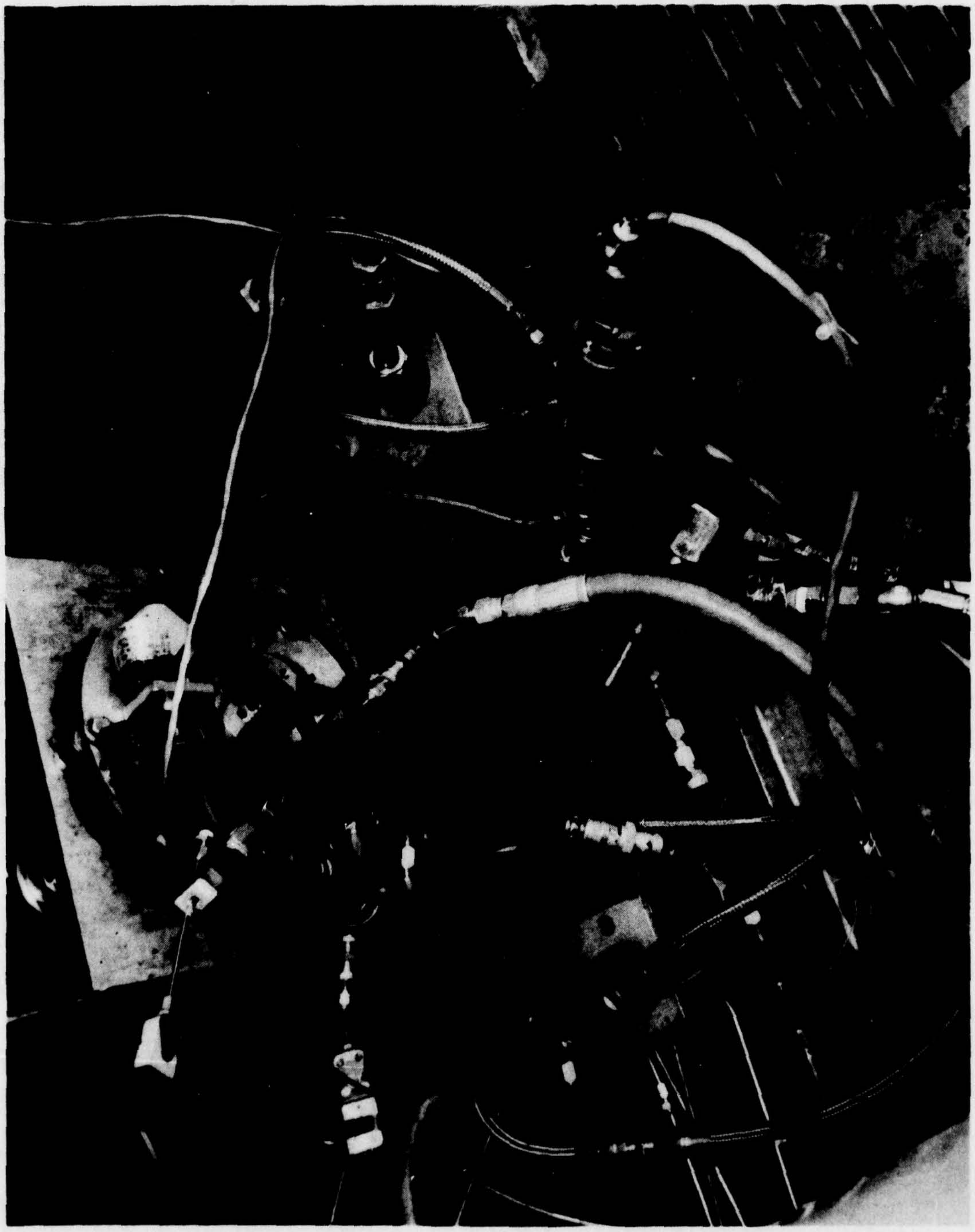


Figure 26. VD fuel pump test stand, side view.

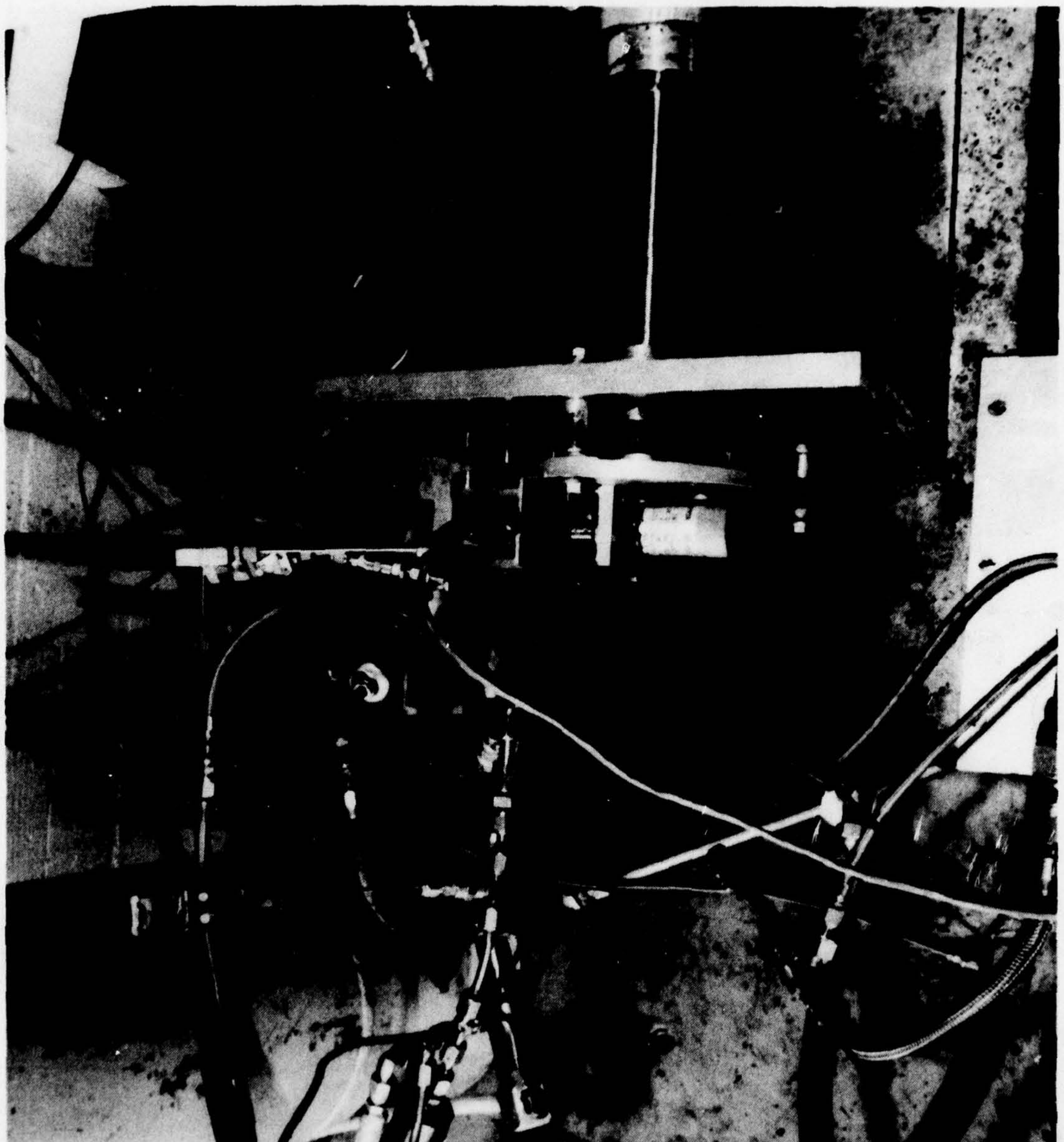


Figure 27. VD fuel pump test stand, top view.

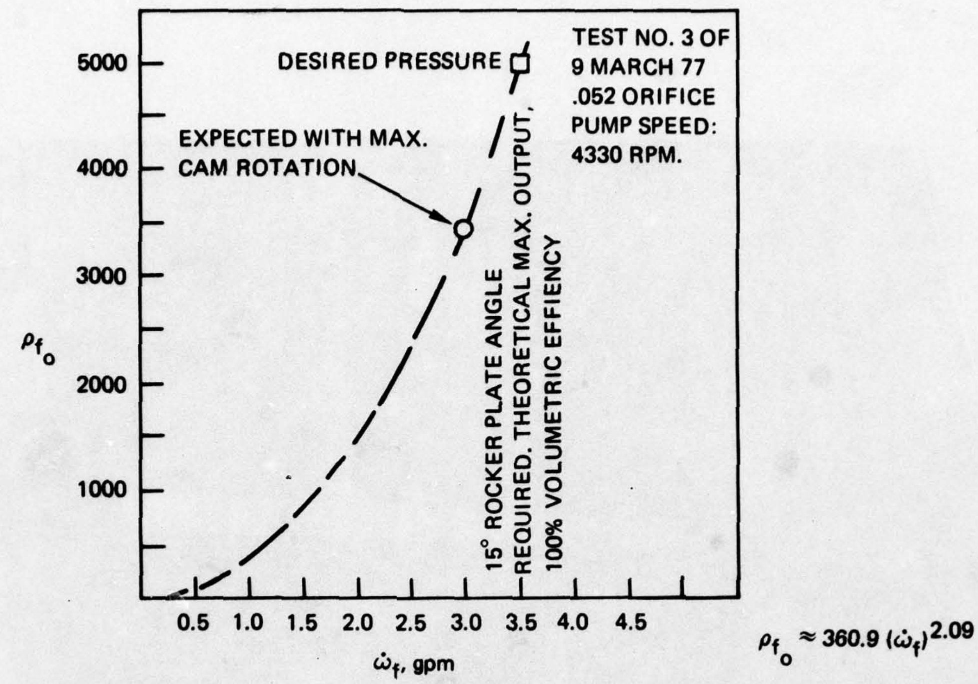


Figure 28. Theoretical analysis of VD fuel pump data of 9 and 16 March 1977.

$$\dot{\omega} = C_D A \rho V$$

$$\dot{\omega} = C_D A \rho \sqrt{2\, g \frac{\Delta P}{\rho}}$$

or

$$\Delta P = \frac{W^2}{d^4\, C^2 \gamma}\, 3.63$$

For VD pumps

$$R_e = \frac{\rho D V}{\mu} = \frac{1.937\ (.100)\ 35.7}{2.12 \times 10^{-5}\ (12\ \text{in./ft})}$$

$$R_e = 2.7180 \times 10^5$$

then

$$0.6 < C < 0.8$$

therefore

$$(\Delta P)_{max} = \left(\frac{3.36\ (62.4)\ 1.2}{1728}\right)^2 \frac{3.63}{(0.1)^4\ (0.6)^2\ 62.4\ (1.2)}$$

$$\Delta P = 28.5 \text{ psi max}$$

Therefore the minimum P_{f_i} such that the flow into the cylinders is not choked is approximately 29 psi.

Figure 29. Minimum inlet pressure required to prevent choking of VD fuel pump.

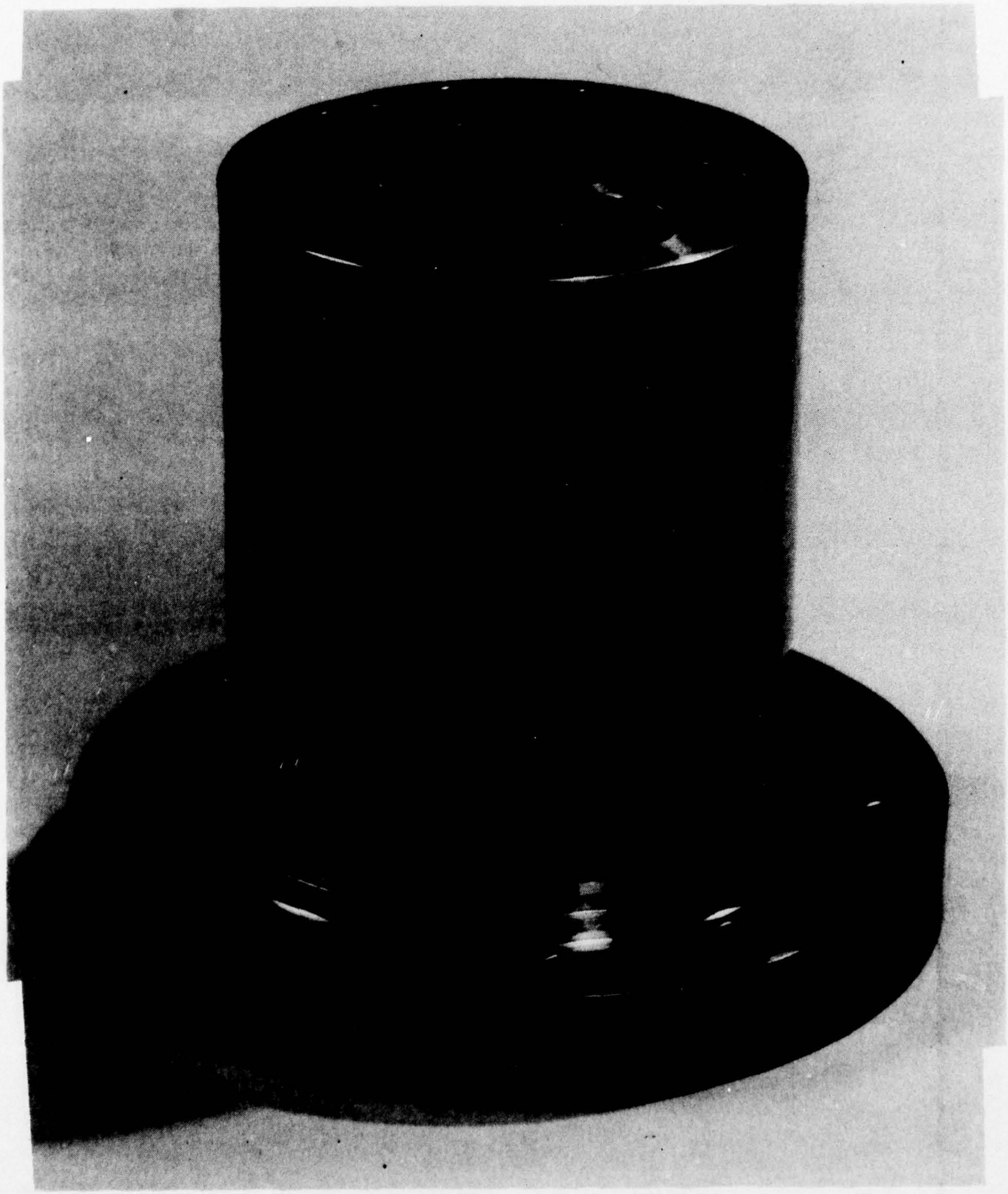


Figure 30. Cylinder block with beveled aft edge.

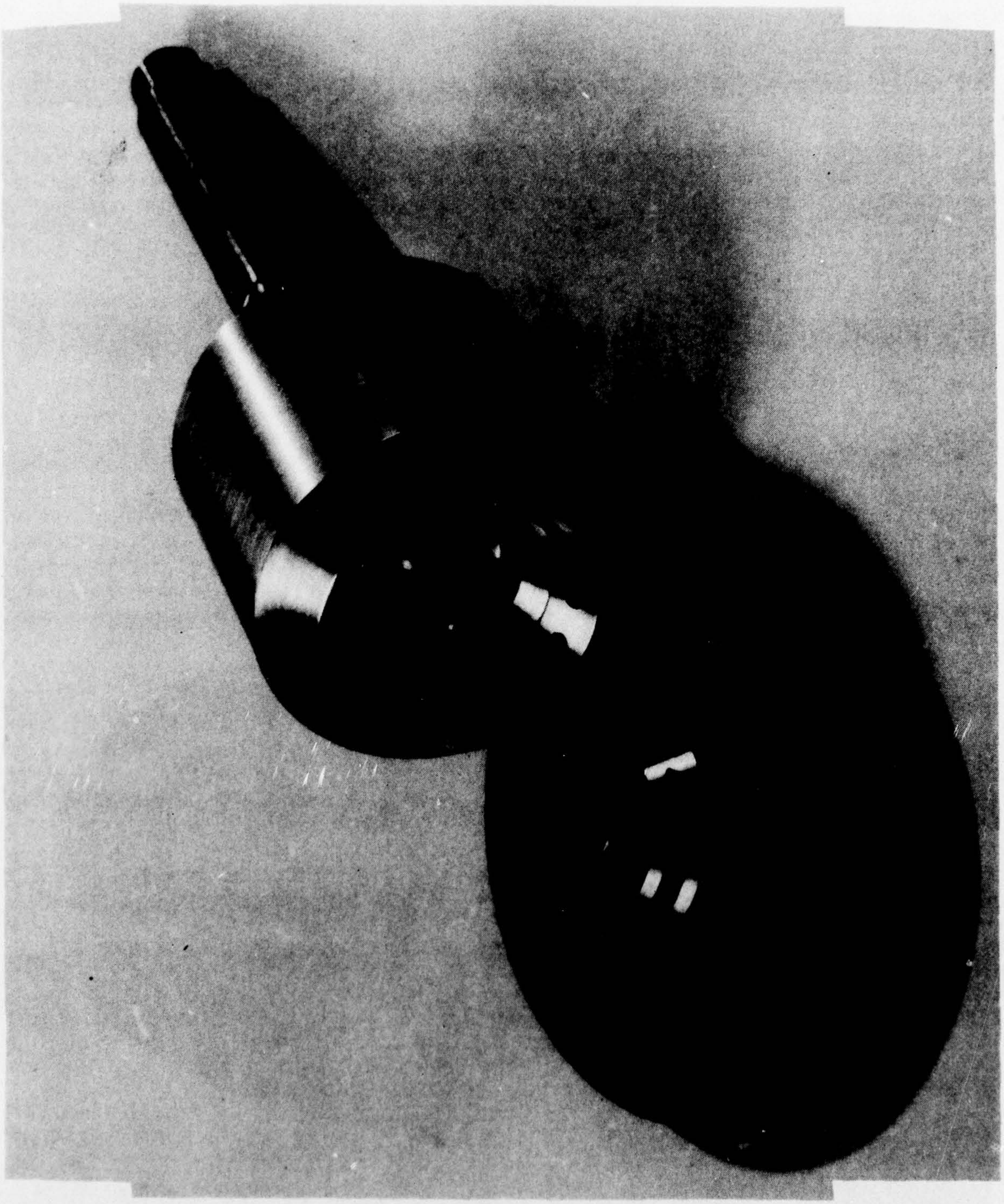


Figure 31. Cylinder block and pump head with enlarged ports.

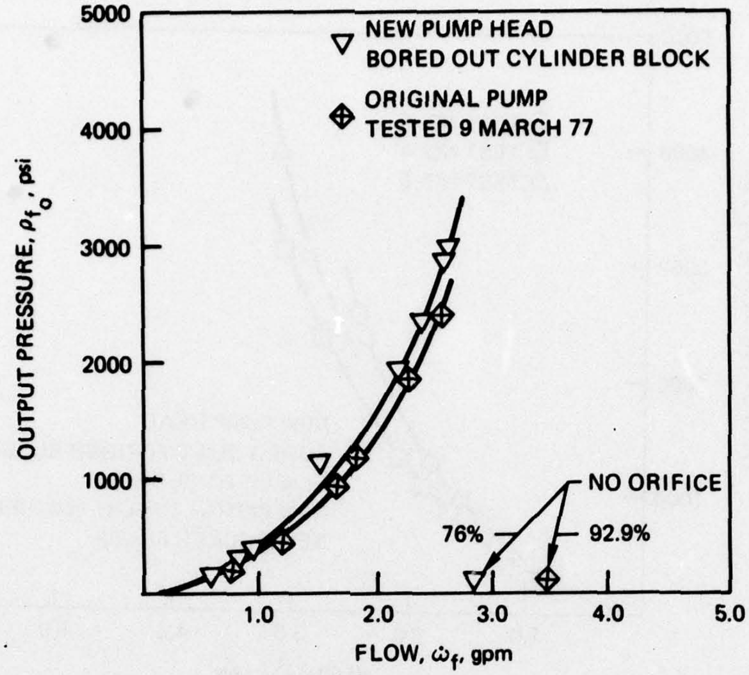


Figure 32. VD fuel pump performance (3- to 12-deg angle-plate, 0.052-in. orifice). Data of 20 June 1977.

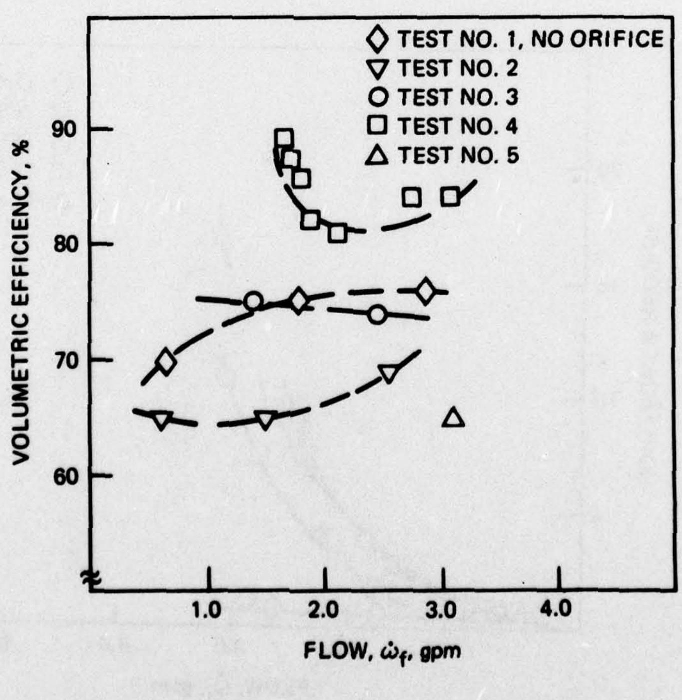


Figure 33. VD fuel pump volumetric efficiency. Data of 20-21 June, 1977.

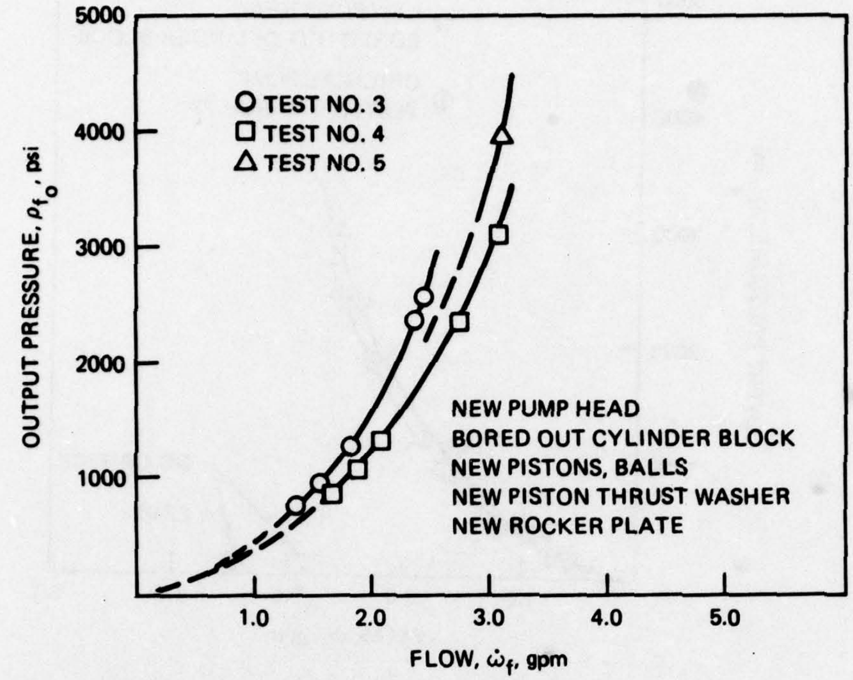


Figure 34. VD fuel pump performance (6- to 15-deg angle-plate, 0.052-in. orifice). Data of 20 June 1977.

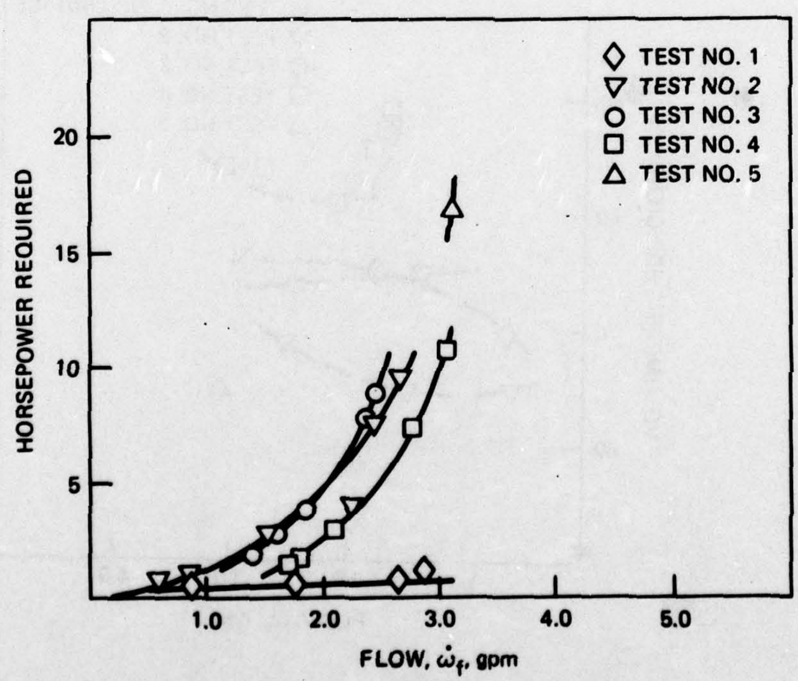


Figure 35. VD fuel pump performance, horsepower vs flow. Data of 20 June 1977.

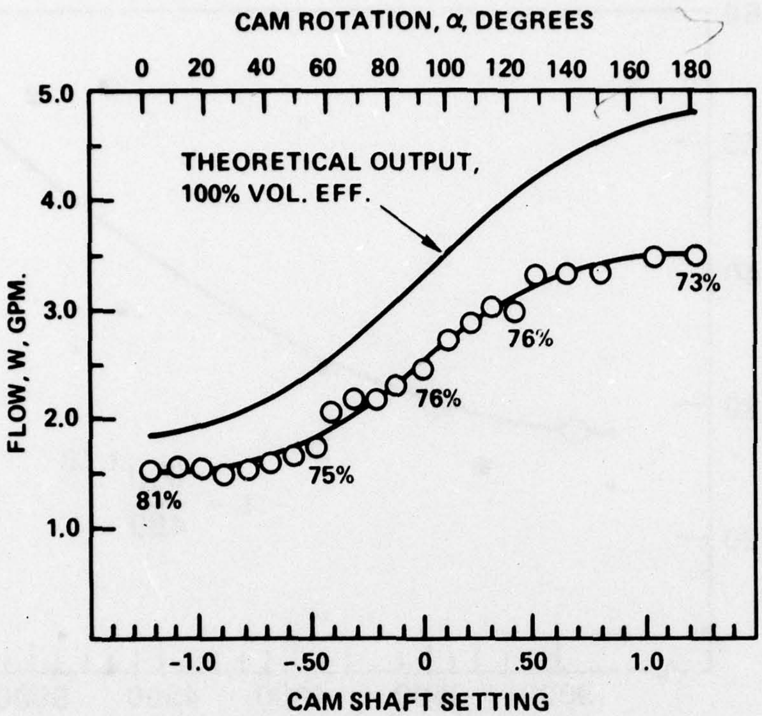


Figure 36. VD fuel pump output vs cam rotation for 45-deg conical valve, 0.052-in. orifice. Circles indicate actual data of 12 July 1977. Percentages shown are actual volumetric efficiencies. Teflon O-rings were used on the pistons.

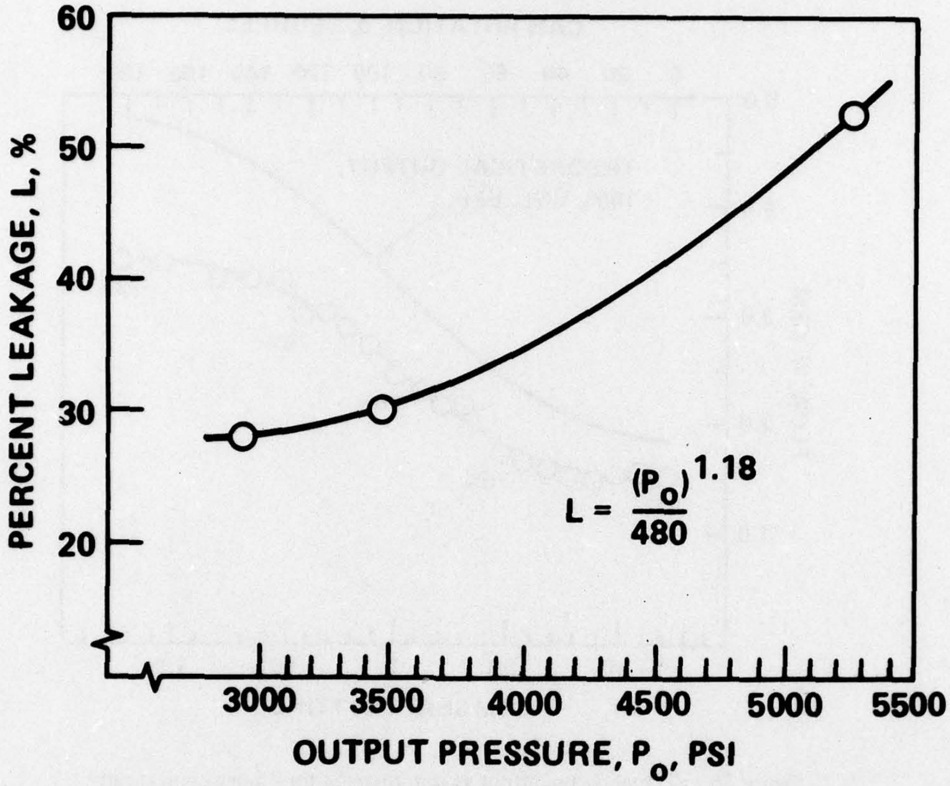


Figure 37. Internal leakage vs output pressure for 45-deg conical valve.

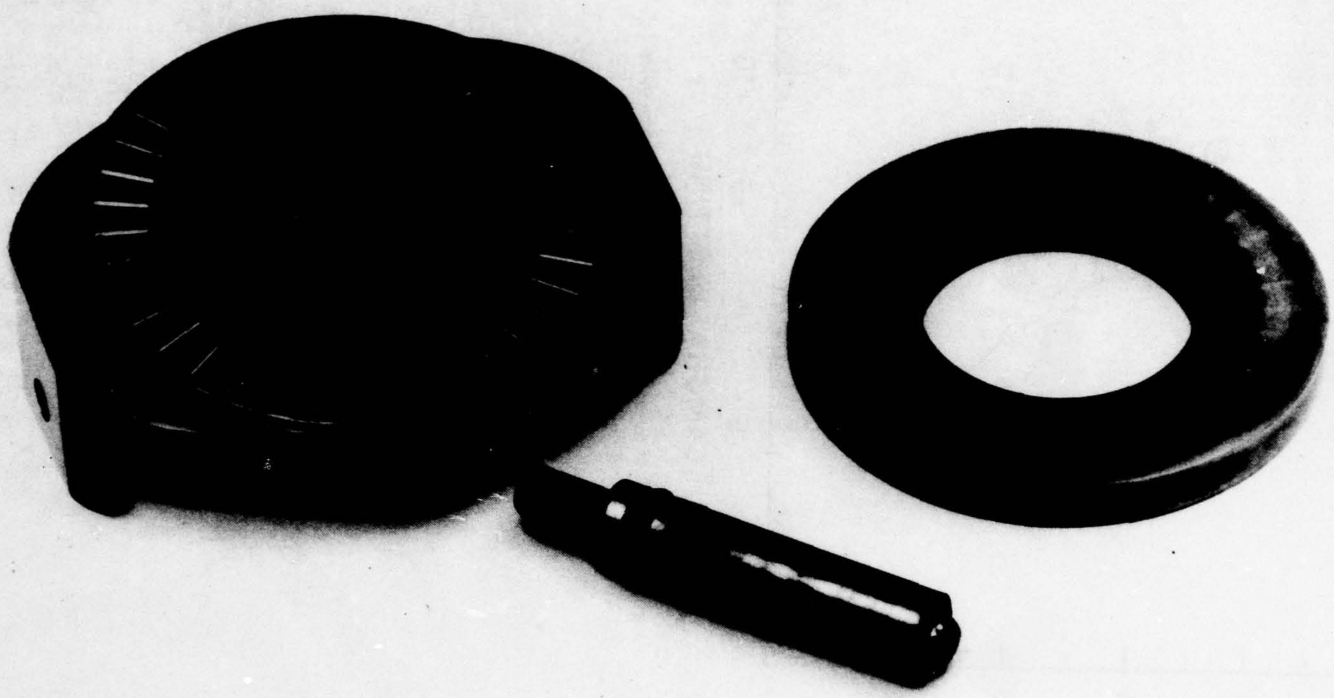


Figure 38. Latest modifications to angle-plate, pistons, and thrust washer.

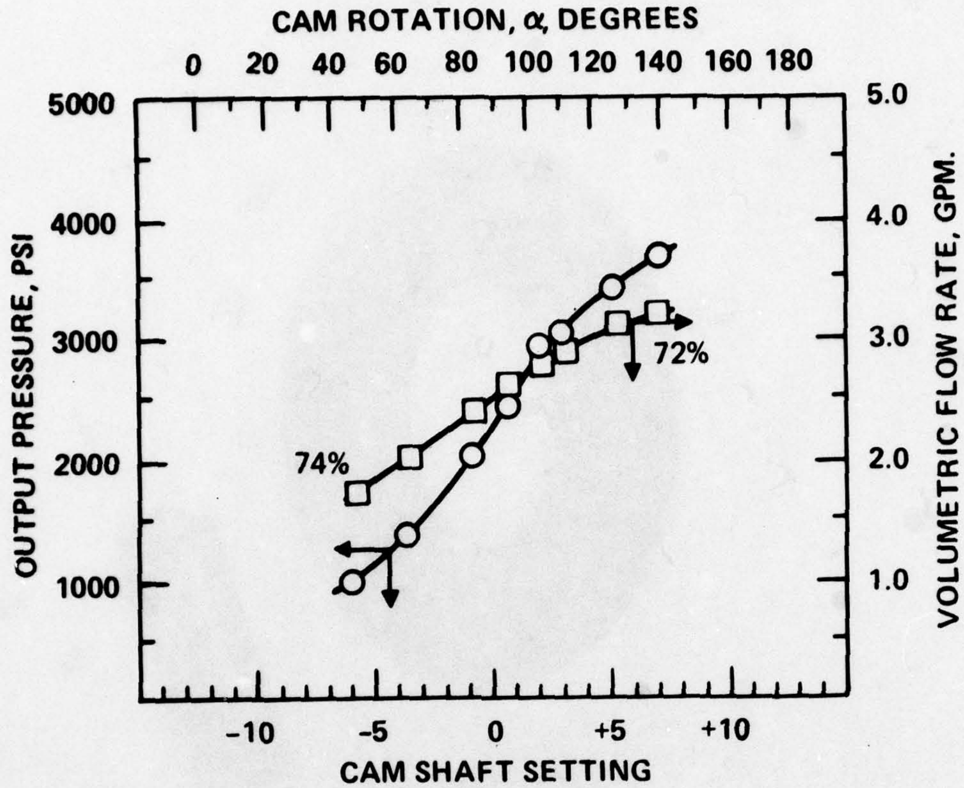


Figure 39. VD fuel pump performance (45-deg conical valve, 0.0465-in. orifice, C = 0.79).

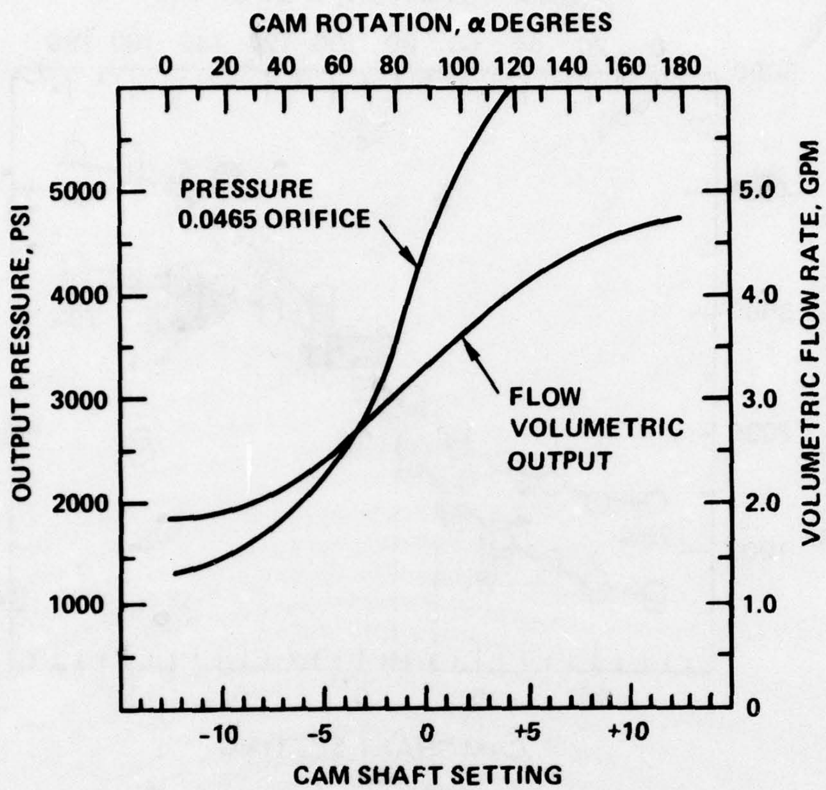


Figure 40. VD fuel pump performance, theoretical 100% efficiency.

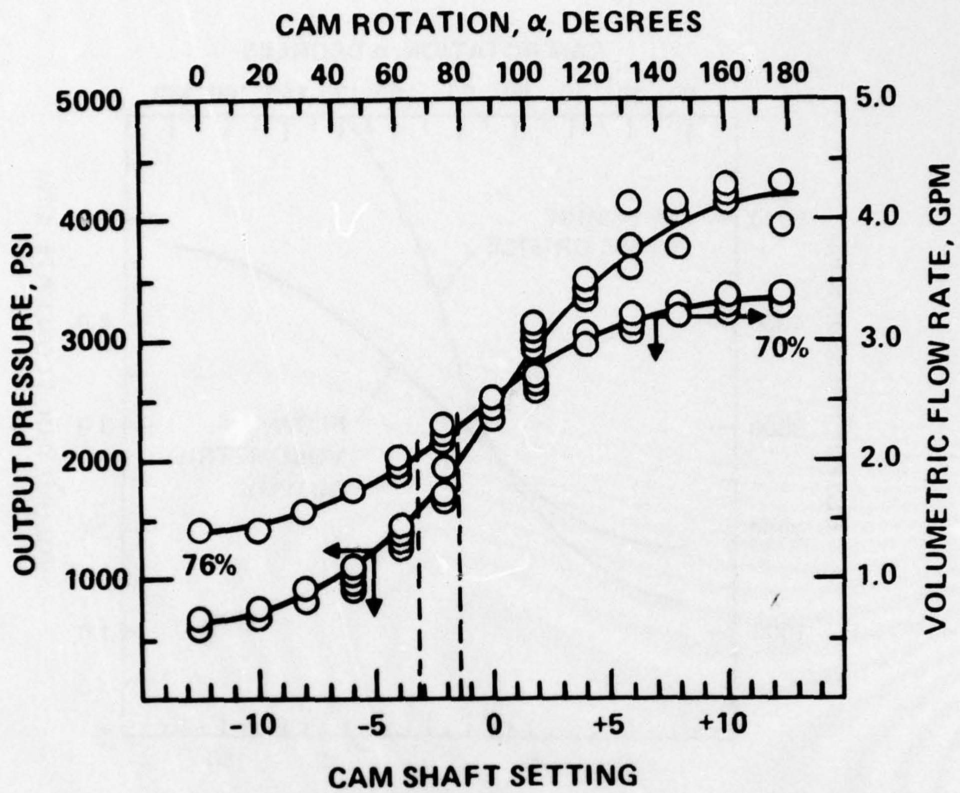


Figure 41. VD fuel pump performance (40-deg conical valve, 0.0465-in. orifice, C = 0.79).

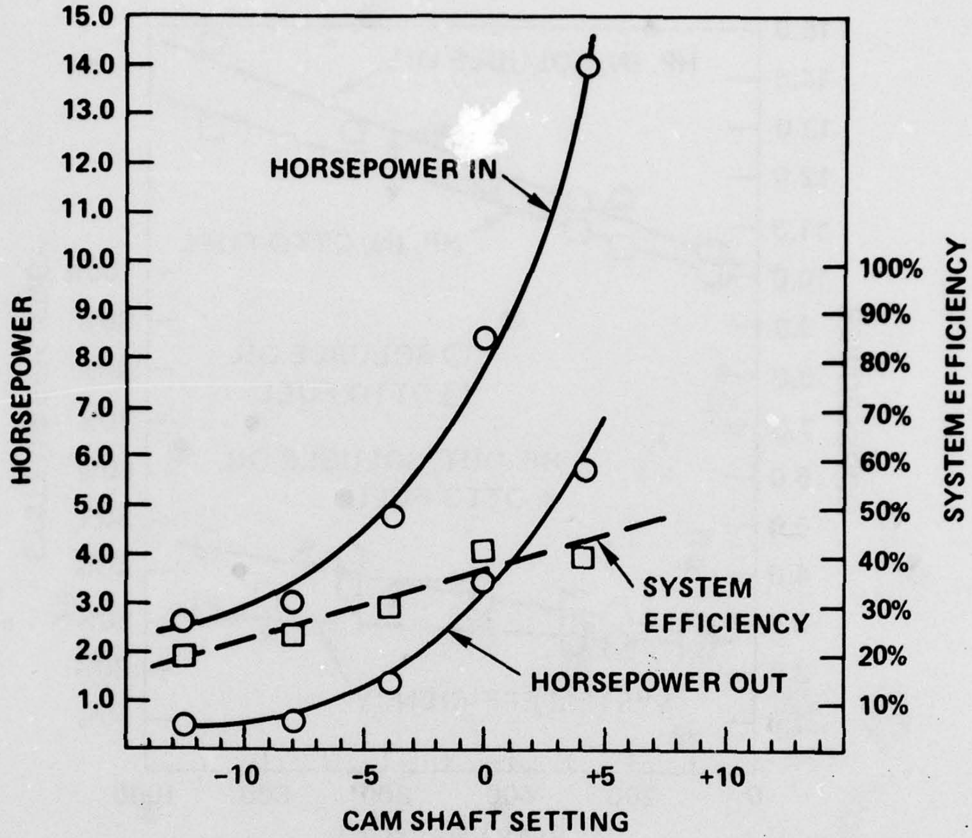


Figure 42. VD fuel pump energy analysis (40-deg conical valve, 0.0465-in. orifice).

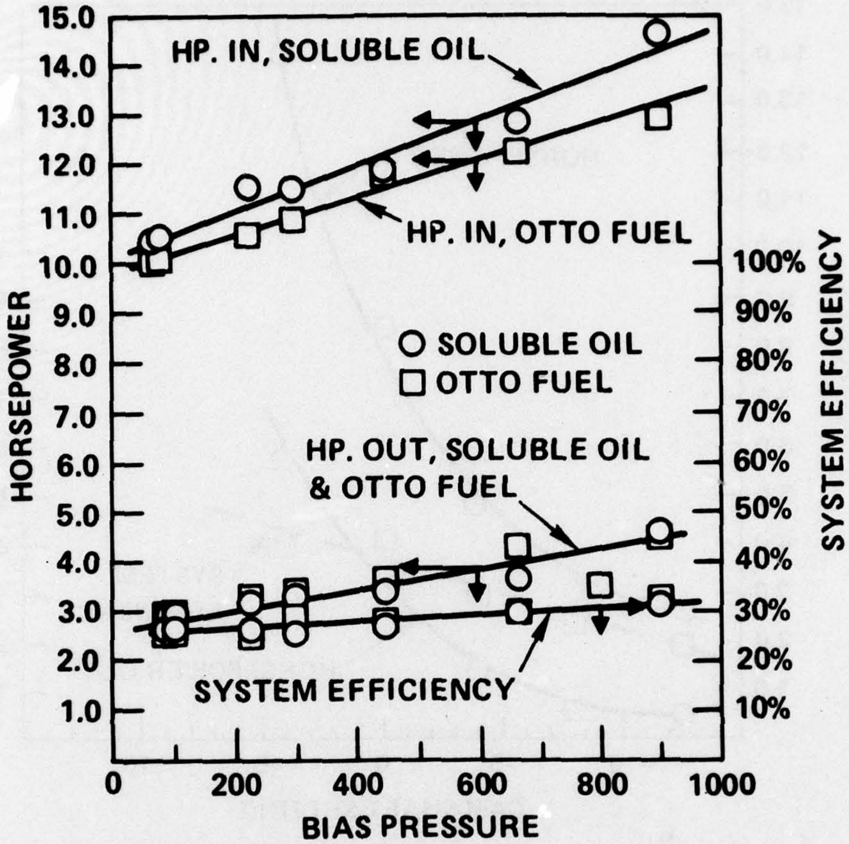


Figure 43. Equivalent fixed-displacement ALWT backup fuel pump No. 4 (0.038-in. orifice, 4100 rpm).

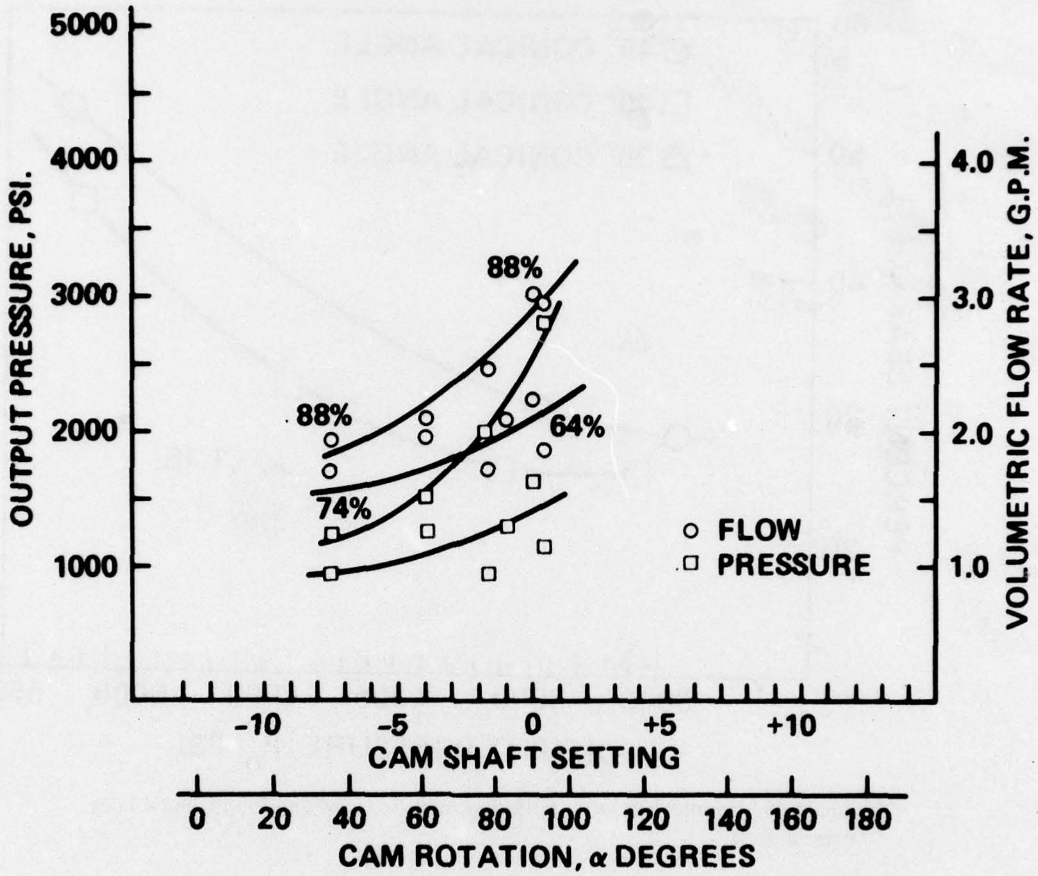


Figure 44. VD fuel pump performance (35-deg conical valve, 0.0465-in. orifice).

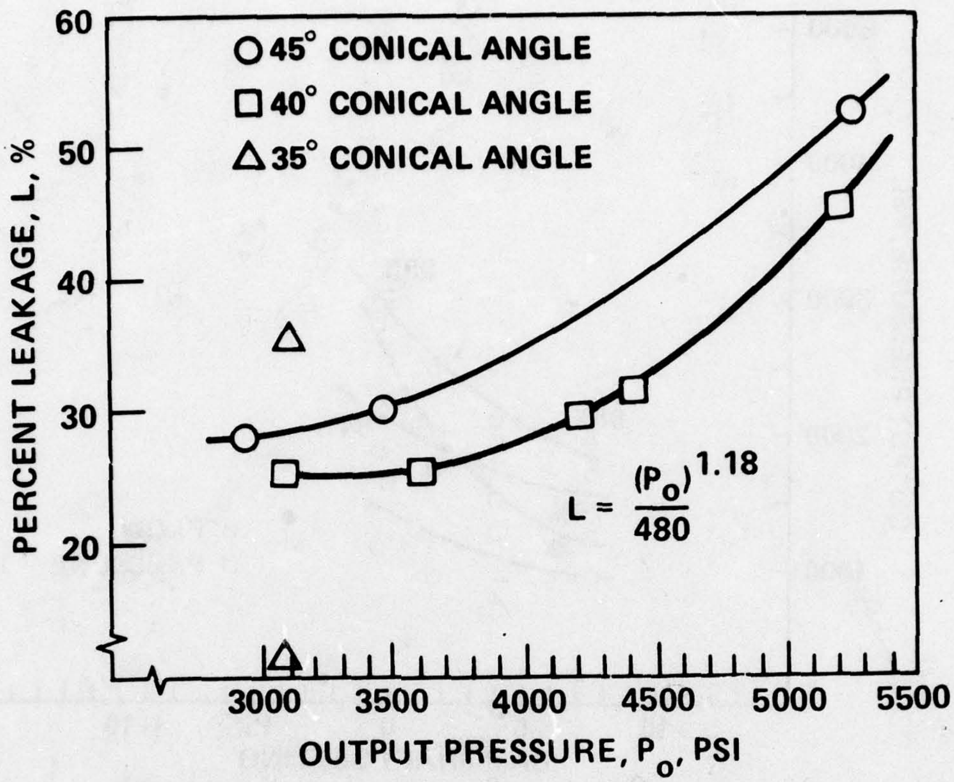


Figure 45. Internal leakage vs output pressure for valves of 35, 40, and 45 deg conical angle.

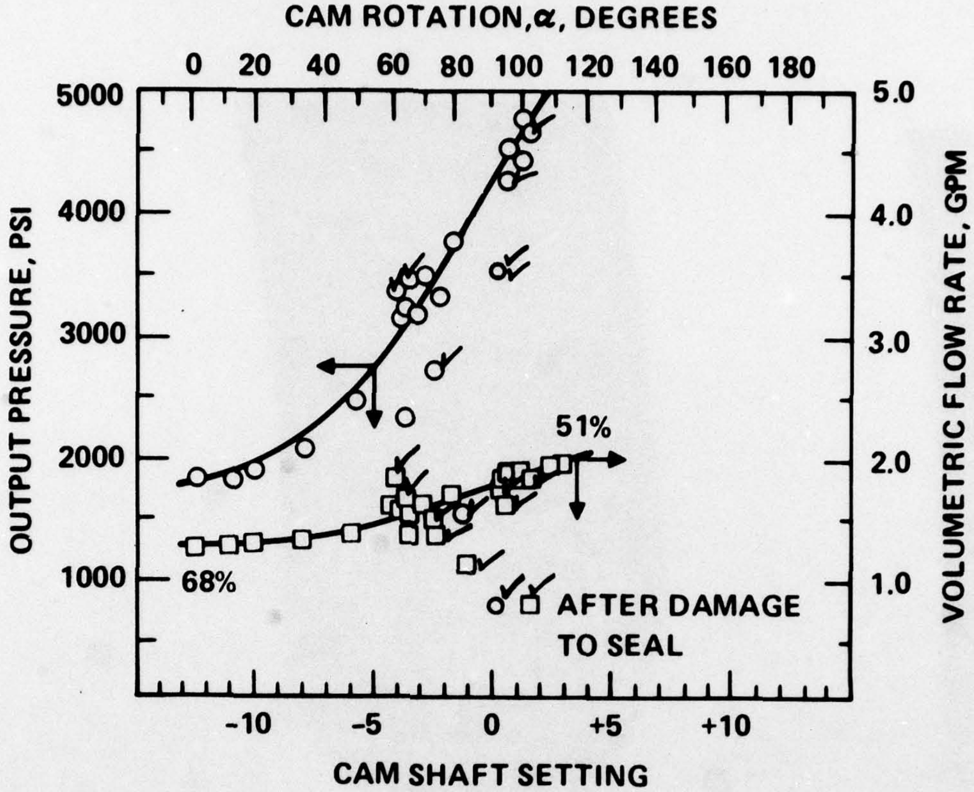


Figure 46. VD fuel pump performance (0.038-in. orifice, C = 0.64).

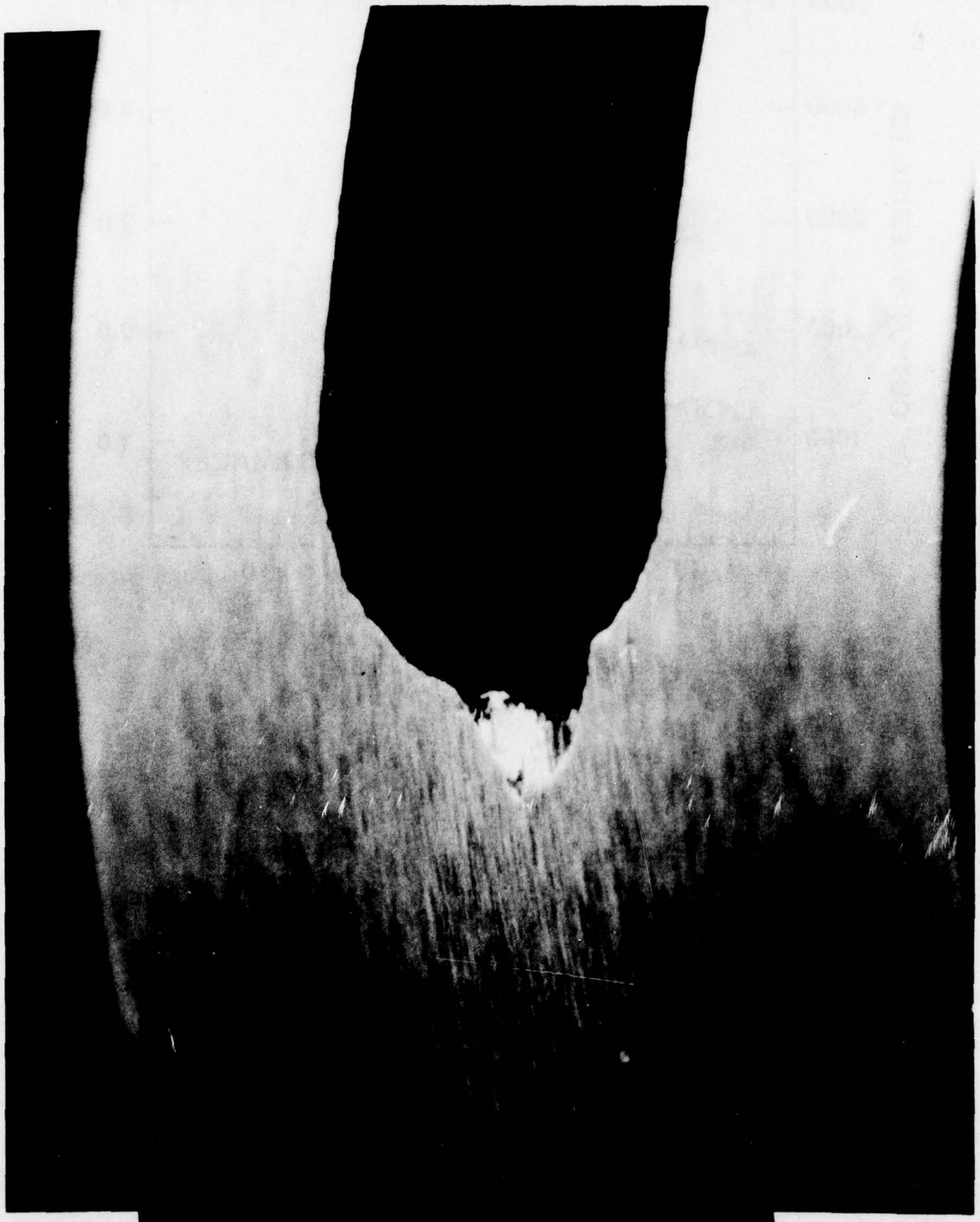


Figure 47. Damaged intake slot of pump head.

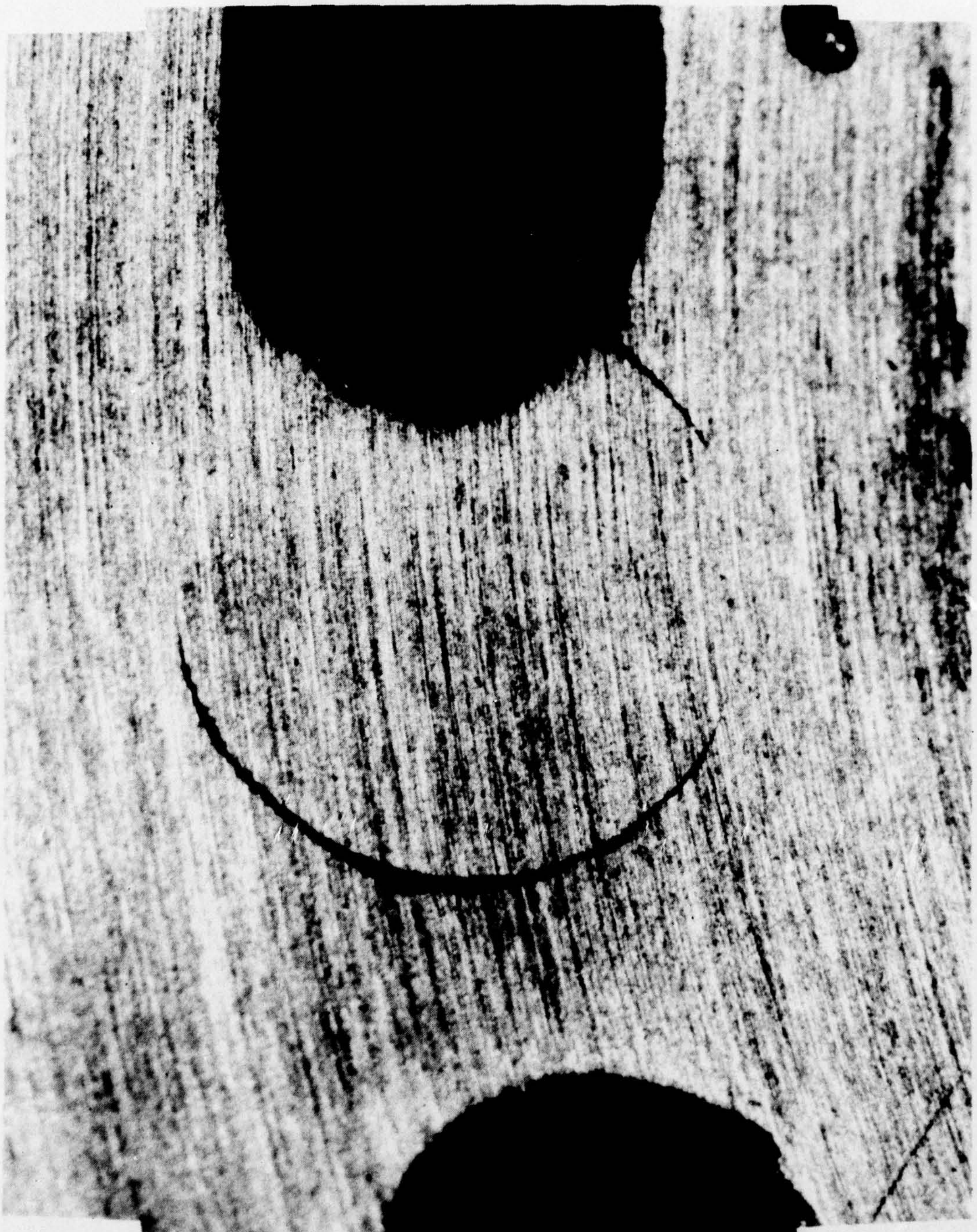


Figure 48. Reworked edge of intake slot of pump head.

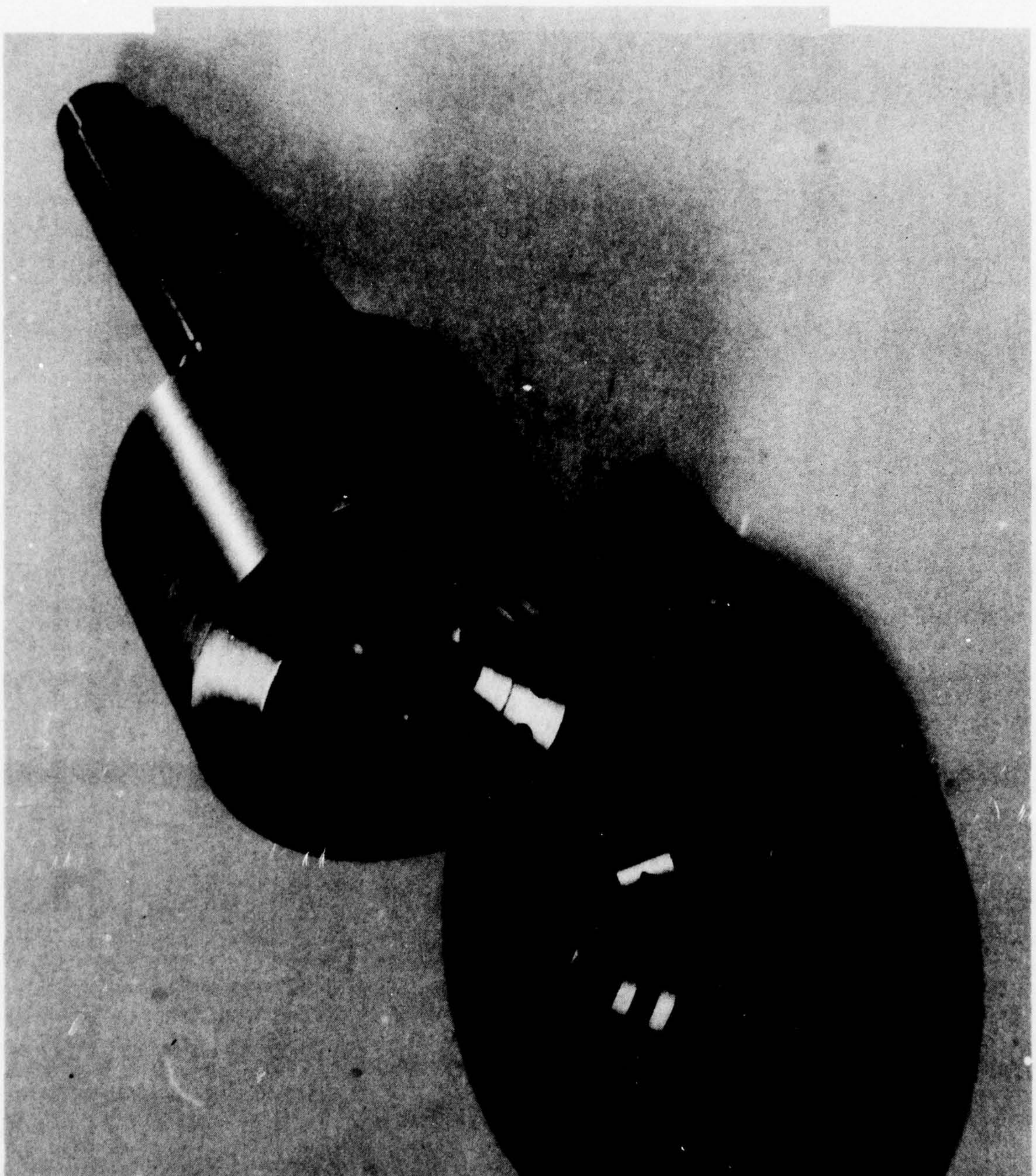


Figure 49. Reworked cylinder block and pump head.

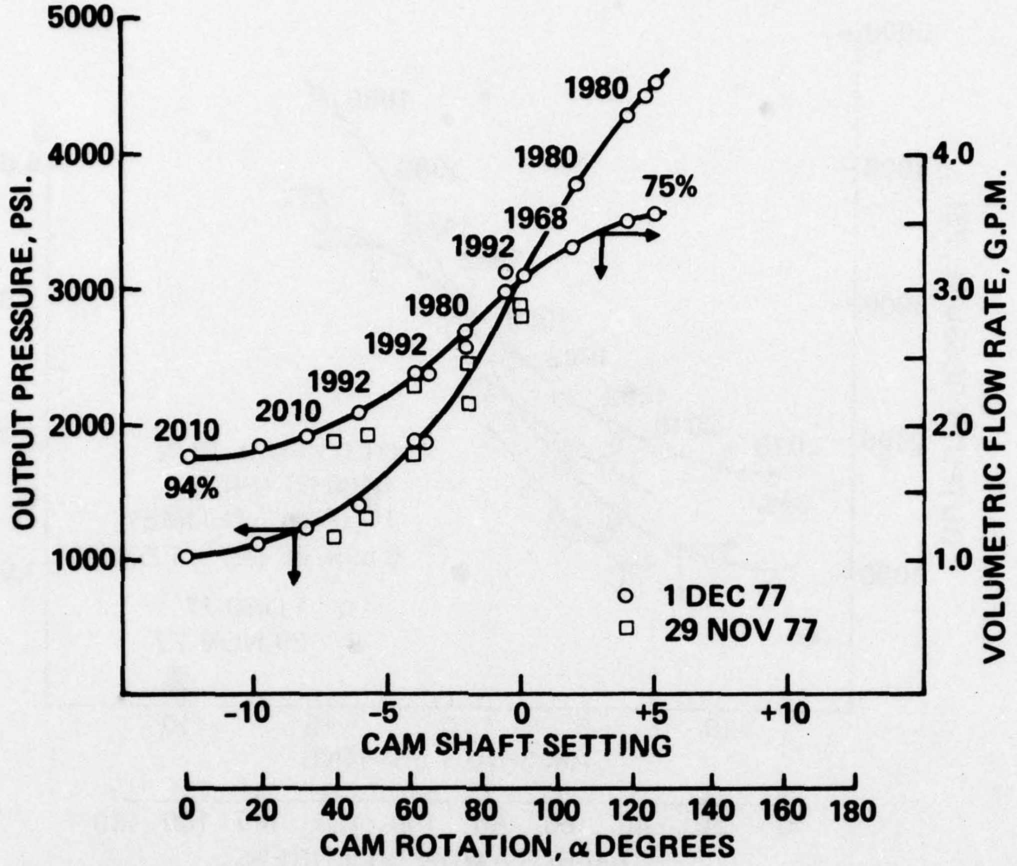


Figure 50. VD fuel pump (reworked) performance (40-deg conical valve, 0.0465-in. orifice).

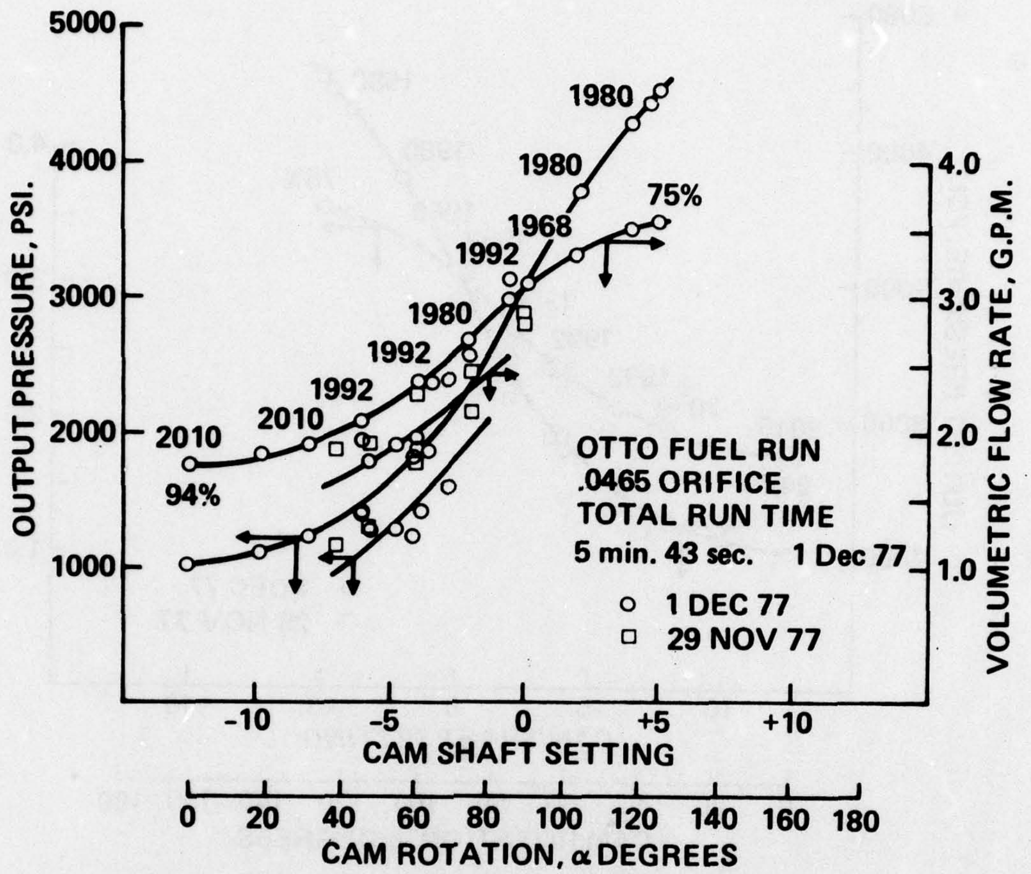


Figure 51. VD fuel pump (reworked) performance run with Otto fuel (40-deg conical valve, 0.0465-in. orifice).

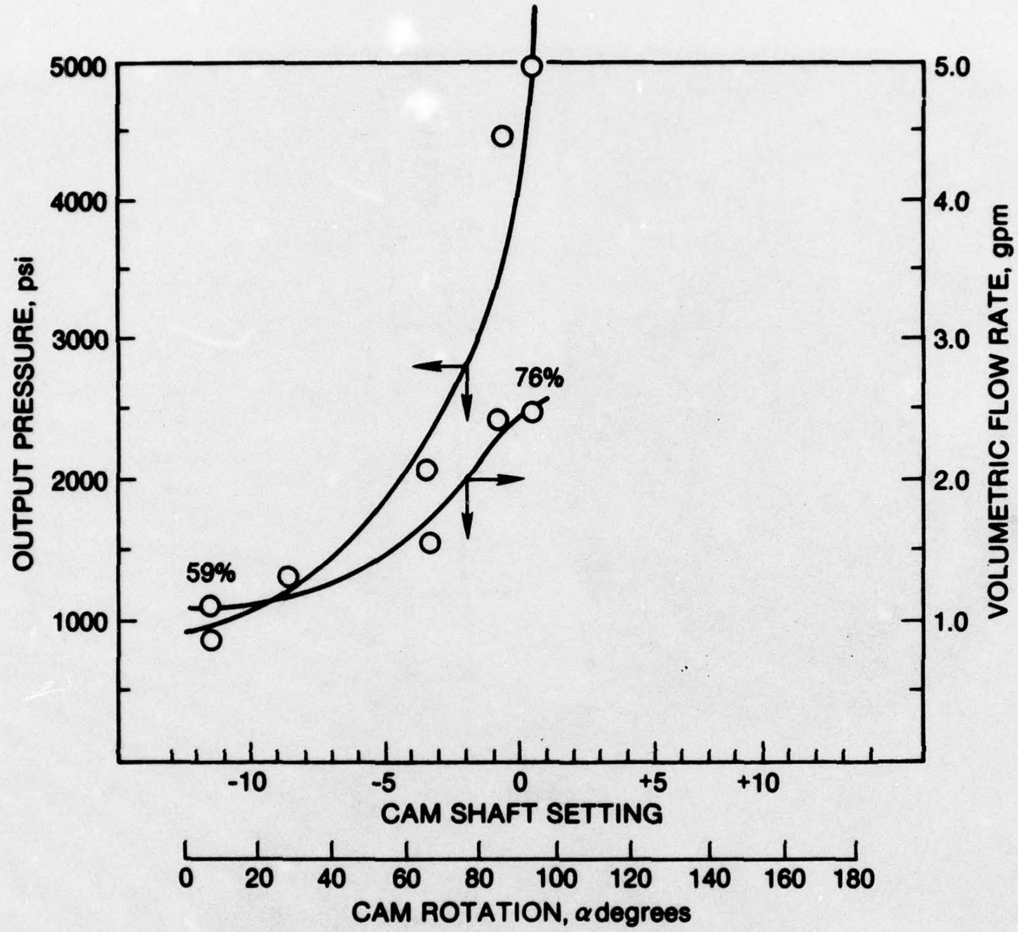


Figure 52. VD fuel pump performance (30-deg conical valve, 0.043-in. orifice). Data of 28 March 1977.

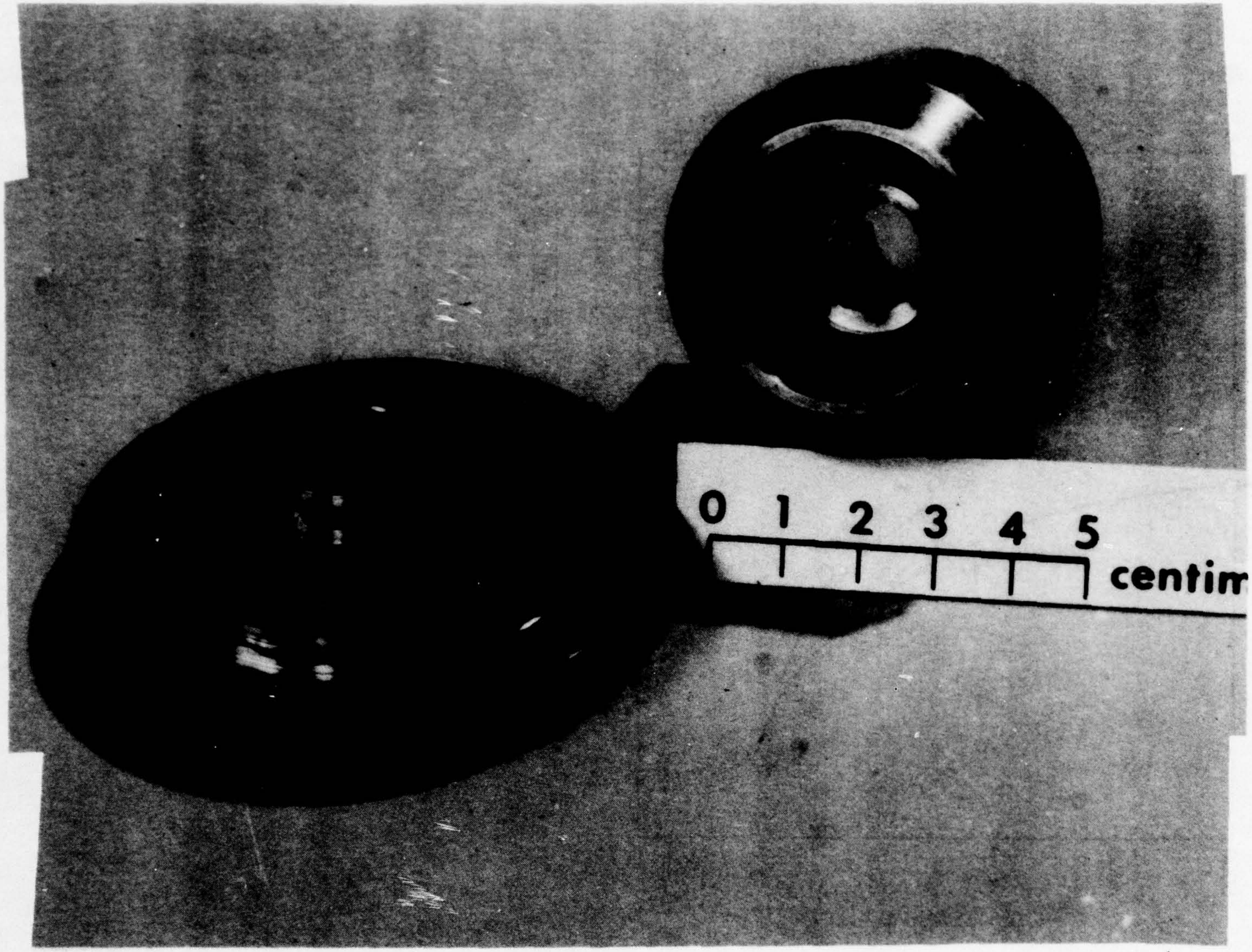


Figure 53. 30-deg conical valve, pump head, and cylinder block.

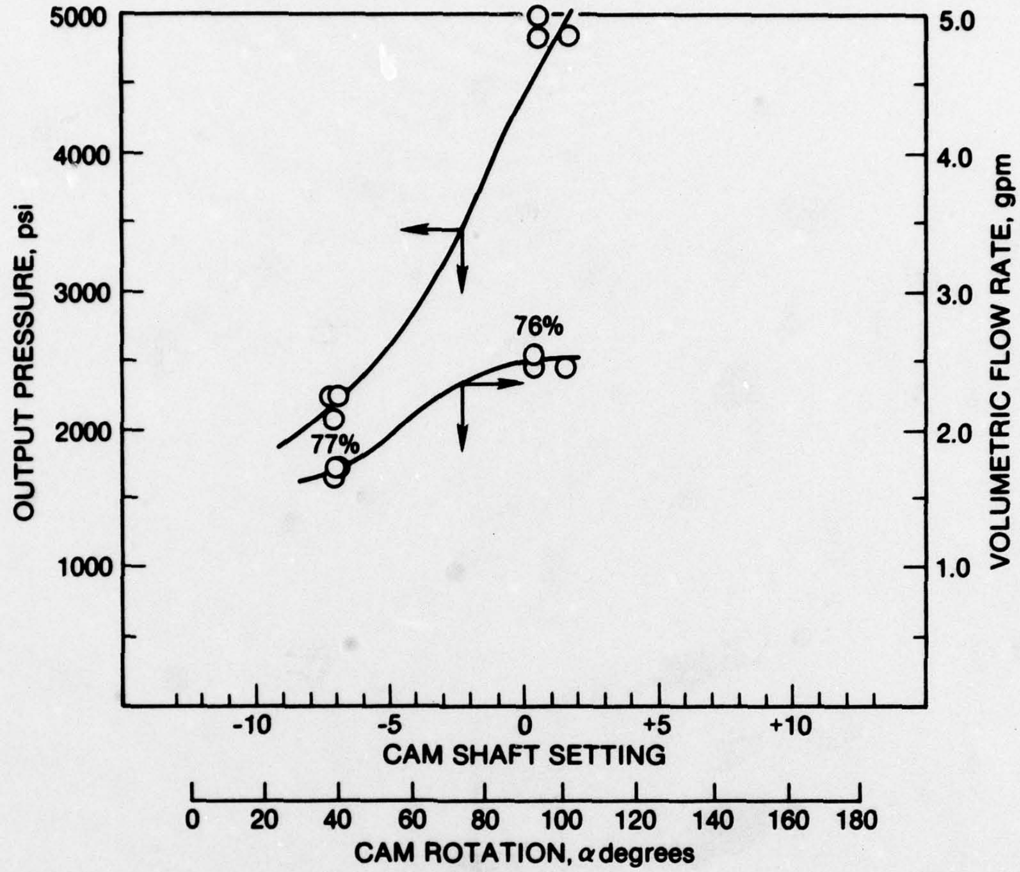


Figure 54. VD fuel pump performance (40-deg conical valve, 0.043-in. orifice). Data of 5 May 1978.

APPENDIX
MATERIALS TESTING FOR CYLINDER BLOCK AND PUMP HEAD

IN REPLY REFER TO:
LJM:pfw
Ser 611/61/78
10 April 1978

MEMORANDUM

From: L. J. Martini. Code 6114
To: R. W. Shaddock, Code 6114

Subj: Materials Testing for ALWT Variable Displacement Fuel Pump

Encl: (1) Layout, Test Fixture., NUC 05614
(2) Test Pieces, Block and Head, NUC 05613
(3) Test Fixture Force Analysis
(4) Results Matrix
(5) Actual Data, 2 pages
(6) Photographs of Test Pieces, 9 pages

1. The materials test fixture shown in enclosure (1) was manufactured to determine the optimum materials for the piston block and head of the ALWT Variable Displacement Fuel Pump. The test fixture was designed to simulate the actual fuel pump environment, bearing pressures and surface speeds. The test pieces are dimensioned the same as the block and head at their interface surfaces; see enclosure (2).

2. Enclosure (3) presents the resultant bearing force between the simulated block and head given a load pressure and tank pressure. The load pressure acting on the 1.125 dia. piston (item 3 of the Test Fixture Layout, enclosure (1)) minus the tank pressure acting on the effective bearing surfaces of the test pieces (Items 8 and 9 of enclosure (1)) results is the net bearing force pushing the two test pieces together. The 35° and 40° conical angle ranges are approximations based on the theoretical calculations of the forces acting on the actual block and head in the Variable Displacement Fuel Pump. The various test points designate the materials tested and accumulated time of run.

3. The tool steel head against the bronze block (baseline test) had to be run for seven minutes at approximately 300 lbs net bearing force before the type of failure that had occurred in the VD fuel pump could be simulated; see enclosure (3). This failure consists of excessive wear of the bronze and microscopic surface cracking of the tool steel; see enclosure (6) pages 1 and 2. Graph-Air against Graph-Air was the next test and showed very poor results, surface cracking in only 10 sec. run time; see enclosure (6) pages 3 and 4. The Nitronic material failed by imbedding itself in the bronze after three minutes at 250 lbs net bearing force; see enclosure (6), page 5. The Nitronic against tool steel failed by galling which caused noticeable vibration after 1:19 minutes at 90 lbs net bearing force; see enclosure (6), page 6. The Graph-Air against bronze showed excessive wear of the bronze with the addition of circumferential scratches (possibly caused by the graphite in the Graph-Air) and deep radial cracks in the Graph-Air after 5 minutes at 250 lbs net bearing force; see enclosure (6), pages 7, 8 and 9.

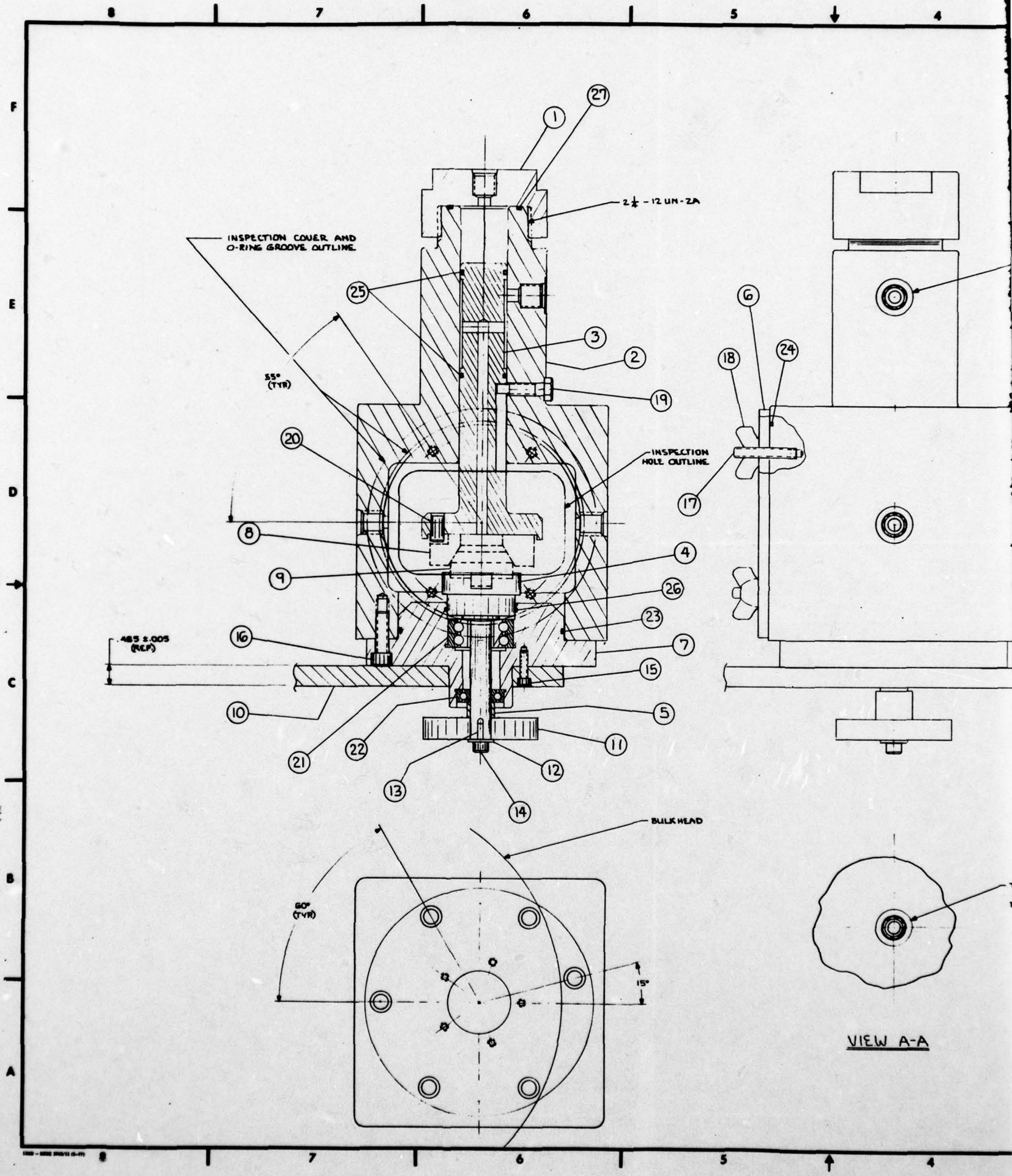
[redacted]

LJM:pfw
Ser 611/61/78
10 April 1978

4. The matrix shown in enclosure (4) summarizes the various materials tried and the results. The final conclusion is that the original tool steel head and bronze block are the optimum materials. It is recommended that heat treating the tool steel head to a slightly lesser hardness (R_c 50 to 53 instead of R_c 60 to 62) will improve its toughness and eliminate the surface cracking without degrading its wear properties.

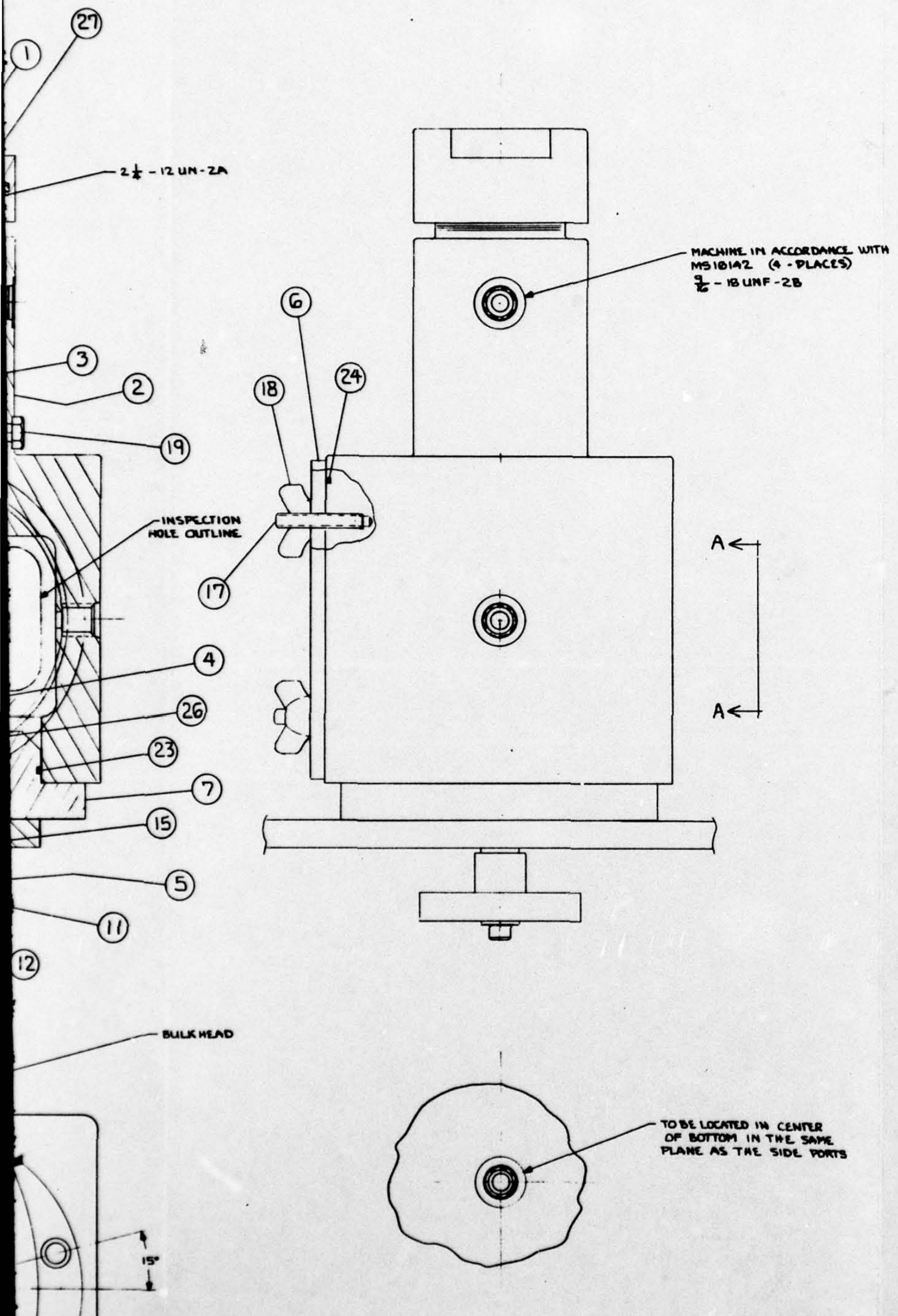
L. J. MARTINI

Copy to: w/o encls:
0608
Gould (J. Taburiaux)



INSPECTION COVER AND
O-RING GROOVE OUTLINE
55°
(TYP)
$2\frac{1}{4}$ - 12 UN - 2A
INSPECTION
HOLE OUTLINE
.485 ±.005
(REF)
BULKHEAD
60°
(TYP)
15°
VIEW A-A

REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVAL
3-C	A	ITEM #16 LENGTH CHANGED 1¼ TO 1	1-4-78	[illegible]



VIEW A-A

QTY	REQ	FSCM NO.	PART OF IDENTIFYING NUMBER	DESCRIPTION OR NOMENCLATURE	MATERIAL/SPEC/REF DESIG	ITEM NO.
1			2-030	PREFORMED PACKING		27
1			2-031	PREFORMED PACKING		26
2			2-212	PREFORMED PACKING		25
1			2-048	PREFORMED PACKING		24
1			2-154	PREFORMED PACKING		23
1			# 29R8	NDH BEARING	SINGLE ROW	22
1			# 5203	NDH BEARING	DOUBLE ROW	21
1				PIN	3/16 DIA. X 5/8	20
1			NUCO5610	STOP, PISTON	SEELBOLT TYPE SB AN	19
4				NUT, WING	1/4 - 20UNC-2B	18
4				STUD	1/4 - 20UNC-2A X 1½	17
Ⓐ 6				SOCKET HEAD CAP SCREW	3/8 - 16UNC-2A X 1	16
5			NAS1351-3-8			15
1			MS16998-41	SOCKET HEAD CAP SCREW	MK-46	14
1			MS20066-116	KEY, SQUARE	MK-46	13
1			1557061	WASHER	MK-46	12
1			1568895	GEAR, PUMP	MK-46	11
1				BULKHEAD, TEST		10
1			NUCO5613	TEST PIECE, BLOCK		9
1			NUCO5613	TEST PIECE, HEAD		8
1			NUCO5612	HOUSING, BEARING		7
1			NUCO5612	COVER, INSPECTION	17-4 PH	6
1			NUCO5611	SPACER	300 CRES	5
1			NUCO5612	SHAFT	17-4 PH	4
1			NUCO5612	PISTON	17-4 PH	3
1			NUCO5612	BODY	AL. ALY	2
1			NUCO5612	CAP	AL. ALY	1

PARTS LIST

INTERPRET DRAWING IN ACCORDANCE WITH MIL-STD-100

DO NOT SCALE THIS DRAWING

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES. BREAK ALL EDGES MAX. REMOVE ALL BURRS. TOLERANCES ARE: .XX ± .XXX ± ANGLES ± FILLETS MAX. SURFACE ROUGHNESS

NEXT ASSY | USED ON

APPLICATION

RELEASED DATE

APPROVED

APPROVED

CHECKED

PREPARED R. HENRY 12-14-77

PROJECT NUMBER

COGNIZANT CODE

APPROVED FOR

BY DIRECTION

NAVAL OCEAN SYSTEMS CENTER
SAN DIEGO, CA 92152

LAYOUT, TEST FIXTURE TO TEST HEAD & BLOCK MATERIALS FOR VARIABLE DISPLACEMENT PUMP

SIZE F | FSCM NUMBER 53910 | DRAWING NUMBER NUCO5614

SCALE | UNIT WT | SHEET OF

DWG NO NUCO5614 SHT

Enclosure 1

2

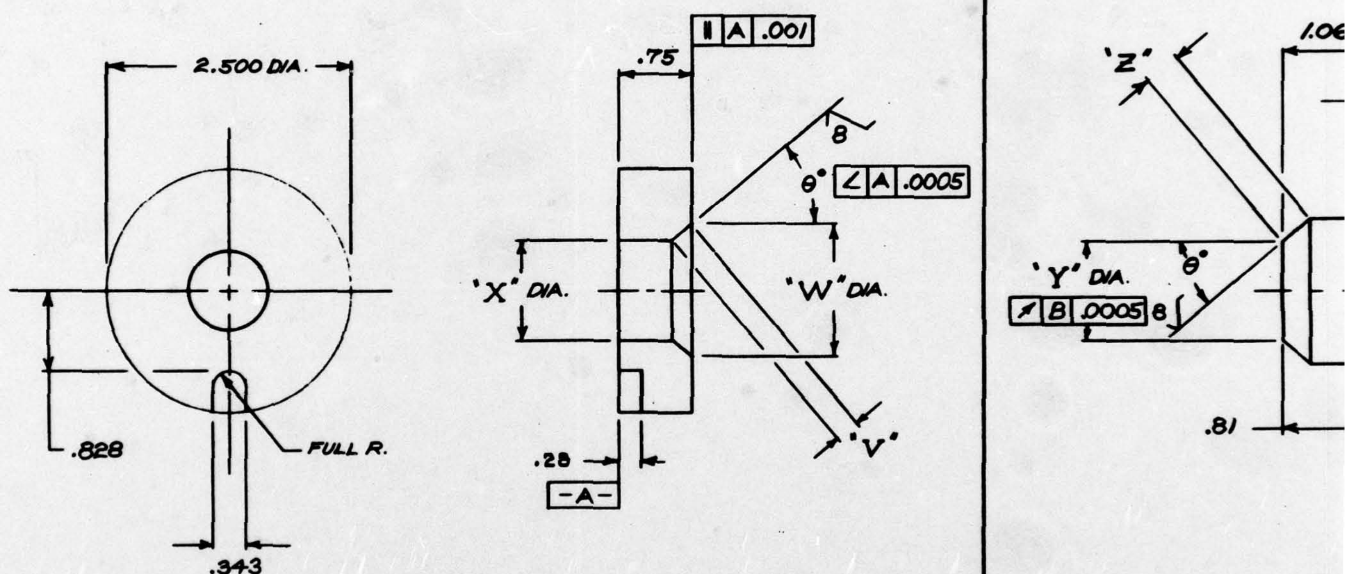
NOTES:

1. GRAPH AIR: HEAT TREAT TO RC 60/62 BY HEAT TREAT TO 1470°F, AIR COOL TO UNDER 100°F & TEMPER AT 300°F TO 400°F
2. TOOL STEEL 52100: HEAT TREAT IN ACCORDANCE WITH MIL-H-6875 TO RC 60/62
3. NITRONIC 60: ANNEAL @ 1950°F FOLLOWED BY RAPID COOLING

MATERIAL

DASH LETTER	BLOCK	HEAD
A	GRAPH AIR	52100 TOOL STEEL
B	GRAPH AIR	GRAPH AIR
C	NITRONIC 60	52100 TOOL STEEL
D	BRONZE 660	52100 TOOL STEEL
E	BEARIUM	52100 TOOL STEEL

"BLOCK"



4	.255 ±.005	1.368 REF.	1.113 ±.001	1.013 +.000 −.030	.486 REF.	30° ± 0°-1'
3	.255 ±.005	1.368 REF.	1.007 ±.001	.900 +.000 −.030	.424 REF.	45° ± 0°-1'
2	.255 ±.005	1.368 REF.	1.076 ±.001	.976 +.000 −.030	.456 REF.	35° ± 0°-1'
1	.255 ±.005	1.368 REF.	1.040 ±.001	.940 +.000 −.030	.435 REF.	40° ± 0°-1'
DASH NO.	V	W	X	Y	Z	θ SEE NOTE

QTY	REQ	FSCM NO.	PART OR IDENTIFYING NUMBER

NUC05614

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE THIS DRAWING

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN
BREAK ALL EDGES MAX.
REMOVE ALL BURRS
TOLERANCES ARE:
.XX ± .03 .XXX ± .015
ANGLES ±
FILLETS, MAX.
SURFACE ROUGHNESS 125

INTERPRET DR
IN ACCORDANC
- MIL-STD-1

PROJ NO.
COG CODE
REL DATE

1100-0002 2000/3 (6-77)

[redacted]

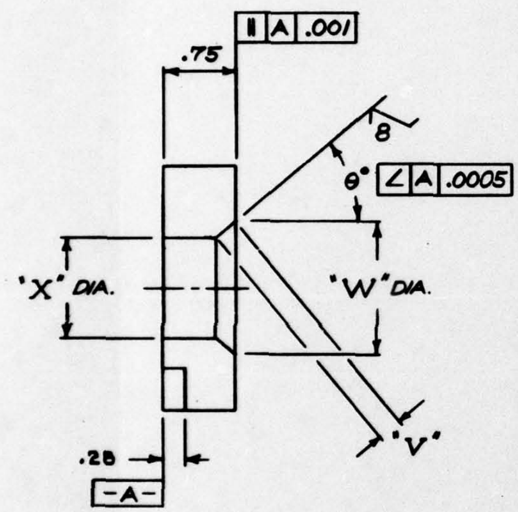
TO RC 60/62 BY
AIR COOL TO
T 300°F TO 400°F
T TREAT IN
-6875 TO RC 60/62
,@ 1950°F FOLLOWED

REVISIONS				
ZONE	LTR	DESCRIPTION	DATE	APPROVAL

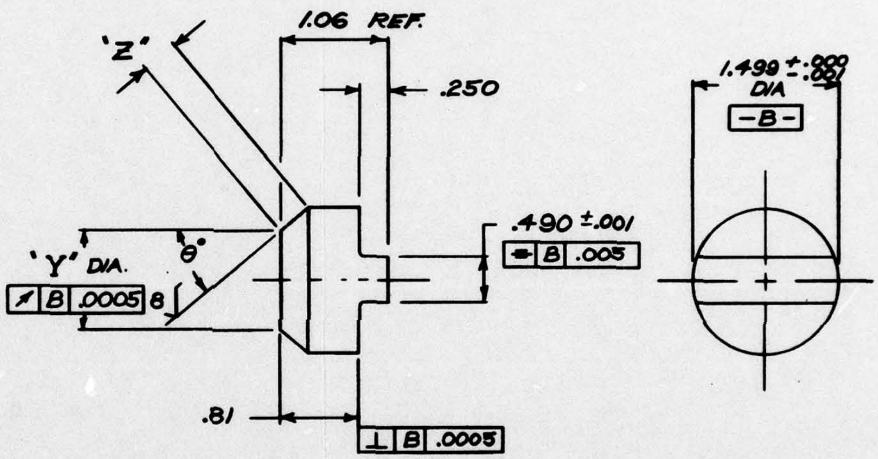
MATERIAL

DASH LETTER	BLOCK	HEAD
A	GRAPH AIR	52100 TOOL STEEL
B	GRAPH AIR	GRAPH AIR
C	NITRONIC 60	52100 TOOL STEEL
D	BRONZE 660	52100 TOOL STEEL
E	BEARIUM	52100 TOOL STEEL

K"



"HEAD"



NOTE: ANGLED SURFACES TO BE LAPPED TOGETHER.

…H NO.	V	W	X	Y	Z	θ SEE NOTE
4	.255 ±.005	1.368 REF	1.113 ±.001	1.018 +.000 −.030	.486 REF	30° ± 0°-1'
3	.255 ±.005	1.368 REF	1.007 ±.001	.900 +.000 −.030	.424 REF	45° ± 0°-1'
2	.255 ±.005	1.368 REF	1.076 ±.001	.976 +.000 −.030	.456 REF	35° ± 0°-1'
1	.255 ±.005	1368 REF	1.040 ±.001	.940 +.000 −.030	.435 REF	40° ± 0°-1'

QTY REQ	FSCM NO.	PART OR IDENTIFYING NUMBER	DESCRIPTION OR NOMENCLATURE	MATERIAL/SPEC/REF DESG	ITEM NO.

PARTS LIST

NEXT ASSY	USED ON
NUC05614	

APPLICATION

DO NOT SCALE THIS DRAWING
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN
BREAK ALL EDGES MAX.
REMOVE ALL BURRS
TOLERANCES ARE:
.XX ± .03 .XXX ± .015
ANGLES ±
FILLETS, MAX.
SURFACE ROUGHNESS 125

INTERPRET DRAWING IN ACCORDANCE WITH - MIL-STD-100

APVD
CHKD
PREP

PROJ NO.
COG CODE
REL DATE

APPROVED FOR
BY DIRECTION

NAVAL OCEAN SYSTEMS CENTER
SAN DIEGO, CA 92152

TEST PIECES
BLOCK AND HEAD

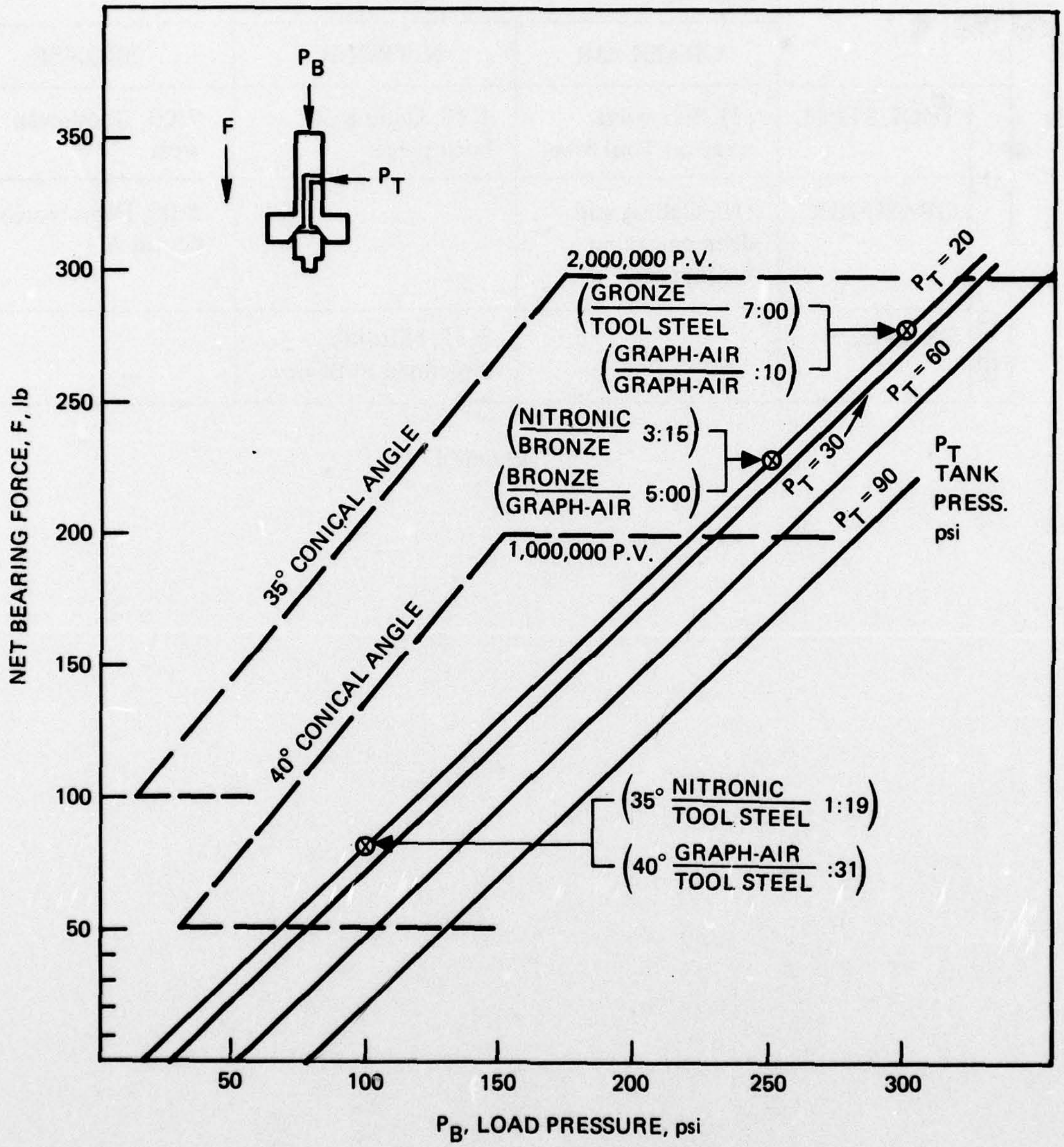
SIZE	FSCM NUMBER	DRAWING NUMBER
C	55910	NUC05613

SCALE UNIT WT SHEET OF

DWG NO. NUC05613 SHT

Enclosure 2

TEST FIXTURE FORCE ANALYSIS (3 February 1978)



Enclosure 3.

[redacted]

RESULTS MATRIX

		BLOCK		
		GRAPH-AIR	NITRONIC	BRONZE
HEAD	TOOL STEEL	:31, Excessive wear on Tool Steel	1:19, Galling on both pieces	7:00, Good even wear
	GRAPH-AIR	:10, Galling and deep cracks on both pieces		5:00, Deep cracks in Graph-Air
	BRONZE		3:15, Nitronic impedded in Bronze.	

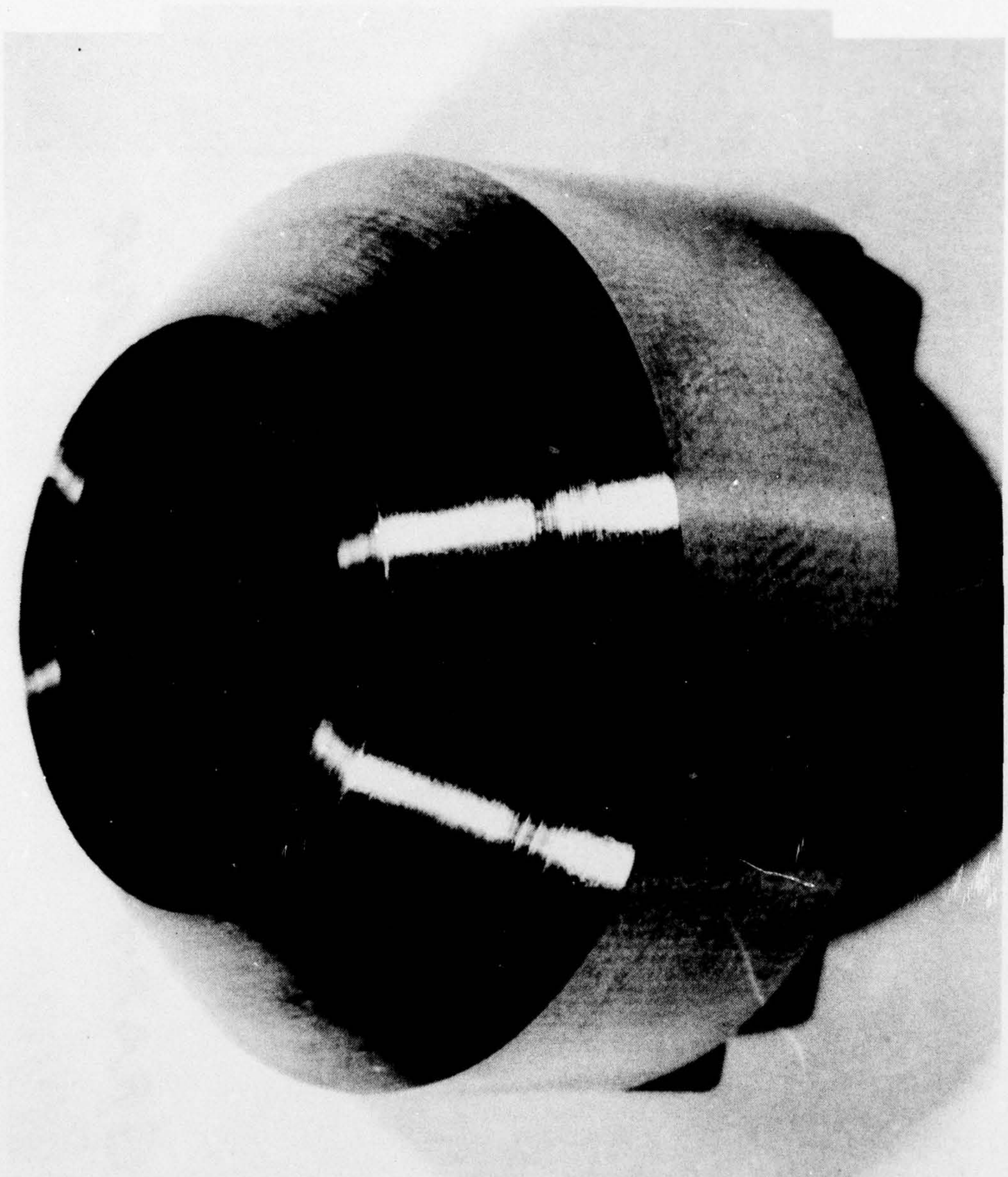
Enclosure 4.

TEST FIXTURE TESTS

Date	Cylinder Block	Pump Head	Bias Pressure P_B, psi	Tank Pressure P_T, psi	Run Time min:sec	Comments
20 Jan '78	Graph-Air (A-1)	52100 Tool Steel (A-1)	60	60	0:00	
			95	17	0:31	Pump head wore badly.
28 Feb '78	Bronze (40°)	Tool Steel (40°)	80	18	0:15	Run-in time.
			80	18	1:00	Inspect surfaces ok.
			149	14	1:42	Surfaces look good, continue.
			200	19	2:03	Surfaces look good, continue.
			250	19	3:00	Surfaces look good, continue.
			250	19	3:23	Surfaces look good, continue.
24 Feb '78	Bronze (40°)	Tool Steel (40°)	240	19	–	
			250	20	7:23	Surfaces look ok, continue.
			305	18	9:23	Slight leakage, continue.
			300	18	14:23	Cracks on pump head.
24 Feb '78	Graph-Air (40°)	Graph-Air (40°)	300	20	0:10	Excessive wear and cracks on block and head.
24 Feb '78	Nitronic (40°)	Bronze (40°)	90	20	1:00	Run-in time.
			200	20	1:15	Surfaces seated.
			250	20	2:00	Started leaking. Nitronic imbedded in Bronze.
24 Feb '78	Nitronic (35°)	Tool Steel (35°)	100	20	0:10	Run-in time.
			100	20	1:00	Slight galling.
			100	0	1:09	Vibration and leakage. Galled surfaces.
14 March '78	Bronze (40°)	Graph-Air (40°)	100	20	:30	Surfaces look ok, continue.
			100	20	1:00	Surfaces look ok, continue.
			100	20	2:00	Surfaces look ok, continue.
			150	20	3:00	Scratches on Bronze block.
			200	20	4:00	Slight leakage.
			250	20	5:00	Leakage. Bronze chips noticed.
			250	20	9:00	Excessive wear of Bronze block, and cracks on Graph-Air Head.

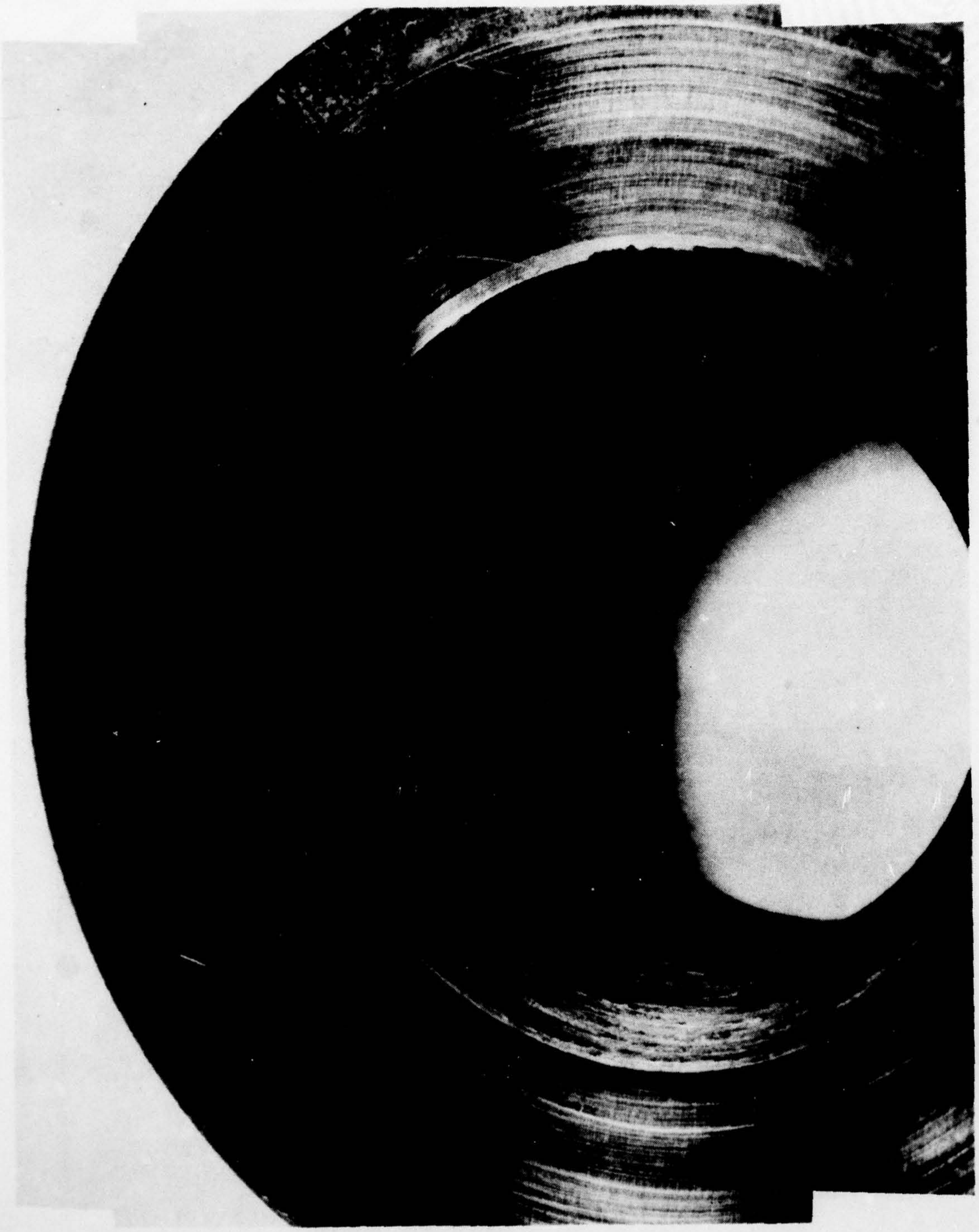
Enclosure 5.

Enclosure 6-1.

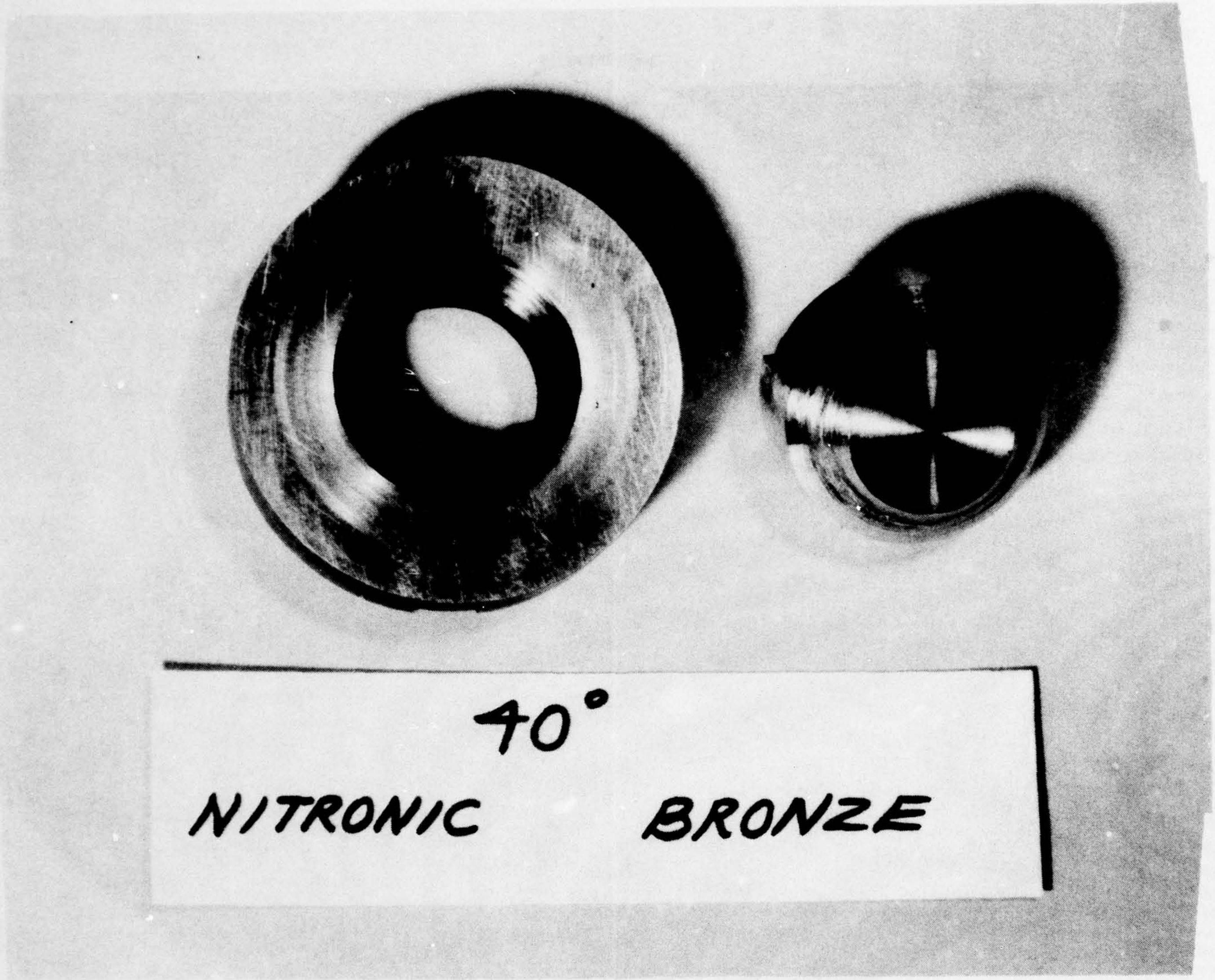


Enclosure 6-2.

Enclosure 6-3.



Enclosure 6-4.



Enclosure 6-5.

AD-A064 625 NAVAL OCEAN SYSTEMS CENTER SAN DIEGO CA F/G 19/8
VARIABLE-DISPLACEMENT FUEL PUMP DESIGNED FOR ALWT.(U)
SEP 78 I J MARTINI
UNCLASSIFIED NOSC/TR-308 NL

2 OF 2
AD
A064625




END
DATE
FILMED
4-79
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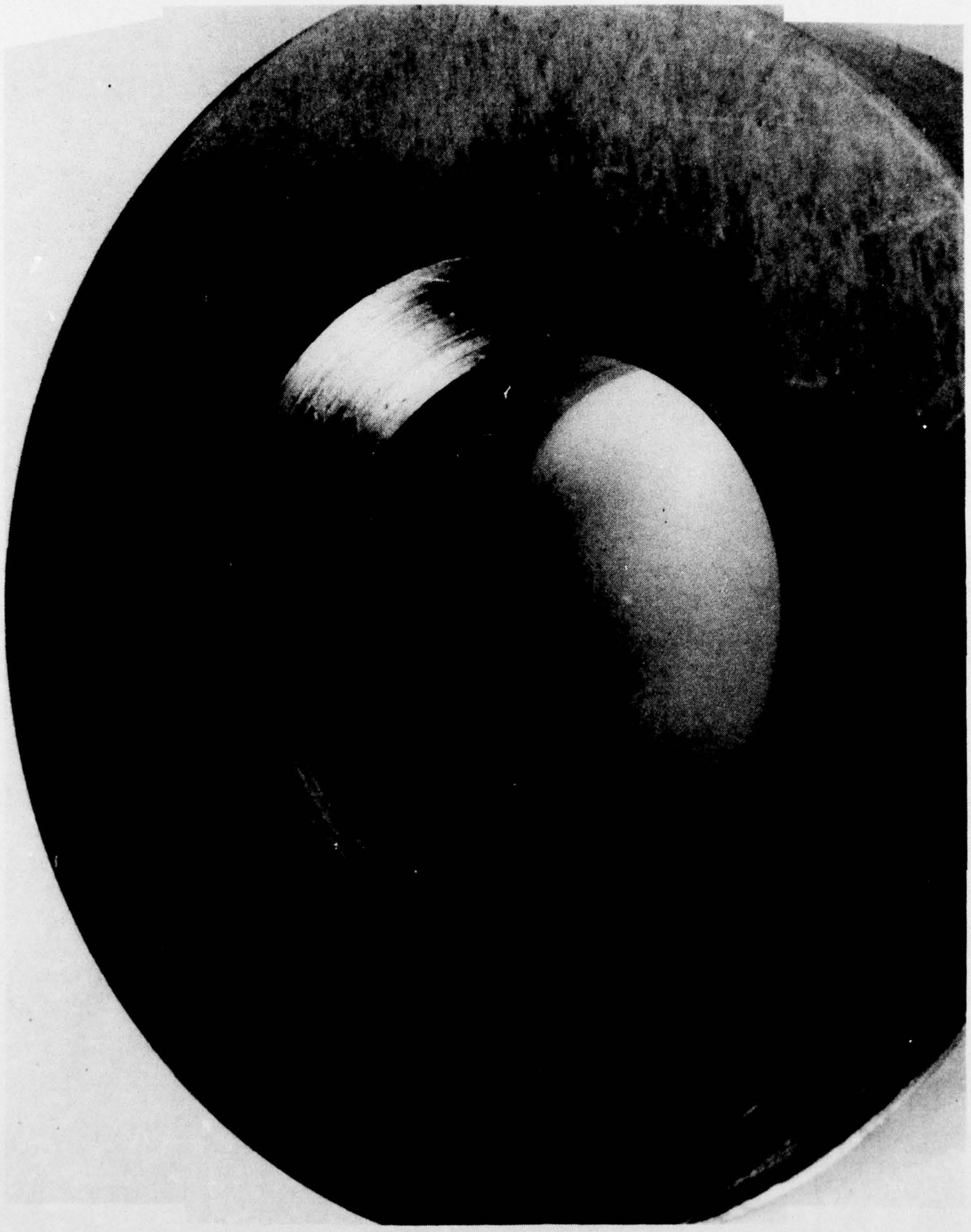
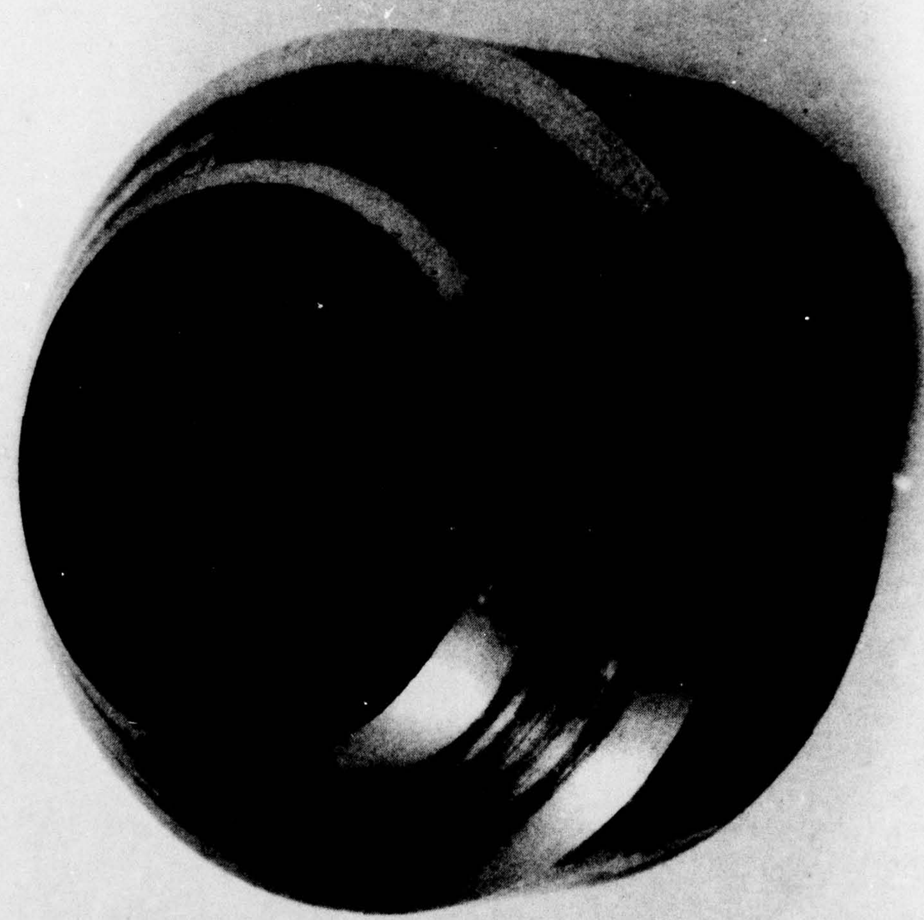
Enclosure 6-6.

Enclosure 6-7.



Enclosure 6-8.



Enclosure 6-9.