REPORT NO. 312-18

FINAL REPORT FOR THE DEVELOPMENT PROGRAM OF THE 05-055 HYDRAULIC TRANSFORMER UNDER NASA CONTRACT NO. NAS 8-5264

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TABLE OF CONTENTS

Section No.	Paragraph No.		Description	Page No.
1.0			SUMMARY	1-1
2.0			INTRODUCTION	2-1
3.0			DESCRIPTION OF UNIT	3-1
	3.1		Description of Operation	3-3
	3.2		Performance Characteristics	3-6
	3.3		Description of Components	3-8
		3.3.1	Rotary Valve and Sleeve	3-8
		3.3.2	Pistons, Sleeves and Dashpots	3-13
		3.3.3	Oil Side Check Valves	3-17
		3.3.4	Hydraulic Motor Drive	3-20
		3.3.5	Main Housing Assembly	3-23
4.0			DEVELOPMENT HISTORY	4-1
	4.1		Analysis and Design	4-1
	4.2		Component Design and Development	4-2
		4.2.1	Hydraulic Motor Tests	4-3
		4.2.2	Rotary Valve Development	4-5
	4.3		Test Equipment	4-18
	4.4		Prototype Fabrication and Testing	4-23
5.0			PERFORMANCE ANALYSIS	5-1
	5.1		Piston Analysis	5-1
	5.2		Rotary Valve Operation	5-8
	5.3		Check Valve Operation	5-15

TABLE OF CONTENTS (Continued)

Section No.	Paragrapn No.	Description	Page No.
6.0		PERFORMANCE TEST RESULTS	6-1
	6.1	Rotary Valve Tests	6-1
	6.2	Piston Tests	6-5
	6.3	Prototype Unit Tests	6-7
	6.4	Final Disassembly and Inspection of Prototype Unit	6-17

1.0 SUMMARY

This report describes the design, development, fabrication and testing of a prototype Hydraulic Transformer, Hydro-Aire Model No. 05-055, performed in fulfillment of the requirements of Contract No. NAS 8-5264 for NASA Marshall Space Flight Center. The Hydraulic Transformer described is designed to pump hydraulic oil at a flow of 100 GPM with a pressure rise of 4000 psi, and does this work by utilizing as a power source the flow of RP-1 rocket fuel at a pressure of 1900 psig.

The Hydraulic Transformer is described in detail with individual descriptions and illustrations of the unit components. The principle of operation is described and illustrated.

A performance analysis is presented with a mathematical evaluation of the component operating parameters. The development history records the steps in reaching satisfactory component operation leading to fabrication of the prototype Hydraulic Transformer.

The performance test results of the prototype Hydraulic Transformer confirm the general feasibility of the design and the principle of operation. However, the housings of both units of the Hydraulic Transformer were cracked during testing due to a

over

weakness of the housing as well as excessive pressure surges originating in the test facility. Performance curves are given of the Hydraulic Transformer operation.

2.0 INTRODUCTION

The Hydro-Aire Hydraulic Transformer, Part Number 05-055, was developed under NASA Contract No. NAS 8-5264 to satisfy the requirements of NASA Specification M-P&VE-PLB (Revision A), dated September 13, 1962. This report describes the design, development, fabrication, and testing of the prototype Hydraulic Transformer.

The specification requirements, initially related to the Saturn S-IC Vehicle, call for a pump with a variable capacity of zero to 100 GPM of hydraulic oil at a rated pressure of 4000 psi. The power to accomplish this work is to be derived from the RP-1 rocket fuel at a pressure of 1900 psig. One conventional approach to this problem would be to design a fuel-driven motor coupled to a hydraulic pump. Due to the large flows and high pressures involved, such an approach would necessarily result in a sizable unit of considerable weight.

Instead of this conventional approach, a decision was made to design a hydraulic transformer, or intensifier, which combines the functions of hydraulic motor and pump in a single unit. Basically, the hydraulic transformer consists of a series of free pistons moving in individual cylinders under the influence

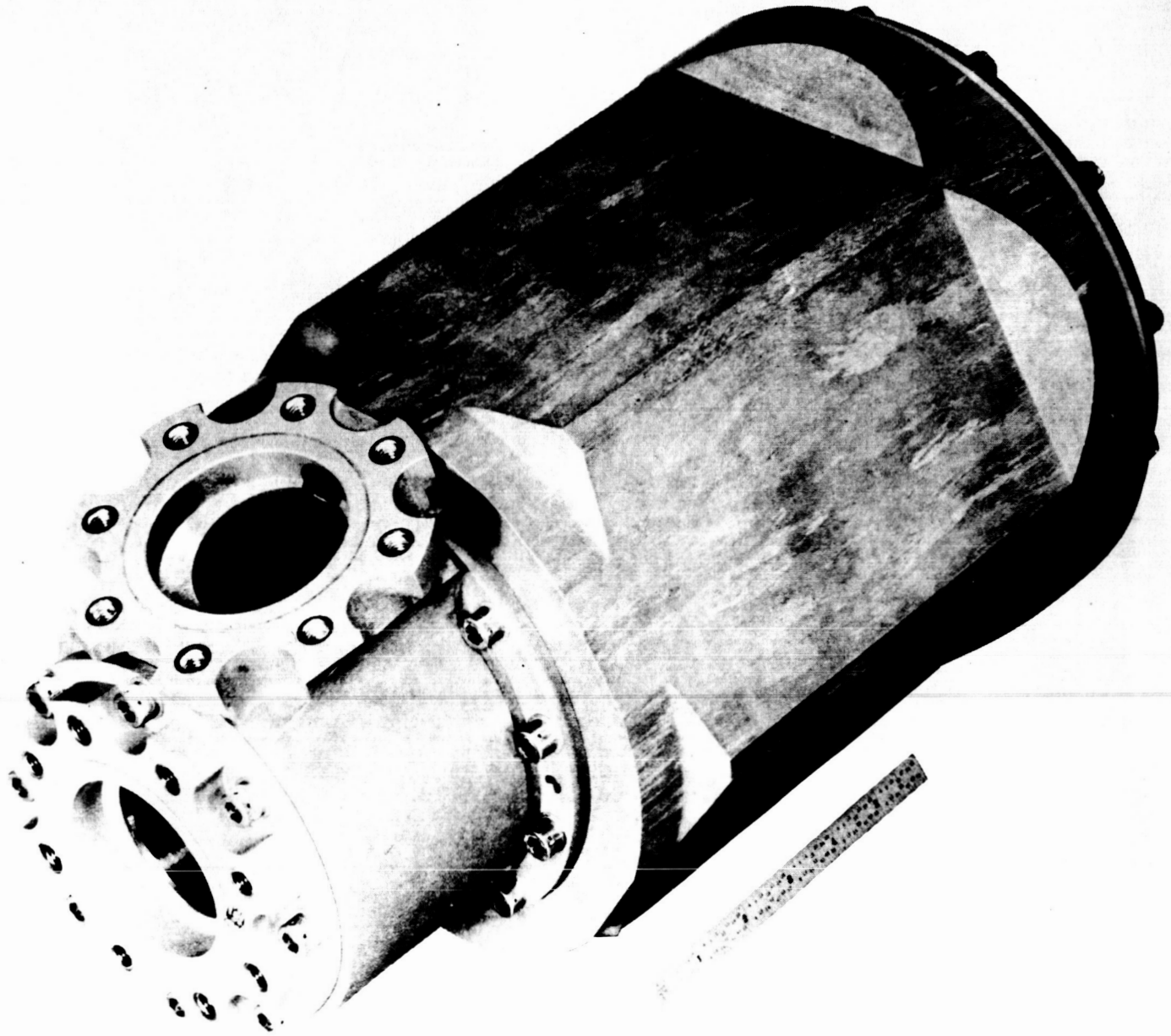
of the pressure of each fluid, the rocket fuel and the hydraulic oil, acting on opposite ends of the pistons. The fact that each piston has a larger fuel side area than oil side area accounts for the intensification of pressure of the hydraulic oil, which is pumped by the lower pressure fuel.

The advantages of the hydraulic transformer over a conventional motor-pump combination include more than a saving in weight. The relatively small number of moving parts reduces friction losses, and thus improves efficiency. The free piston construction means that pumping losses are reduced in proportion to the capacity, and thus efficiency drops relatively little at reduced flows. The hydraulic transformer is designed to give output characteristics of a pressure compensated variable displacement pump, producing flow from zero to rated output at a nearly constant output pressure and constant efficiency. The performance analysis presented in Section 5.0 shows how the characteristics of a free piston unit are adapted to this desired output.

The design of the Hydraulic Transformer went through a number of stages before arriving at the final configuration. These stages of development were recorded in the progress reports for the program, and are summarized in this report. The final

design of the present Hydraulic Transformer configuration is shown in Hydro-Aire Layout Drawing L05-054, and a photograph of the prototype unit is shown in Figure 2-1. Other photographs of individual parts are also presented further in the report.

The two prototype units built for this program were not tested at full rated conditions due to fracture of the housings during the test program. However, sufficient tests were run to verify general performance characteristics. The tests were run at lower than rated pressure levels due to initial difficulties with the test equipment, and the fracture of the housings occurred before the anticipated tests at full performance. Nevertheless, the principle of operation was verified and the trend of results indicates that the required full performance can be achieved with this design.

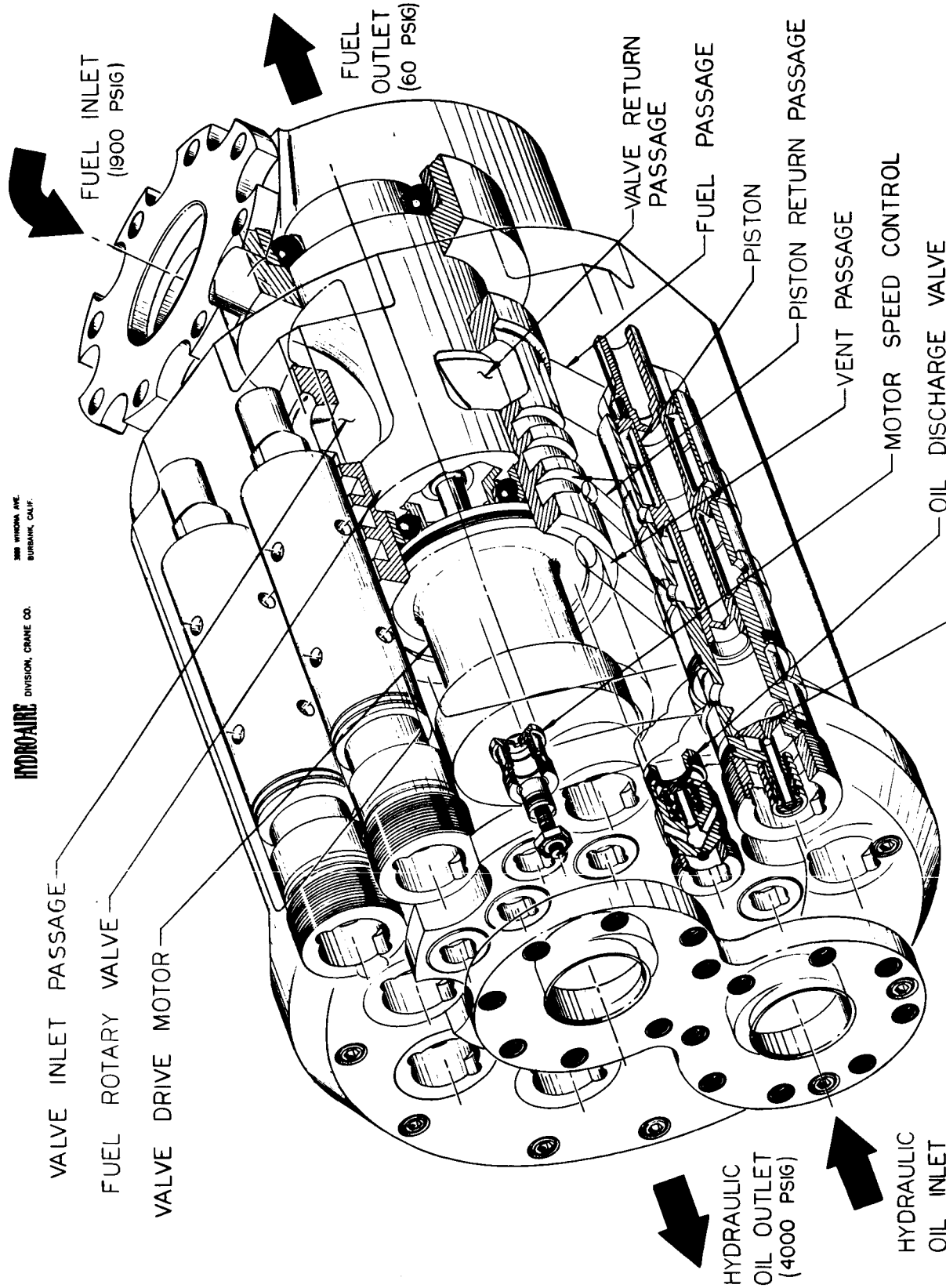


PART NO. 05-055 NEG. NO. _____ DESCRIPTION _____
NAME Prototype Hydraulic Transformer

Figure 2-1

3.0 DESCRIPTION OF UNIT

The requirements of NASA Specification M-P&VL-PLB call for a pump with a variable capacity of zero to 100 GPM of hydraulic oil at a rated pressure of 4000 psi. The power to accomplish this work is to be derived from the RP-1 rocket fuel at a nominal pressure of 1960 psig. Basically, the hydraulic transformer consists of a series of free pistons moving in individual cylinders under the influence of the pressure of each fluid, the rocket fuel and the hydraulic oil, acting on opposite ends of the pistons, as illustrated in Figure 3-1. The fact that each piston has a larger fuel side area than oil side area accounts for the intensification of pressure of the hydraulic oil, which is pumped by the lower pressure fuel. The hydraulic transformer exhibits output characteristics of a pressure compensated variable displacement pump, producing flow from zero to rated output at a nearly constant output pressure.



CUTAWAY VIEW OF HYDRAULIC TRANSFORMER

Figure 3-1

3.1 Description of Operation

A schematic of the hydraulic transformer is shown in Figure 3-2 to aid in understanding the functional description. The operation of the hydraulic transformer is as follows:

a) Fuel at 1900 psig enters the fuel inlet port. A small amount of fuel flows through a passage to a hydraulic motor and provides the fluid power for the hydraulic motor to turn the rotary valve. The bulk of the fuel flows through the rotary valve.

b) The rotary valve directs the fuel, in proper sequence, to the end surface of a set of free acting pistons.

c) The pistons are driven by the fuel pressure, acting on the fuel-side piston area, and thus pressurize hydraulic oil exposed to the smaller area at the oil-side end of the piston. This is the pressure intensifier action.

d) During the piston pressurizing stroke, the hydraulic oil in the pressurizing chamber at the small end of the piston is forced out under a pressure of 4000 psi through a check valve. At the end of the pressurizing stroke, the oil check valve closes.

e) When the piston thrust stroke is completed, the rotary valve supplies high pressure fuel to a mid-point area of the piston, forcing the piston to return and exhausting the fuel in the rear fuel-side chamber through the discharge ports of the rotary valve.

f) During the return stroke of the piston, a new quantity of hydraulic oil enters the oil pressurizing chamber from the oil inlet passages, flowing through an inlet check valve which closes when the return stroke is completed.

g) The hydraulic transformer has 17 pistons which perform the operations above in a sequence determined by the timing of the rotary valve, which in turn must rotate at the correct speed to allow for the dynamic response of the pistons and the oil check valves. A speed controller, on the motor which drives the rotary valve, assures proper rotational speed.

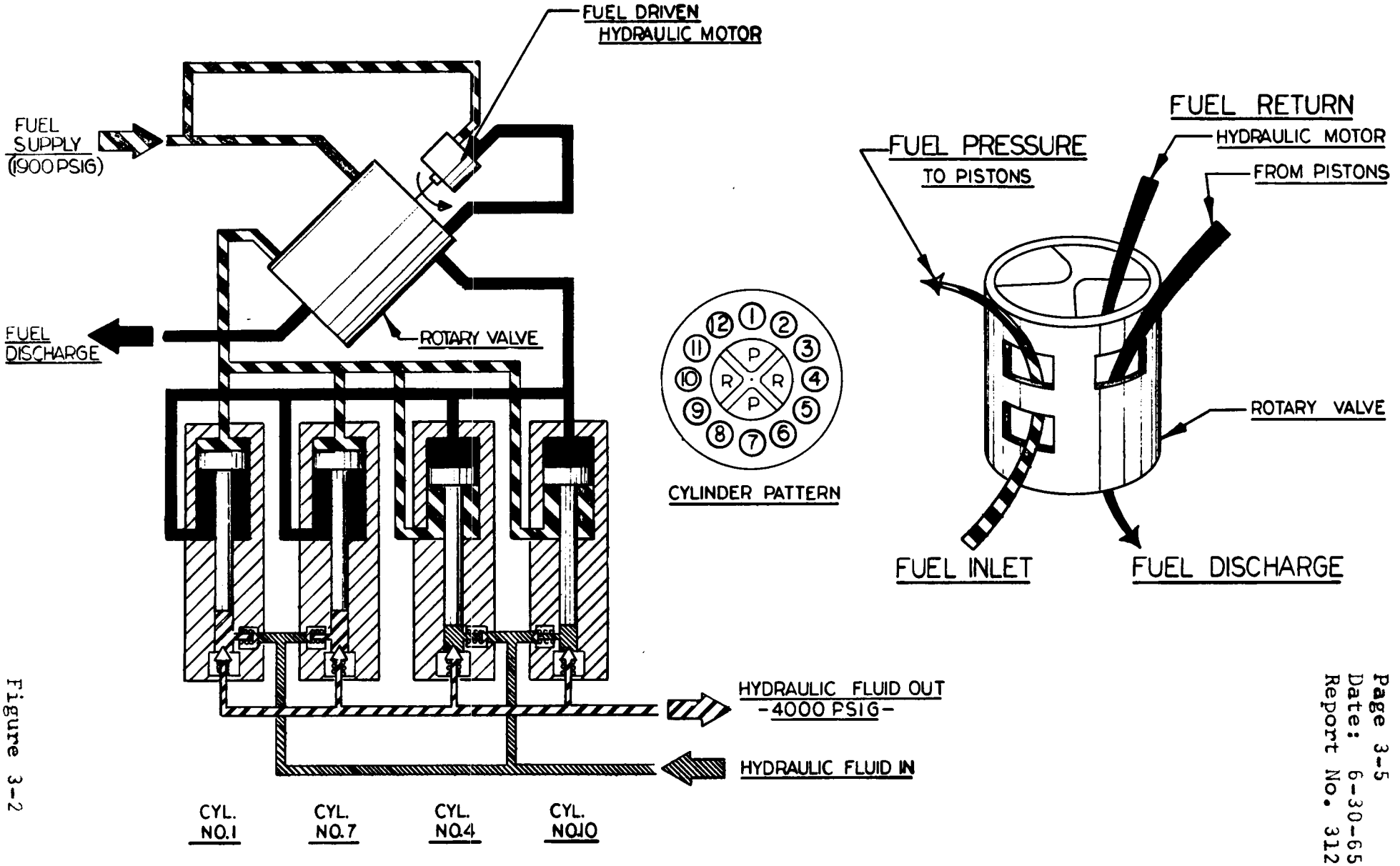


Figure 3-2

SCHEMATIC OF HYDRAULIC TRANSFORMER

3.2 Performance Characteristics

The prototype units built for this program were not tested at full rated conditions, due to fracture of the housings during the test program. However, sufficient tests were run to verify general performance characteristics. Figure 3-3 shows the results of one performance run. Tests were run at lower than rated pressure levels due to initial difficulties with the test equipment, and the fracture of the housings occurred before the anticipated tests at full performance. Nevertheless, the principle of operation was verified, and the trend of the results indicates that the required performance can be achieved. The performance tests are discussed in greater detail in Section 6.0.

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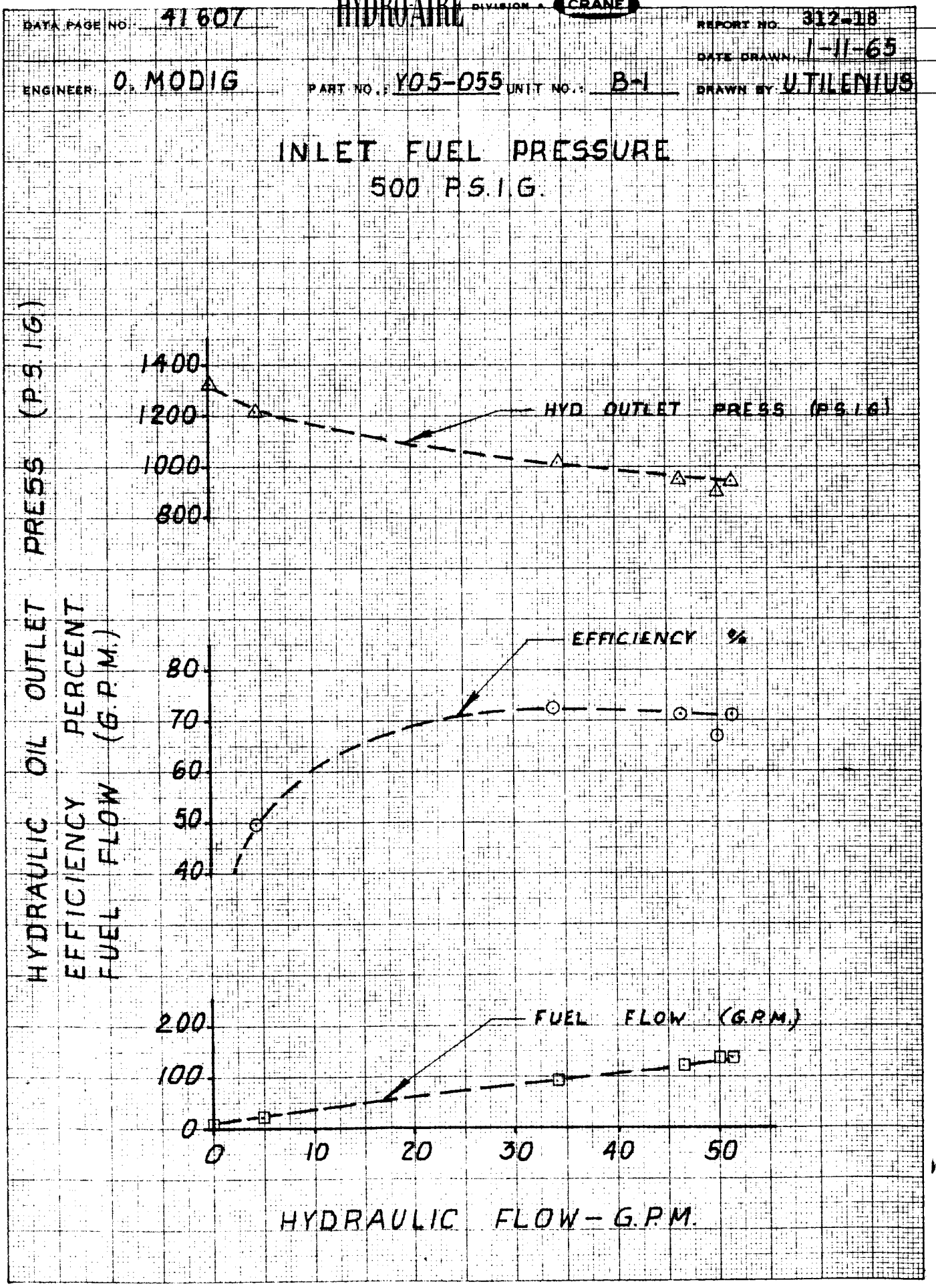


Figure 3-3

3.3 Description of Components

Much of the development work on the Hydraulic Transformer was done using test fixtures which allowed testing of individual components. The bulk of the development work was concentrated on two components: The fuel rotary valve and the pistons. The hydraulic motor and the check valves, on the other hand, required relatively little development effort. A complete parts array is shown in Figure 3-4.

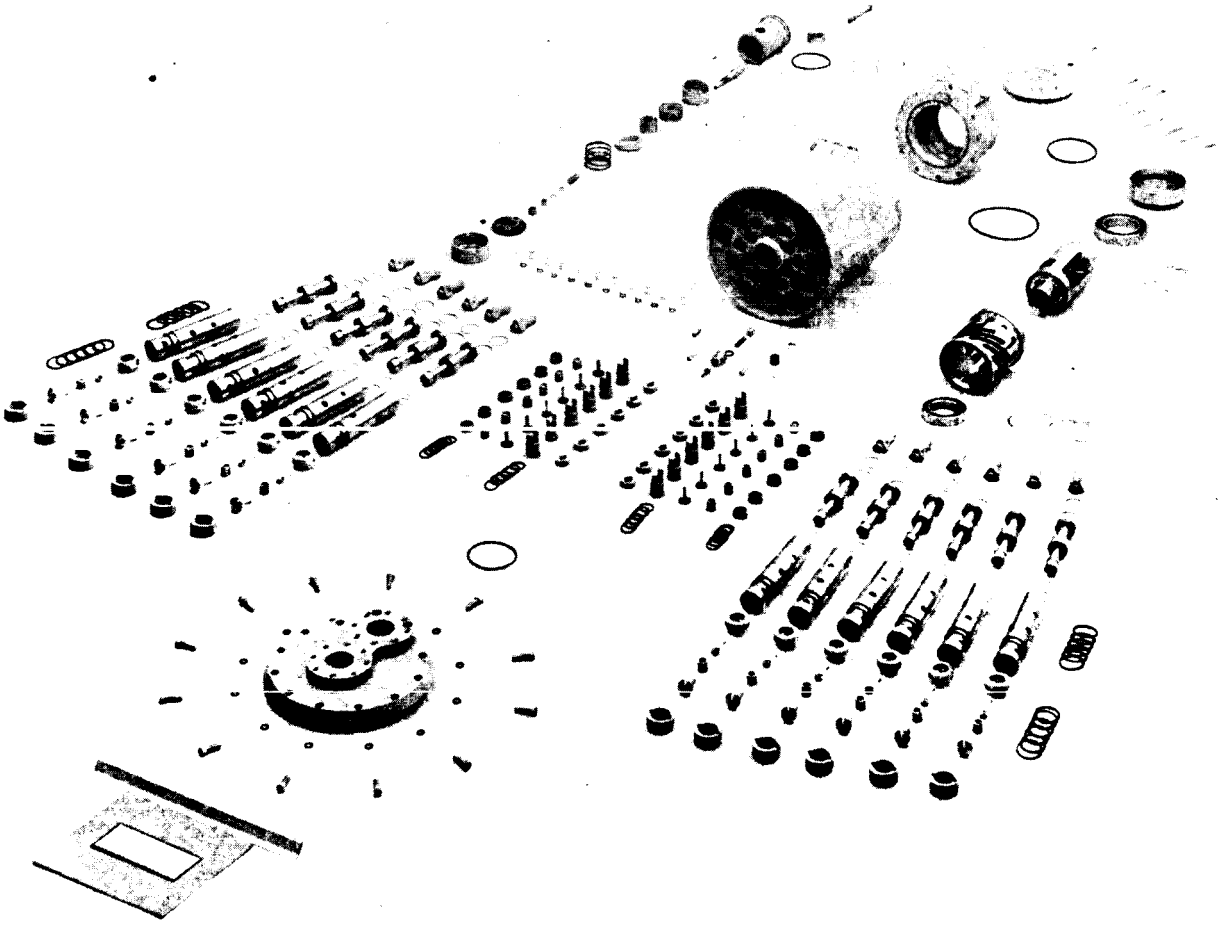
3.3.1 Rotary Valve and Sleeve

The rotary valve for the fuel flow required considerable development work because of the numerous factors involved in its design and fabrication. As this valve must handle a fuel flow of up to 260 GPM with minimum power losses, a compromise must be established between the torque required to rotate it and the leakage flow permissible. Several approaches were studied, and the cylindrical rotary valve design was selected as the most promising for development. A detailed discussion of rotary valve performance is presented in Section 5.2.

Several rotary valve configurations were tested to arrive at a design having low pressure drop and balanced pressure forces. The valve was also stiffened to give better rigidity and less

deformation in operation. With these improvements, the valve could be operated with very small running clearances between the valve body and the surrounding ported sleeve, thus minimizing leakage flow. The rotary valve is shown in Figure 3-5 and the main sleeve in Figure 3-6.

Another feature of the rotary valve requiring development work was choice of valve bearings. Initially, journal bearings were tried, but the resulting starting torque was too great for the hydraulic drive motor. A change was made to needle bearings and finally to ball bearings.



PART NO. 05-055 NEG. NO. _____ DESCRIPTION _____
NAME Complete Parts Array of Hydraulic Transformer

Figure 3-4

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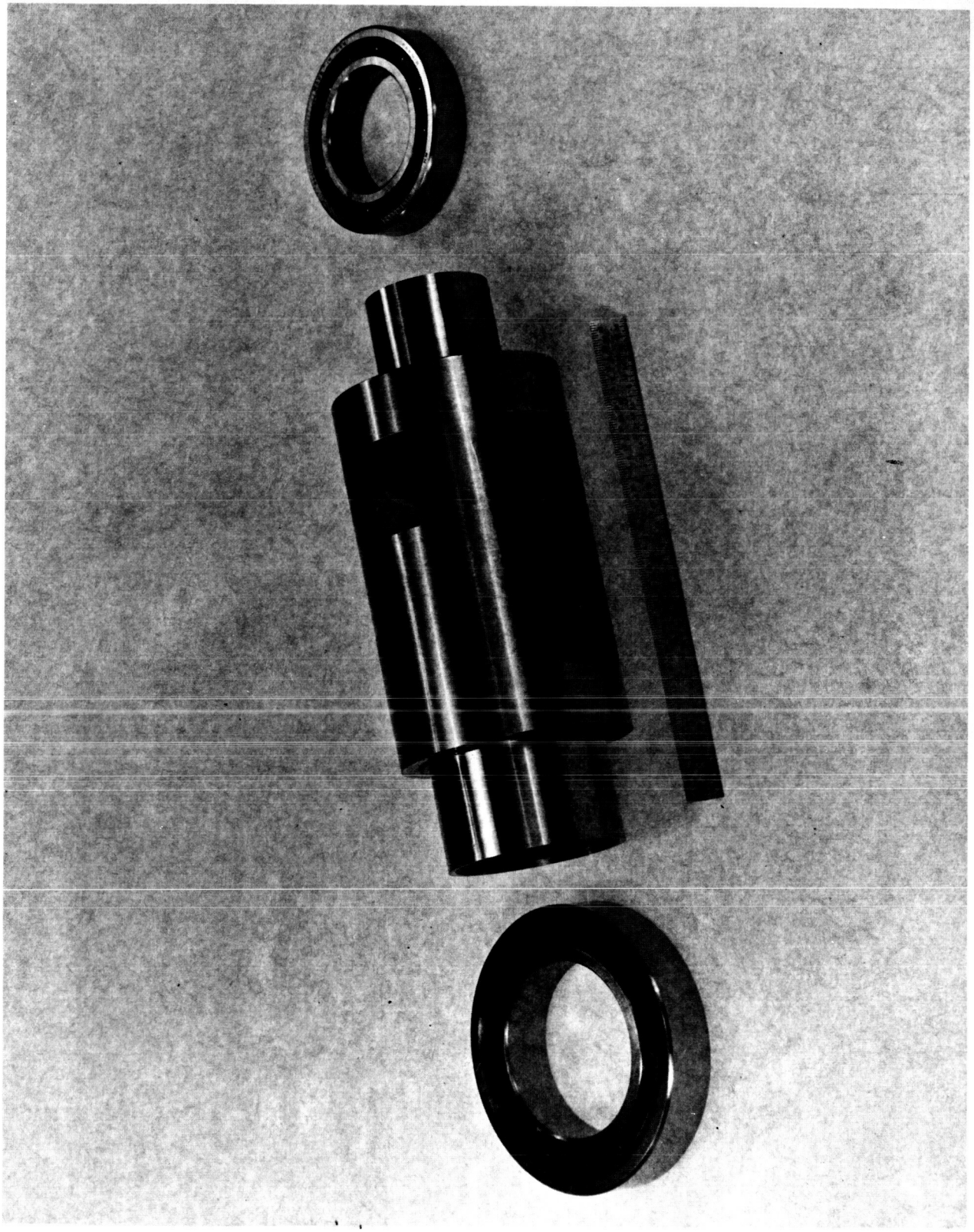
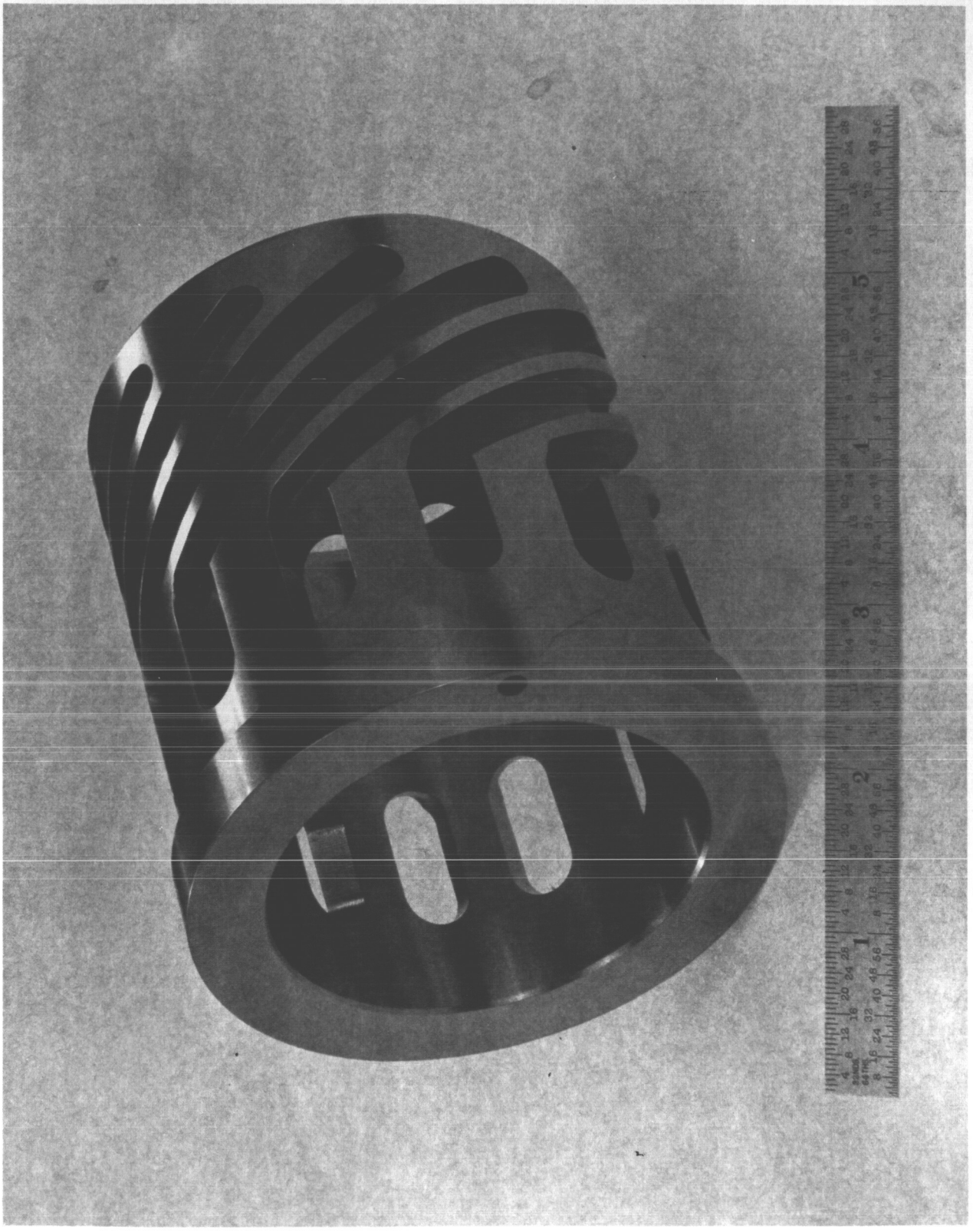


PART NO. Y05-055620 NEG. NO. _____ DESCRIPTION _____

NAME Rotary Fuel Valve

Figure 3-5



PART NO. Y05-055612 NEG. NO. _____ DESCRIPTION _____
NAME Main Sleeve for Fuel Distribution

Figure 3-6

3.3.2 Pistons, Sleeves and Dashpots

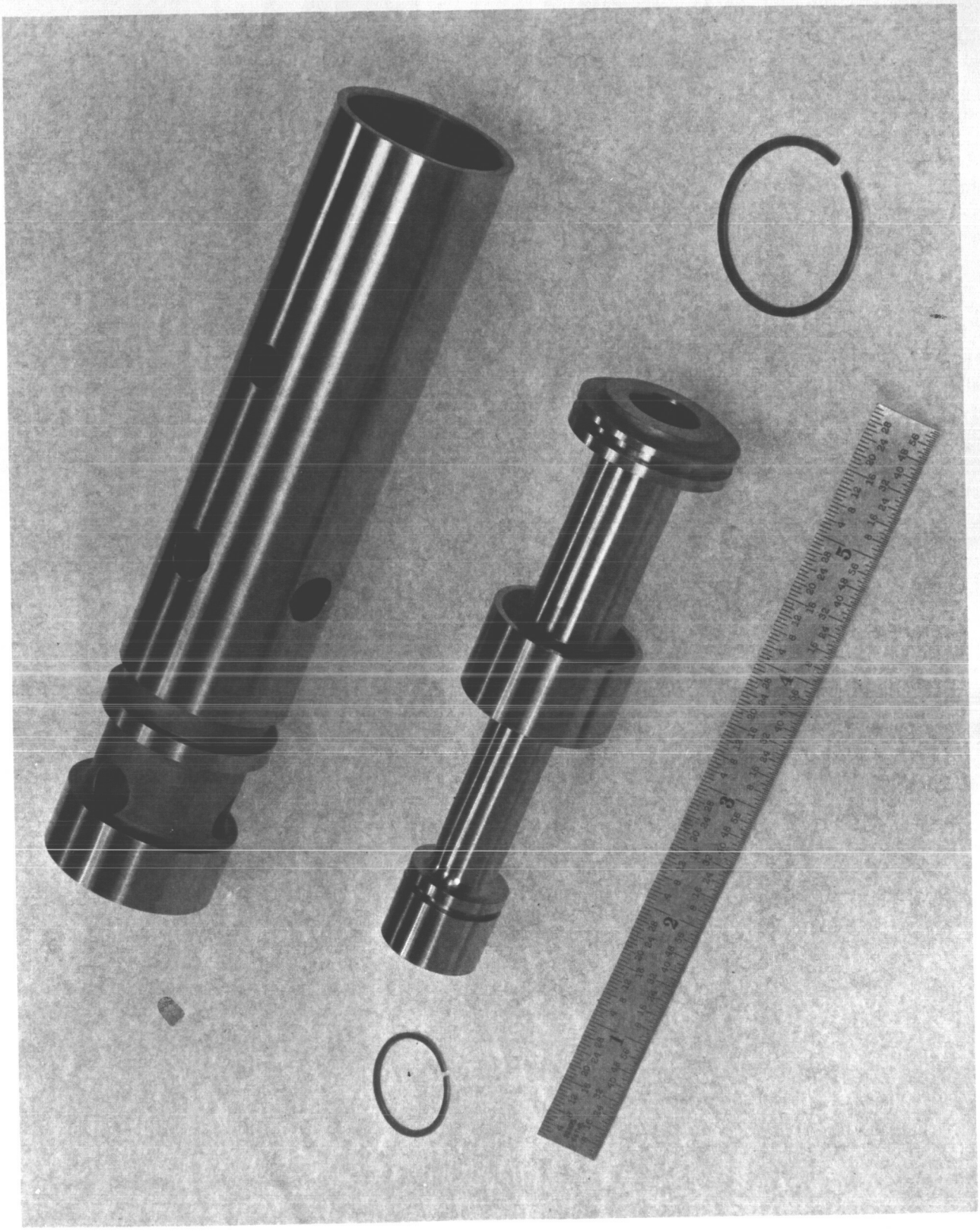
The pistons are the components of the Hydraulic Transformer which perform the pumping operation and the pressure intensification function. Requirements are that they be low in mass, to give fast action, yet strong enough to resist the considerable pressure forces. An analysis of piston action is presented in detail in Section 5.1.

In the course of the development work on the piston assembly, several alternative designs were studied. The original design had springs to return the pistons after the power stroke though this spring design was dropped after consideration of some inherent design problems, such as the high stress load on the springs. Therefore, it was decided to use hydraulic instead of mechanical power to return the pistons. Return action is accomplished by allowing high pressure fuel to enter a cavity in the middle of the piston assembly at the completion of the power stroke, which causes the piston to be returned.

However, this design required that three sealing surfaces be used per piston assembly, which presented some fabrication problems. It was finally found that based upon the use of a

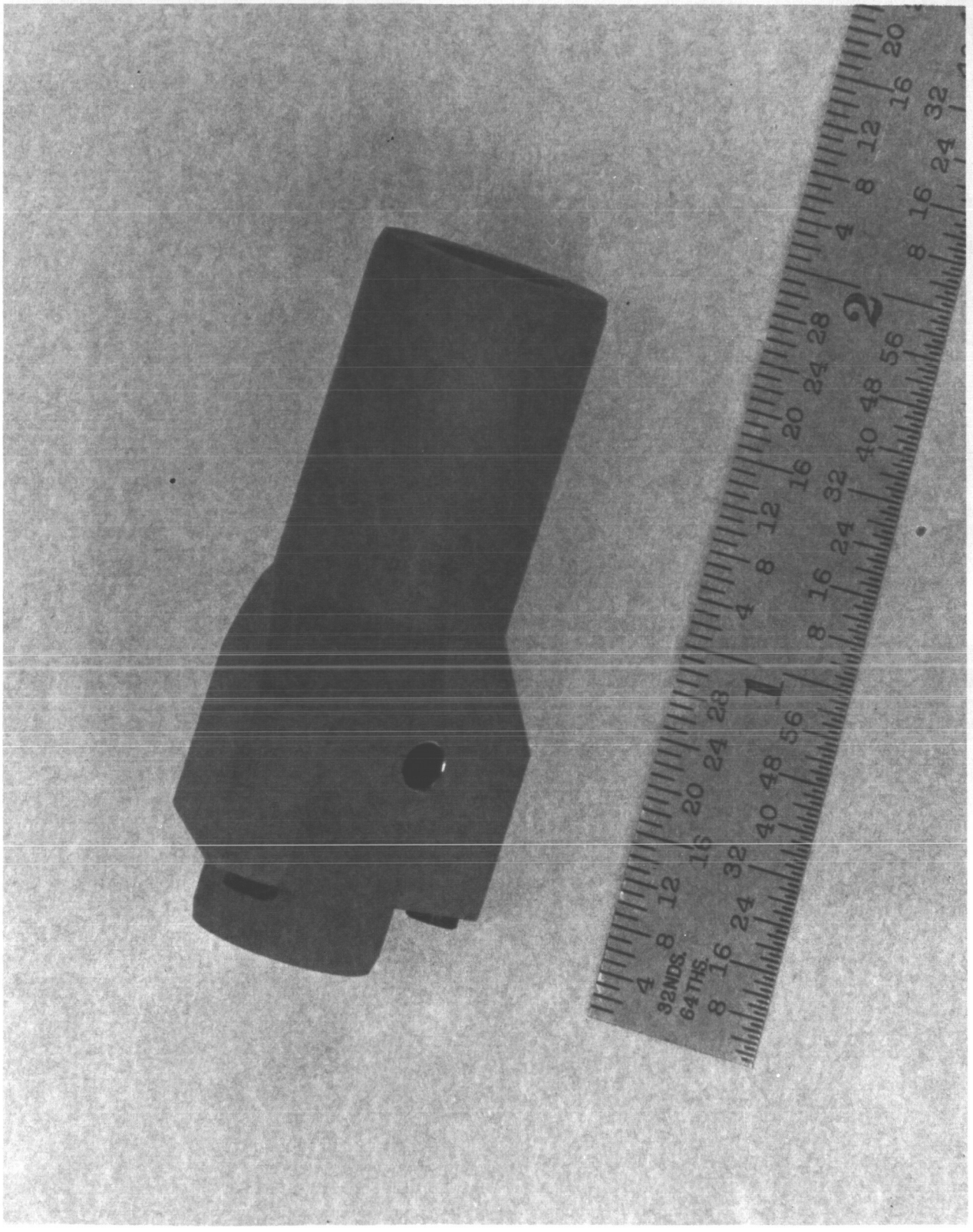
new precision grinding machine, the piston-plunger could be manufactured in one piece and still maintain the necessary diametrical clearances. Sealing is accomplished at the ends with piston rings. A piston and sleeve are shown in Figure 3-7.

The design of the piston assembly also raised some problems. The pistons travel at a considerable velocity and must be decelerated and stopped in a very short distance at the end of the stroke. The solution was rather straightforward for the oil pumping stroke, using the closed end of the cylinder as a dashpot. However, at the fuel end for the return stroke, a dashpot could not be constructed at the end of the cylinder bore. Instead, it was necessary to make the dashpot within the end of the piston, matching a mating projection at the end of the cylinder. A piston fuel dashpot is shown in Figure 3-8.



PART NO. Y05-055625 NEG. NO. _____ DESCRIPTION _____
NAME Piston and Piston Sleeve

Figure 3-7



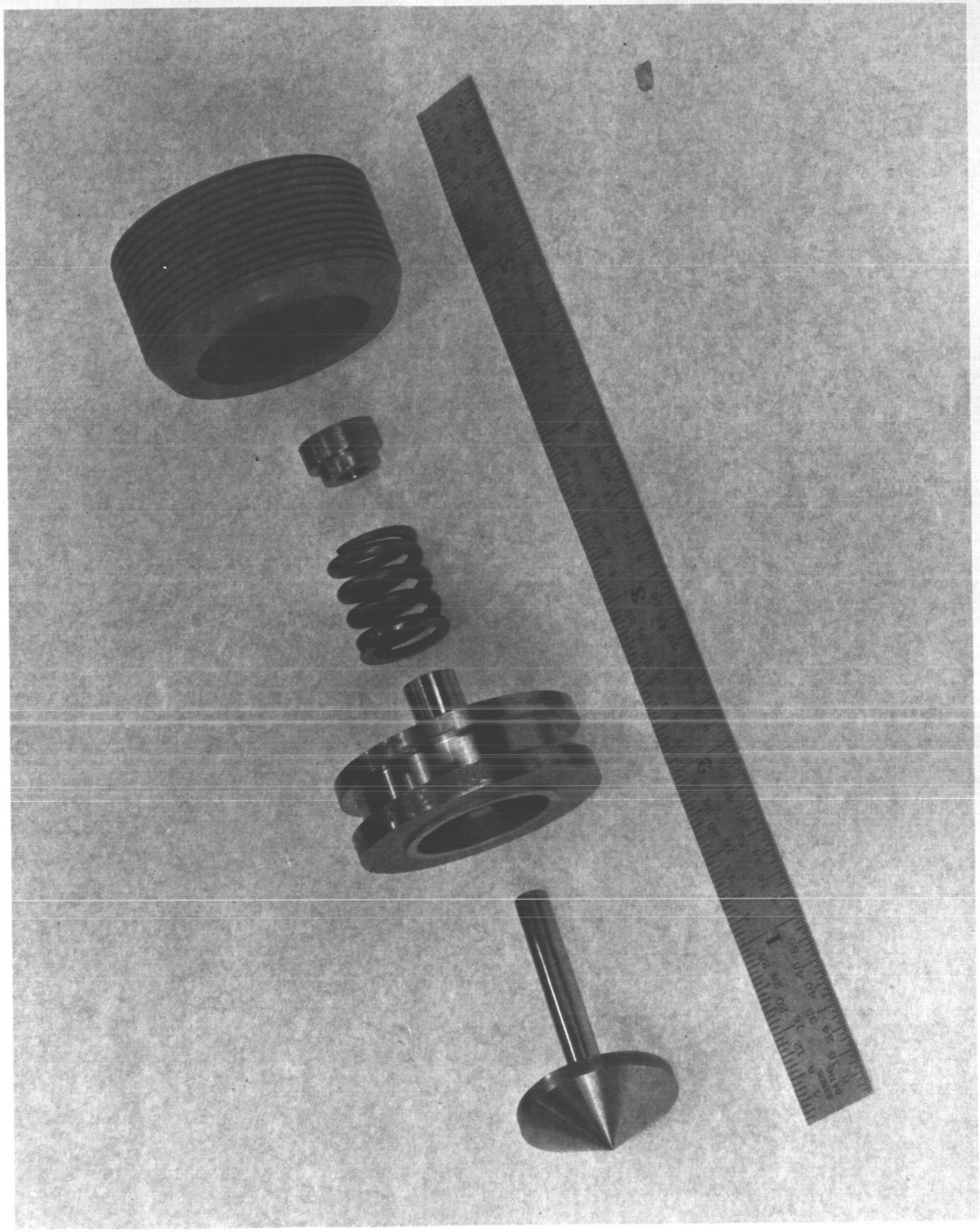
PART NO. Y05-055613 NEG. NO. _____ DESCRIPTION _____
NAME Piston Fuel Side Dashpot

Figure 3-8

3.3.3 Oil Side Check Valves

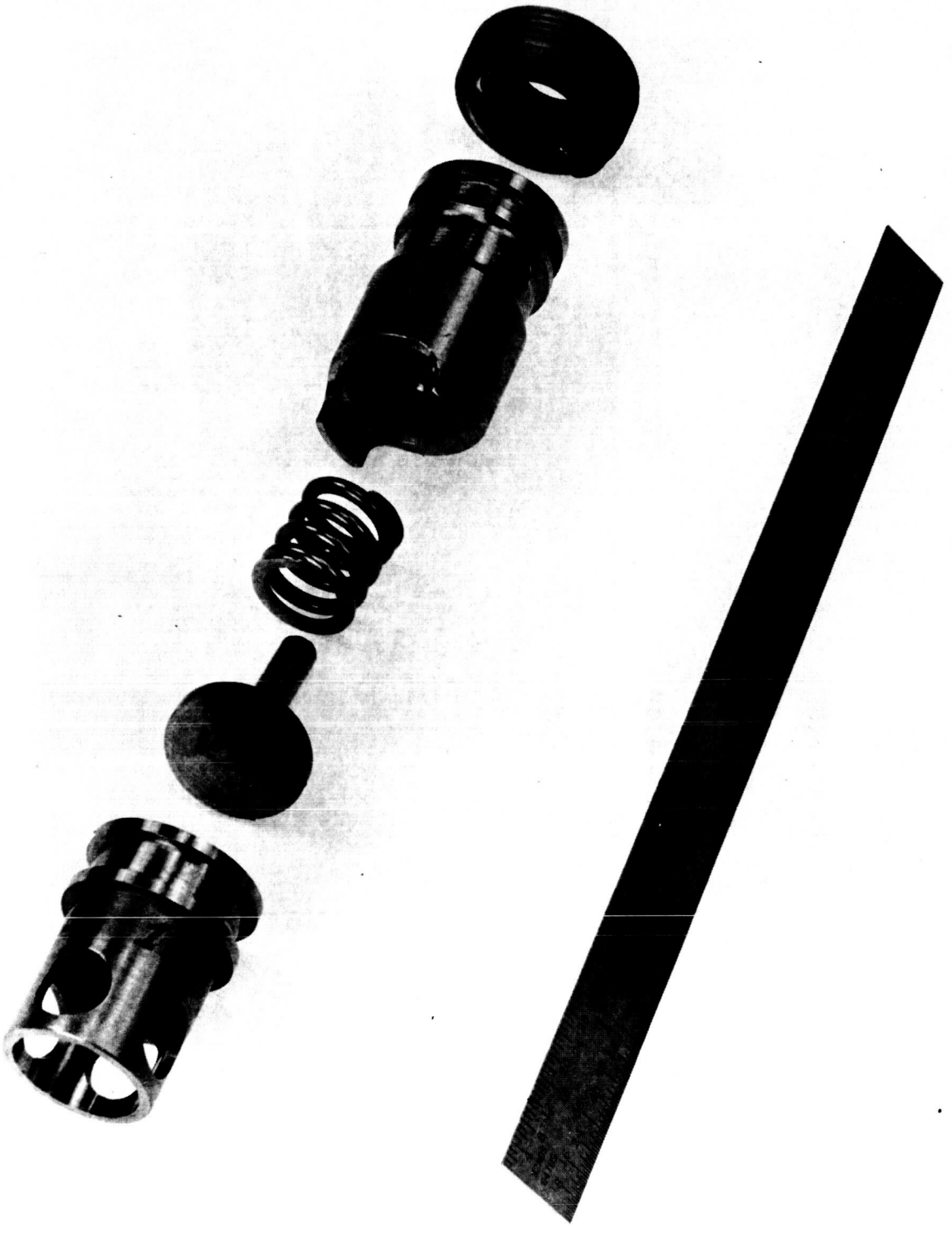
The inlet and discharge hydraulic oil check valves required relatively little development work as the final prototype designs proved very satisfactory in performance. The main feature of check valve operation is that their time of operation must be short, coinciding with the period that the pistons are at rest at the end of a stroke, so that the pistons will not waste power by discharging through the inlet port or pumping from the discharge port.

An analysis of check valve operation is presented in Section 5.3. Photographs of the check valves are shown in Figure 3-9 and Figure 3-10. Both check valves operated satisfactorily in the Hydraulic Transformer, but the discharge check valve body had to be made of steel rather than aluminum, as in the original prototype, and the valve seat was made longer in order to provide greater rigidity in the housing installation.



PART NO. Y05-055628 NEG. NO. _____ DESCRIPTION _____
NAME Hydraulic Oil Inlet Check Valve

Figure 3-9



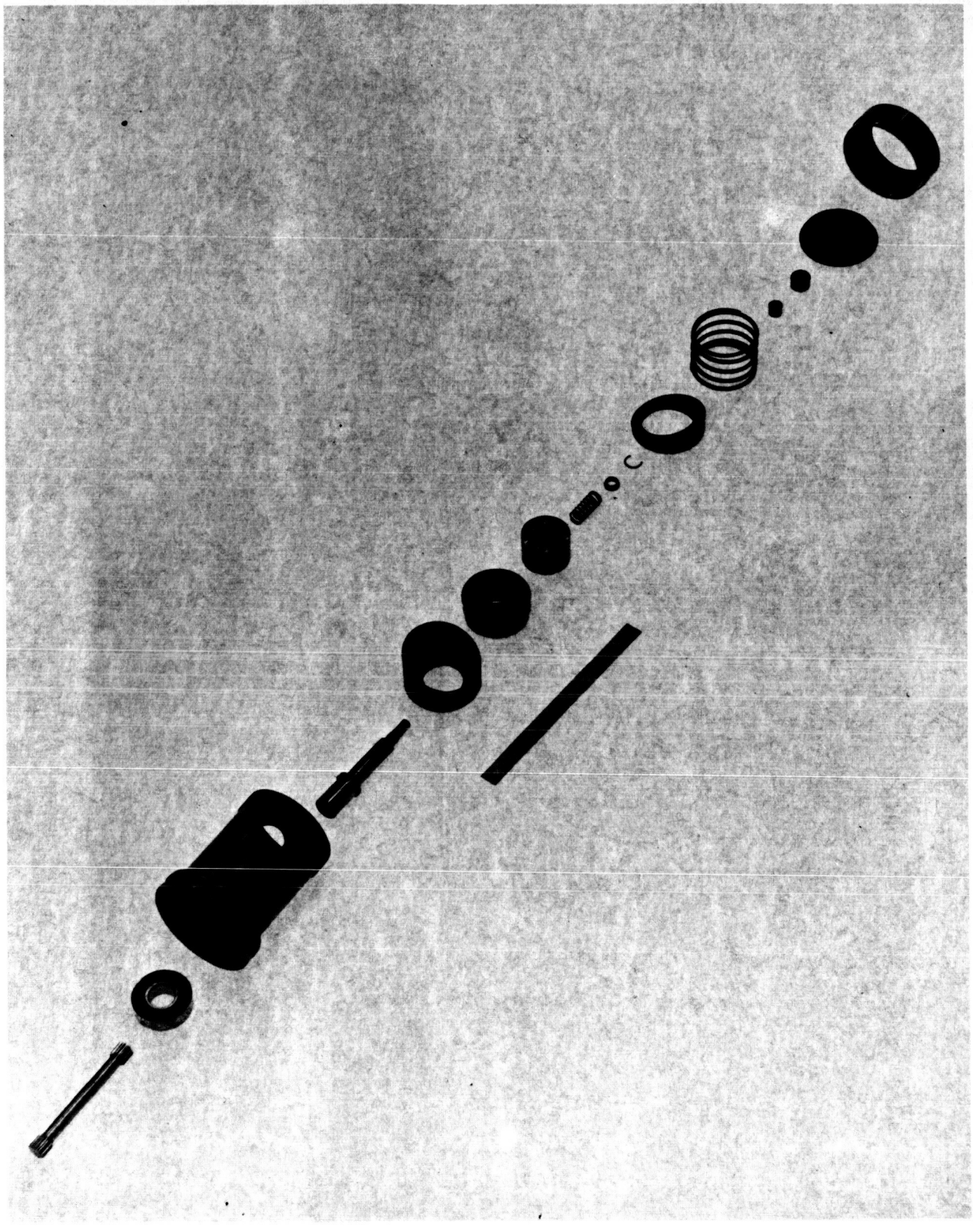
PART NO. Y05-055633 NEG. NO. _____ DESCRIPTION _____
NAME Hydraulic Oil Discharge Check Valve

Figure 3-10

3.3.4 Hydraulic Motor Drive

The hydraulic motor used for the rotary valve drive is a standard hydraulic motor design, which in this case is powered with the RP-1 rocket fuel. Since it wasn't known if the use of rocket fuel would adversely affect motor performance life, a 200-hour endurance test was run. The motor completed the test successfully without damage or any unusual signs of wear. A parts array of the motor is shown in Figure 3-11.

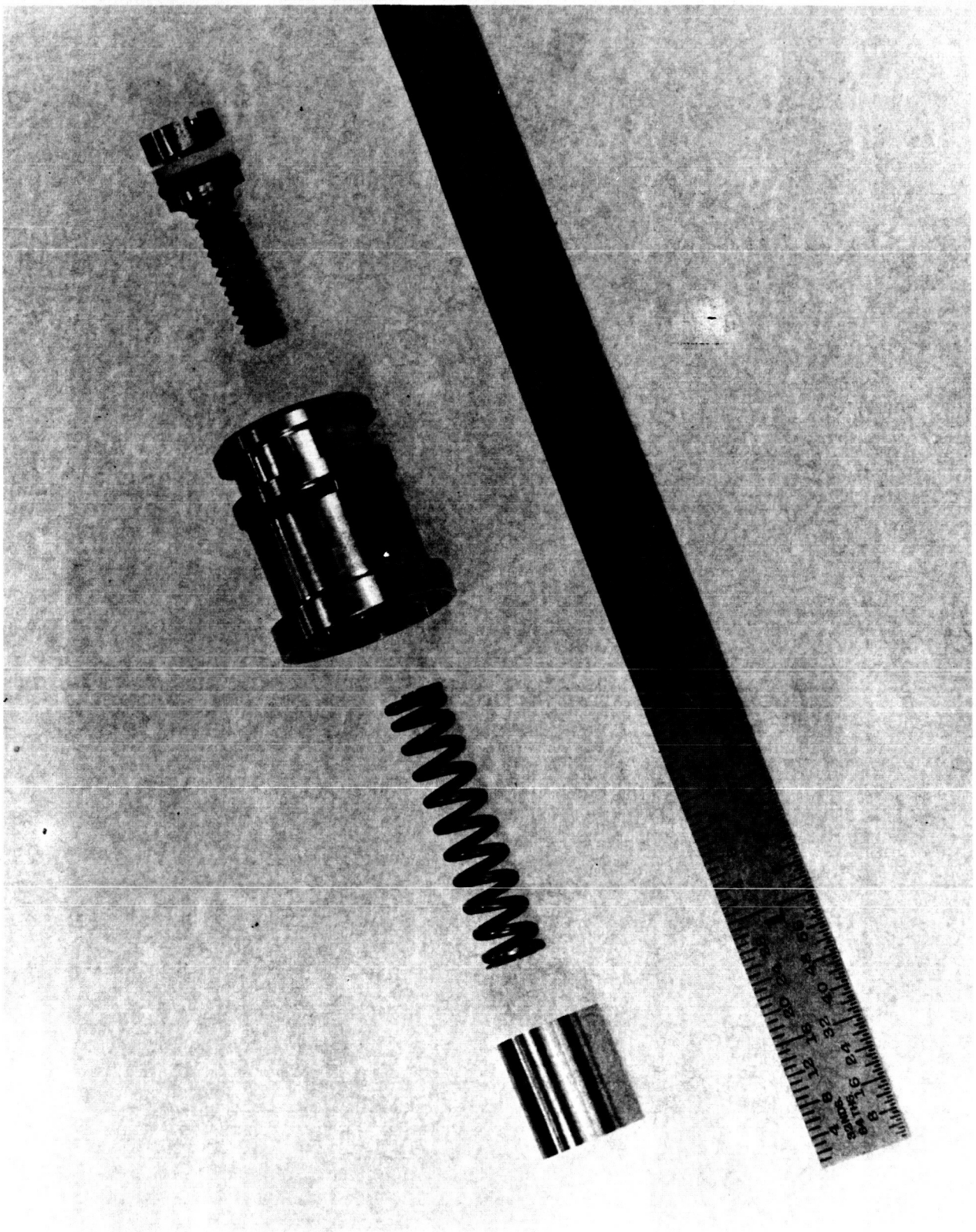
The motor speed control is shown in Figure 3-12. The speed control is mounted in the motor discharge passage within the hydraulic transformer housing, and regulates fluid flow to prevent motor speed from exceeding 1800 RPM.



PART NO. Y05-055650 NEG. NO. _____ DESCRIPTION _____

NAME Hydraulic Motor for Rotary Valve Drive

Figure 3-11



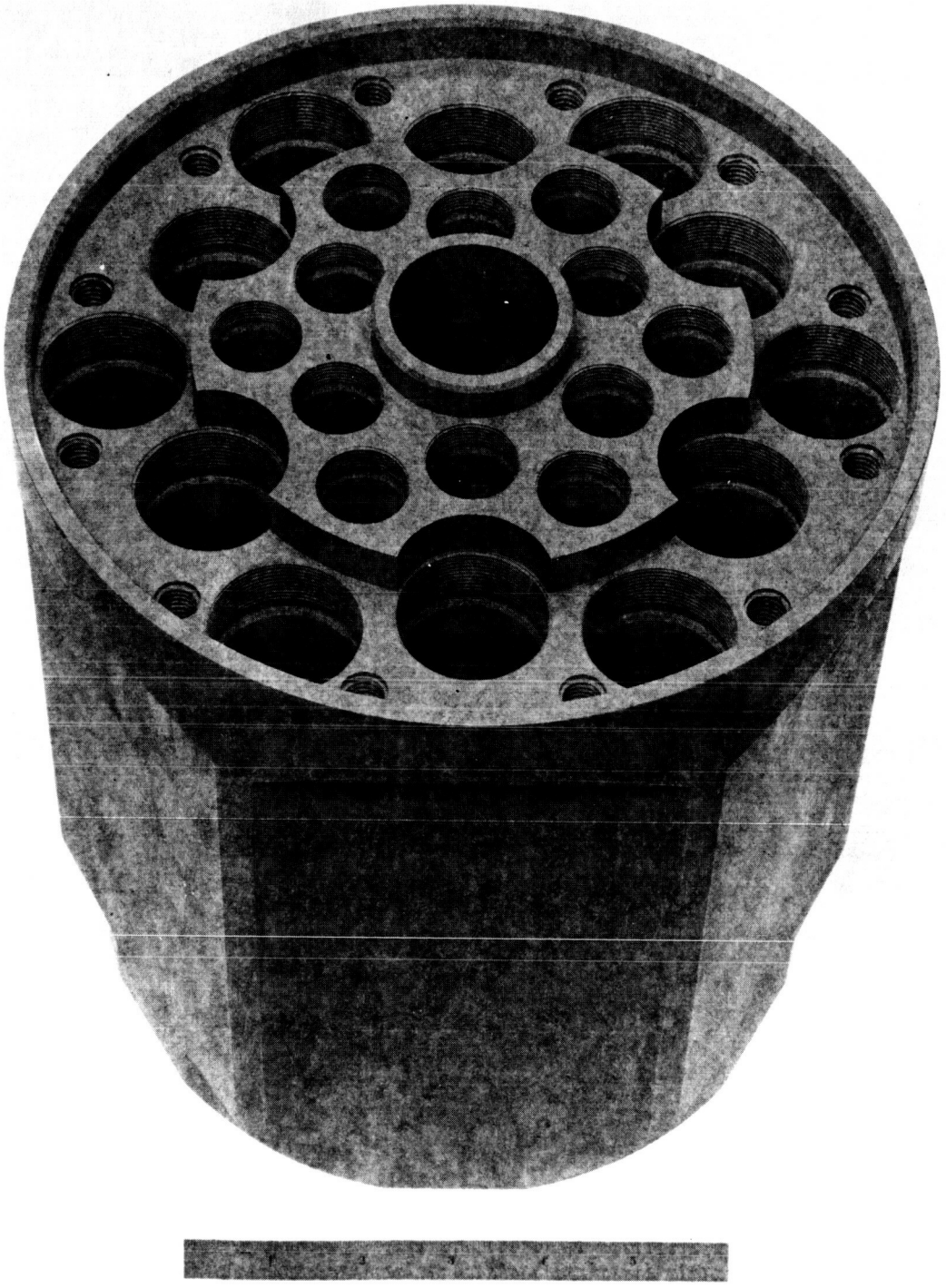
PART NO. Y05-055638 NEG. NO. _____ DESCRIPTION _____
NAME Hydraulic Motor Flow Control

Figure 3-12

3.3.5 Main Housing Assembly

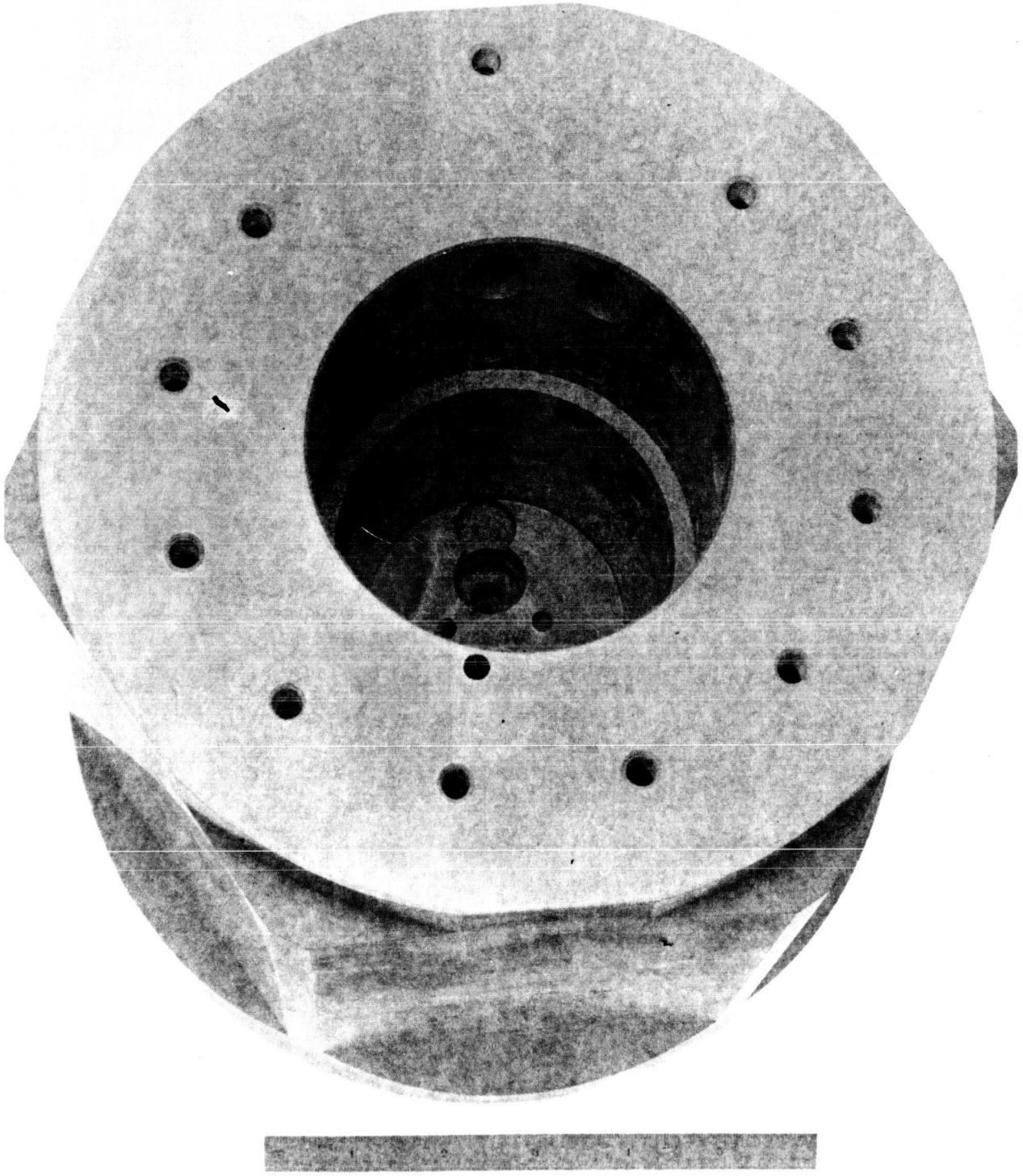
The main housing of the hydraulic transformer is a complicated part involving a great deal of machining, as may be seen in Figures 3-13 and 3-14. The ports on the oil side of the unit are in the oil side end cap shown in Figure 3-15, the discharge being in the center. The fuel side ports are shown in Figure 3-16, with the inlet at the top of the fuel cap.

The housing proved to be a weak point in the hydraulic transformer design. The housing material was aluminum to avoid excessive weight, and the size was kept as compact as possible partly to accommodate the available billet sizes. As a result, several passage walls were quite narrow, and this condition was made critical due to the fact that a number of undercuts in flow passages were machined with too-sharp corners. During testing, cracks originated at these sharp corners and propagated through the passage walls. Other difficulties arose due to the lack of stiffness of the housing and the fuel cap, causing binding of internal parts. These problems are discussed in Section 6.0 in greater detail.



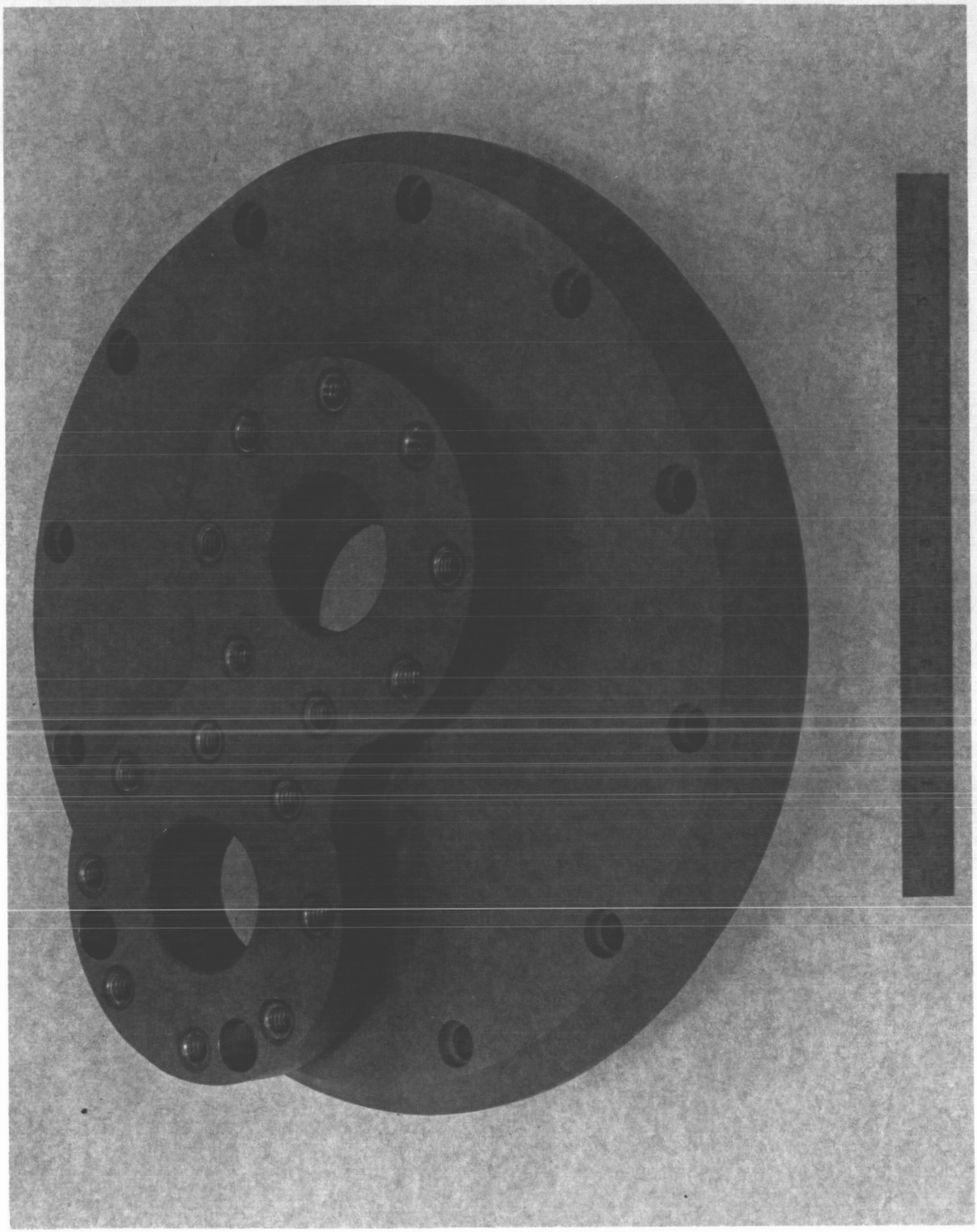
PART NO. Y05-055618 NEG. NO. _____ DESCRIPTION _____
NAME Oil Side of Hydraulic Transformer Housing

Figure 3-13



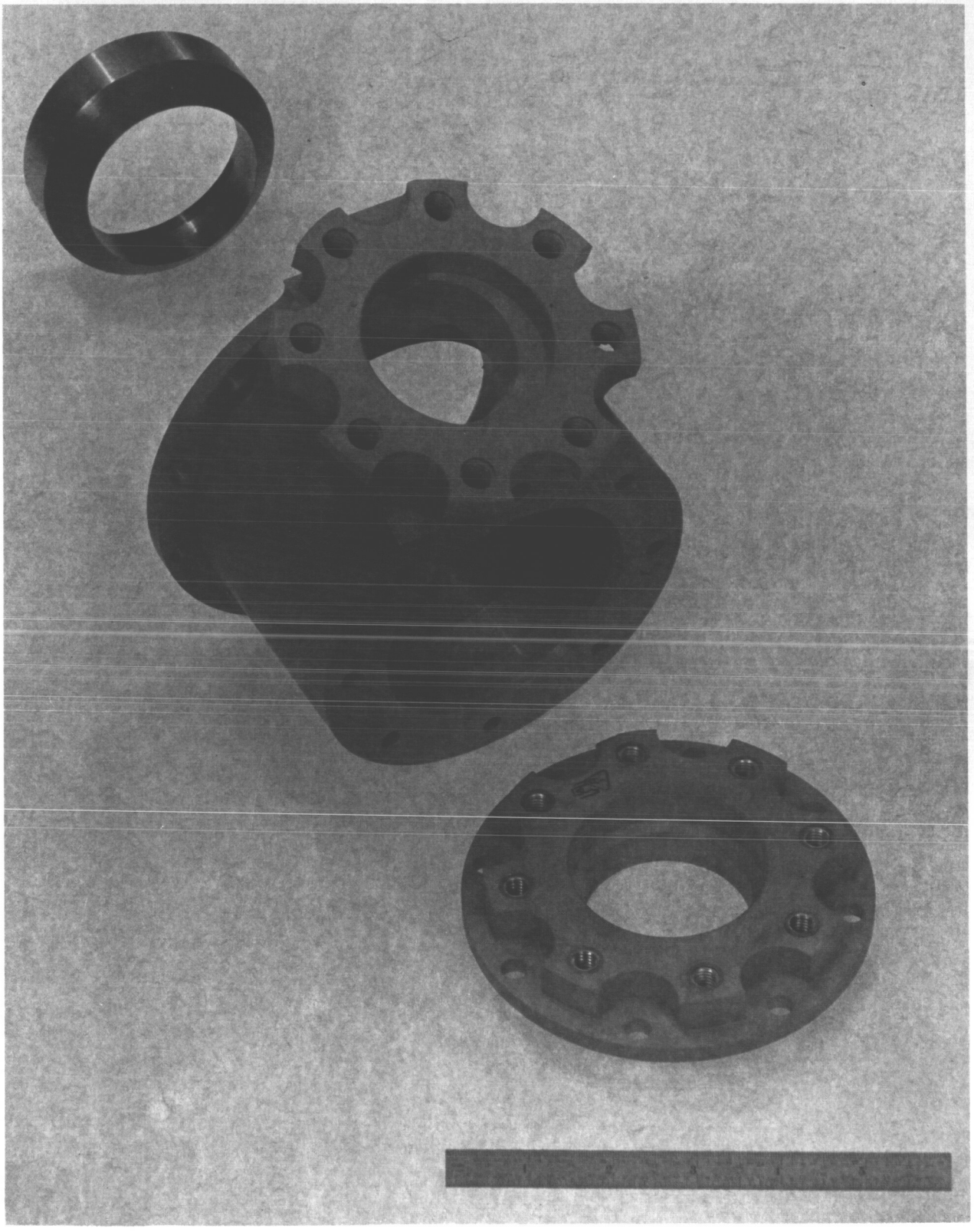
PART NO. Y05-055610 NEG. NO. _____ DESCRIPTION _____
NAME Fuel Side of Hydraulic Transformer Housing

Figure 3-14



PART NO. Y05-055615 NEG. NO. _____ DESCRIPTION _____
NAME Oil Side End Cap

Figure 3-15



PART NO. Y05-055617 NEG. NO. _____ DESCRIPTION _____
NAME Fuel Side Inlet & Outlet Ports

Figure 3-16

4.0 DEVELOPMENT HISTORY

The following account of the development history of the Hydraulic Transformer is derived and summarized from the monthly progress reports submitted during the course of the contract. NASA Contract No. NAS 8-5264 was instituted 28 February 1963, for the design, development and manufacture of two prototype Hydraulic Transformers to meet the requirements of NASA Specification M-P&VE-PLB (Revision A). Testing of a prototype unit was concluded in January 1965, and negotiations undertaken to conclude the contract. The development history of the unit is discussed in several phases as given below.

4.1 Analysis and Design

The initial period of the development program was devoted to analysis of the Hydraulic Transformer requirements and design of a prototype unit. In conjunction with design of the main unit was a parallel program to design fixtures to test individual components. Thus, if changes were necessary in individual component design, they could be accomplished early enough in the program to avoid extensive rework of the prototype unit hardware.

The main components which were tested individually were the hydraulic motor, the rotary valve, the pistons, and the check valves. A number of unforeseen problems did cause problems in component design, which occasioned considerable delay in the program. This delay eventually made it necessary to negotiate an overrun contract to extend the time of the program.

Initial efforts were also directed toward definition of the requirements of the test equipment needed for the final prototype unit. Due to the rather exceptional magnitude of the flows and pressures involved, there was considerable difficulty in finding suitable equipment. Test equipment specifications were written and procurement of equipment was started early in the program.

4.2 Component Design and Development

The design of individual components was developed by performance tests using individual test fixtures and a special test facility. This test facility was built early in the program, and was necessarily of only small capacity. It was not possible to simultaneously test with both high flow and high pressure. One pump was available for low flows at high pressure, and one pump supplied high flow at low pressure. Instrumentation was available to check all the significant operating parameters.

Development testing of the hydraulic motor was undertaken first, since it was planned to use an existing unit with slight modifications. Then, the rotary valve had to be developed to servo as the fuel supply regulation for the piston testing. The pistons and check valves had to be tested together, since their dynamic operation is interrelated.

4.2.1 Hydraulic Motor Tests

The hydraulic motor used to drive the rotary valve is a slightly modified version of the standard Hydro-Aire 68-193 motor. The selection of this motor was made on the basis of the estimated driving torque required for the rotary valve at the required rotational speed selected, 1800 RPM. The only concern was whether the motor would run satisfactorily with RP-1 rocket fuel for the required 200 hours. Due to the power lubricating properties of fuel as compared to oil, it was not known whether the wear of the parts would be excessive.

A 200-hour endurance test was run with the hydraulic motor using RP-1 fuel. The motor was recalibrated at the conclusion of the test, with no apparent loss in performance. After calibration, the motor was disassembled and inspected, and no significant signs of wear were observed. A photograph of the test motor is shown in Figure 4-1.

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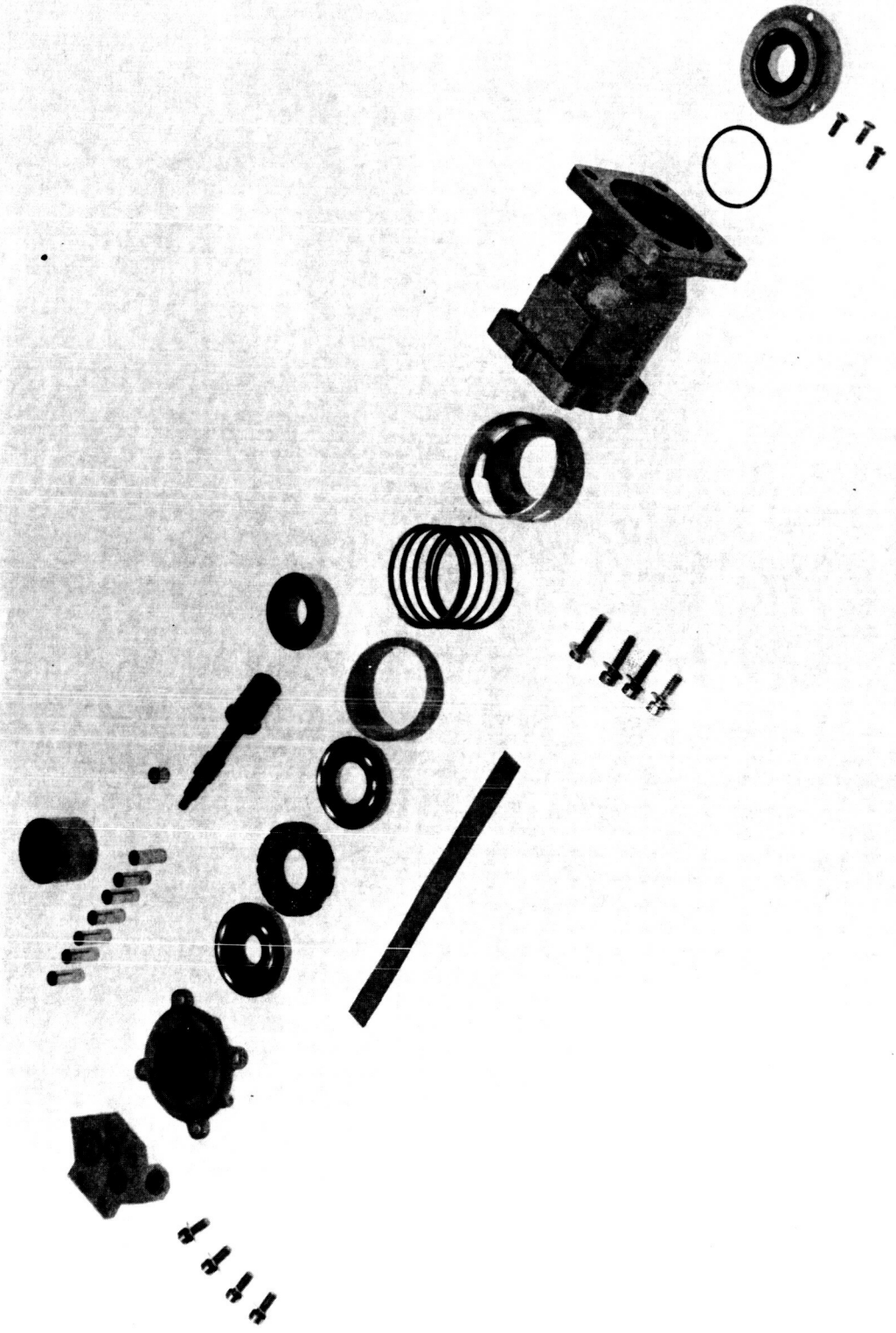


PART NO. 68-193 NEG. NO. _____ DESCRIPTION _____

NAME Motor for Fuel Endurance Test

Figure 4-1

4.2.2 Rotary Valve Development

A rotary valve test fixture was designed to secure rotary valve performance characteristics, including the following:

a) Torque, both starting and running
b) Rotational speed
c) Pressure drop
d) Leakage flow
e) Thrust

This fixture, shown in Figure 4-2, was made so that the rotary valve could be driven either by a dynamometer, or a hydraulic motor, as in the prototype Hydraulic Transformer. A picture of the test fixture using a dynamometer drive is shown in Figure 4-3.

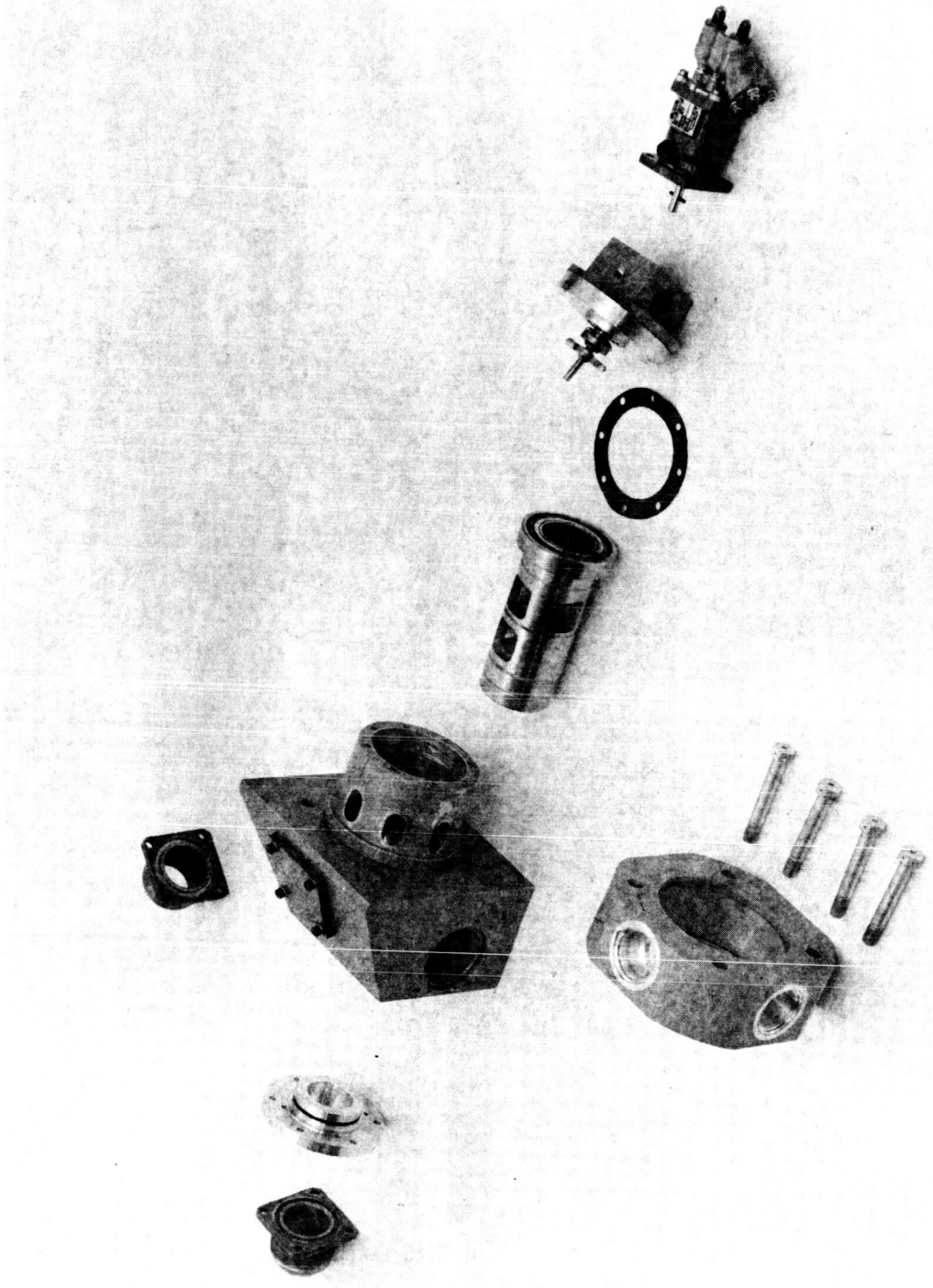
Initial testing indicated that rotary valve starting torque was excessively high at high fuel pressure, and consideration was given to using a high torque motor or alternatively a delay valve to delay the application of high fuel pressure. However, a more straightforward solution was found, by making changes to the rotary valve design:

(a) It was found that the main sleeve, for fuel distribution, was distorting and binding the valve. The sleeve was stiffened and press-fitted in the housing, thus preventing distortion.

(b) Initially the rotary valve had journal bearings, giving too much torque. A change was made to needle bearings, and finally to ball bearings.

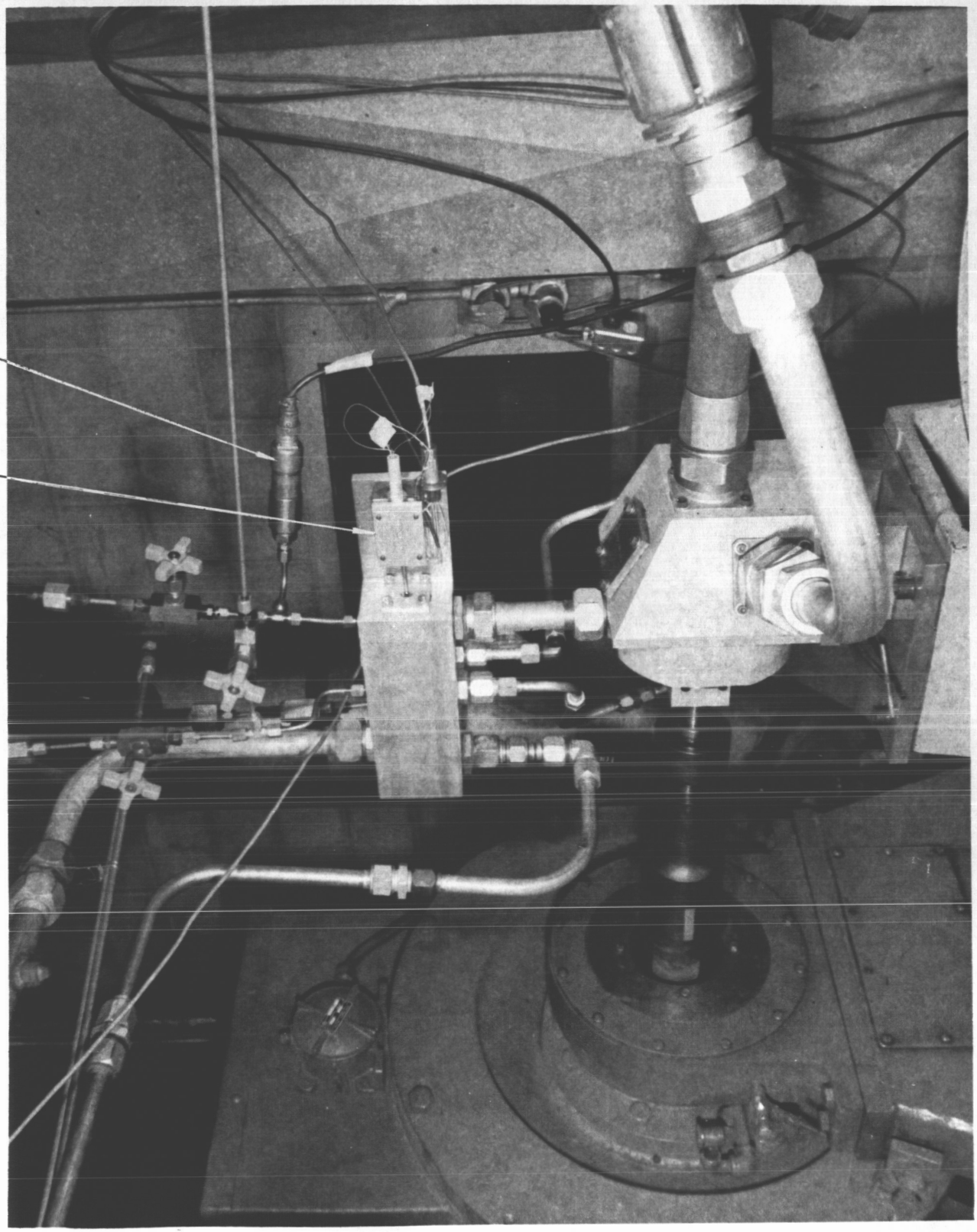
(c) There was an unbalanced pressure force on the rotary valve at starting. This condition was corrected by modification of the valve and sleeve ports.

After the modifications were made, the rotary valve tests indicated that the torque requirements were well within the drive capacity of the hydraulic motor selected for the prototype Hydraulic Transformer. Two of the rotary valve and sleeve configurations tested are shown in Figure 4-4, which may be compared with the final configuration previously shown in Figures 3-5 and 3-6.



PART NO. Y05-055-200 NEG.NO. _____ DESCRIPTION _____
NAME Rotary Valve Test Fixture Parts Array

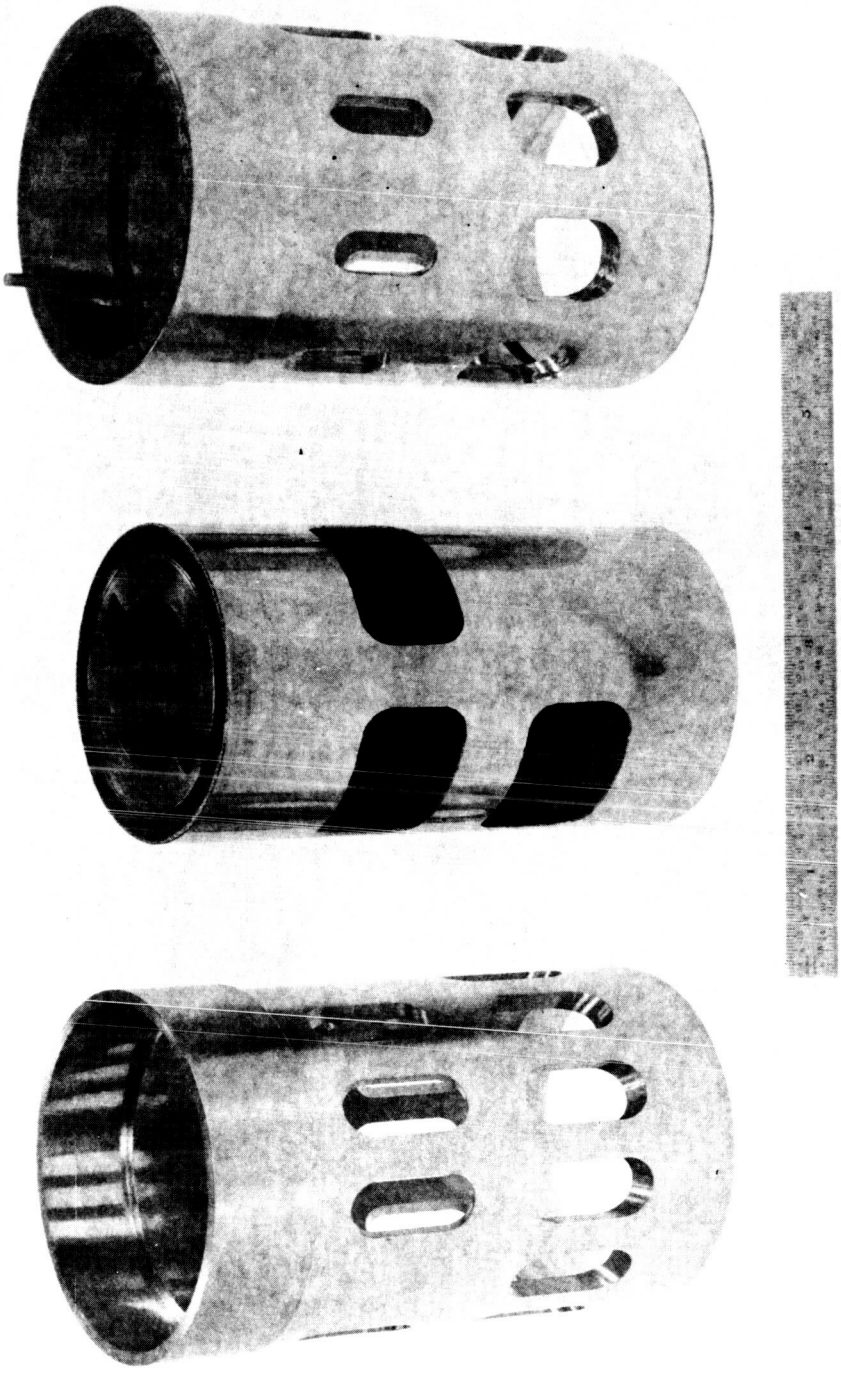
Figure 4-2



PART NO. Y05-055-400 NEG. NO. _____ DESCRIPTION _____

NAME One-Piston Test Fixture and Rotary Valve Fixture with Dynamometer Drive

Figure 4-3



105-055-203

105-055-204

105-055-203A

PART NO. 05-055-202 NEG. NO. _____ DESCRIPTION _____
NAME Rotary Valve Configurations

Figure 4-4

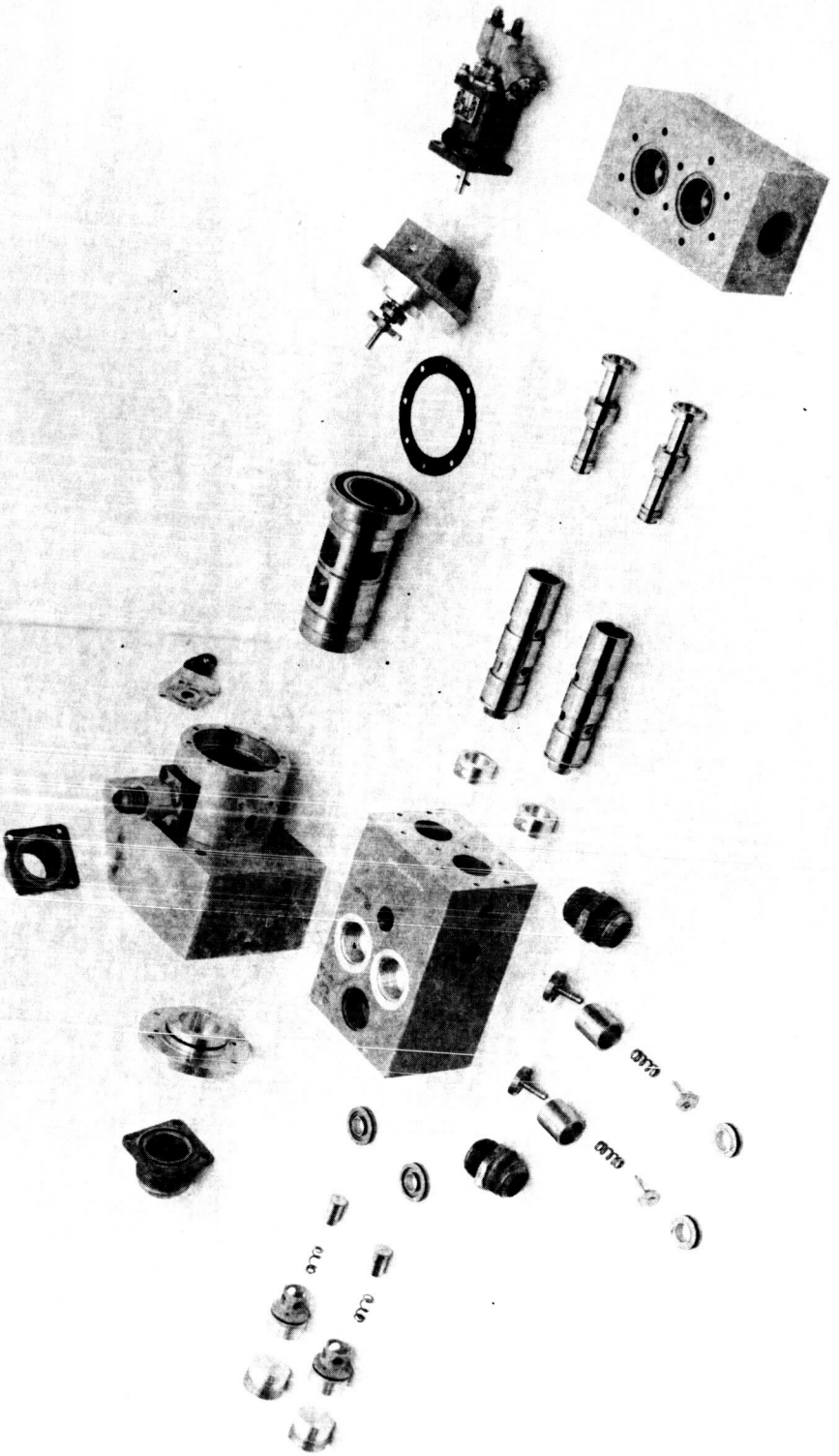
4.2.2 Piston and Check Valve Development

Two different test fixtures were made for testing the pistons and check valves. The check valves could be checked separately for cracking and reseating pressure, but dynamic operation was most convenient to test in the piston test fixture, since the fixture needed check valves anyway.

A one-piston test fixture was made with provision for recording piston displacement vs. time by using a linear potentiometer. The fuel supply to this fixture was regulated by using the rotary valve test fixture, as illustrated in Figure 4-3. The linear potentiometer is also shown and indicated by the symbol "X".

A two-piston test fixture was made, to be used with the rotary valve fixture, as may be seen on Figure 4-5. The two pistons receive fuel pressure from ports 90 degrees apart so that as one piston is in a power stroke, the other is in a suction stroke.

In the course of the development work on the piston assembly, several alternative designs were studied. The original design had springs to return the pistons after the power stroke. This spring design was dropped after careful consideration of some inherent design problems. The main objection to the spring



PART NO. Y05-055-300 NEG. NO. _____ DESCRIPTION _____
NAME Two-Piston Test Fixture
Parts Array

Figure 4-5

return design was the reduction in the unit reliability. The pistons would be traveling at a rate of 3600 cycles per minute and would exert a high stress load on the spring. Therefore, it was decided to use hydraulic instead of mechanical power to return the pistons.

Hydraulic return was accomplished by allowing high pressure fuel to enter a cavity in the middle of the piston assembly at the completion of the power stroke, which caused the piston to be returned. Calculations showed that this design is more efficient and reliable than the original spring return design. However, this design required that three sealing surfaces be used per piston assembly. The piston was designed in two pieces with a ball swivel joint connection to facilitate its fabrication. At least two of the sealing surfaces had to be lap fitted to the one-piece sleeve, while a third used a piston ring as a seal. The ball joint allowed some misalignment, and therefore the concentricity of the sleeve bores and the piston-plunger diameters was not a critical problem. However, during testing it became evident that this design was deficient in that the lips swedged over the ball became cracked under the oscillating load, as is shown in Figure 4-6. It was finally decided, that based upon experience with spool type valves and the availability of a new precision grinding machine, the piston-plunger could be manufactured

in one piece and still maintain the desired diametrical clearances. Therefore, the pistons in the final unit are of one-piece construction.

Considerable attention had to be devoted to piston material. The pistons must be light to assure fast stroking time, yet must be strong enough to resist the high pressure forces at each end. Several types of materials were tested, and the piston design was also modified to eliminate a stress concentration area created by a too-sharp corner radius which caused one piston to collapse under pressure. The final choice of material, based on the tests, was Graphmo Steel, heat treated to a Rockwell C 52 to 55 hardness.

The design of the piston assembly also raised some problems. The pistons travel at a very high velocity and must be decelerated and stopped in a very short distance at the end of the stroke. The solution was rather straightforward for the oil pumping stroke, using the closed end of the cylinder as a dashpot. However, at the fuel end for the return stroke, a dashpot could not be constructed at the end of the cylinder bore. Instead, it was necessary to make the dashpot within the end of the piston, matching a mating projection at the end of the cylinder.

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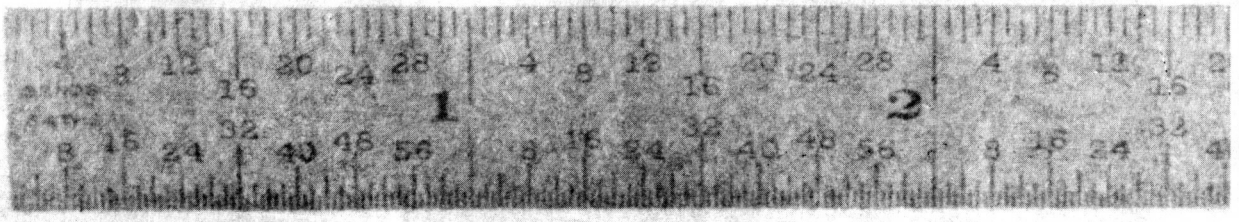
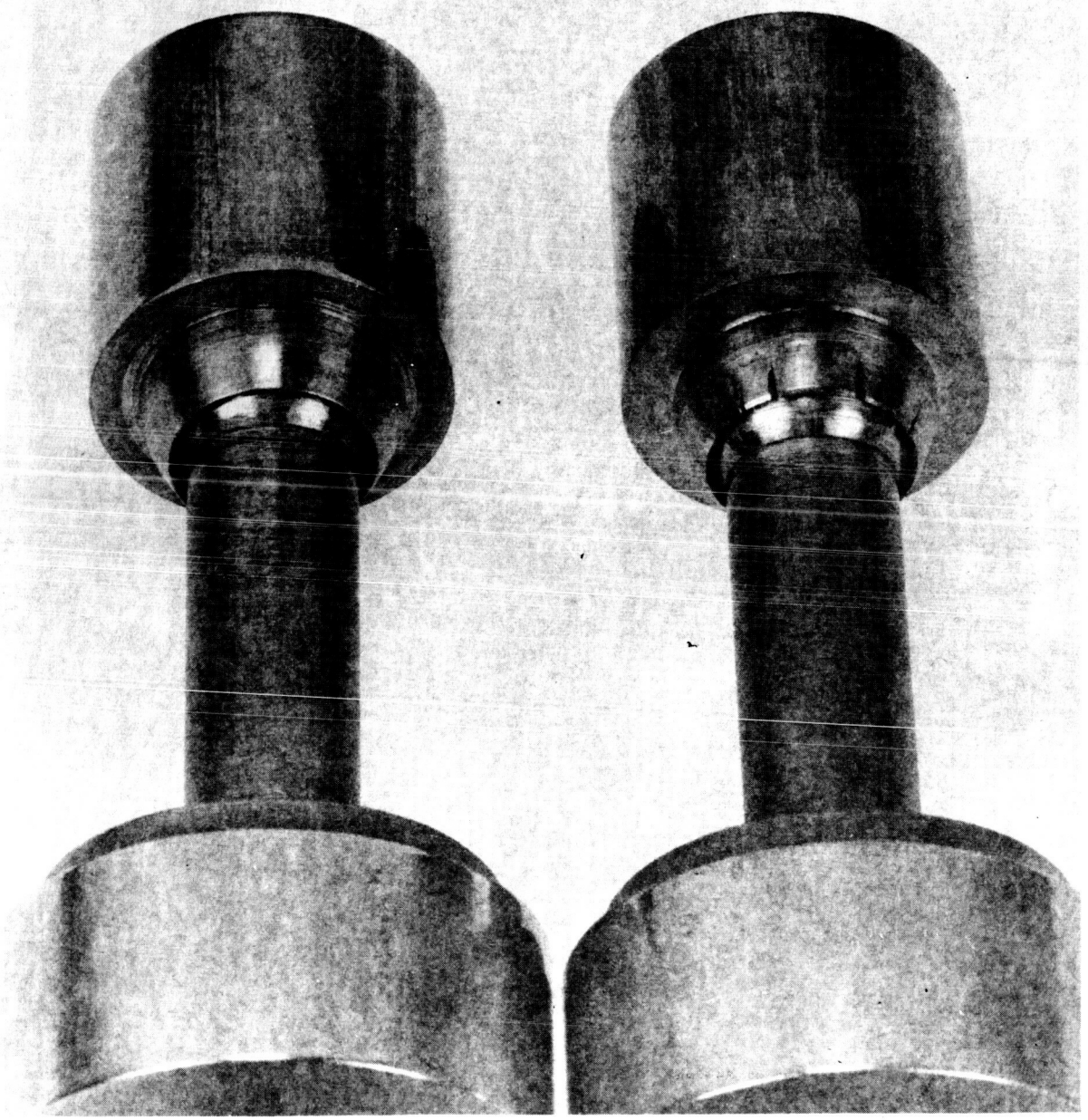


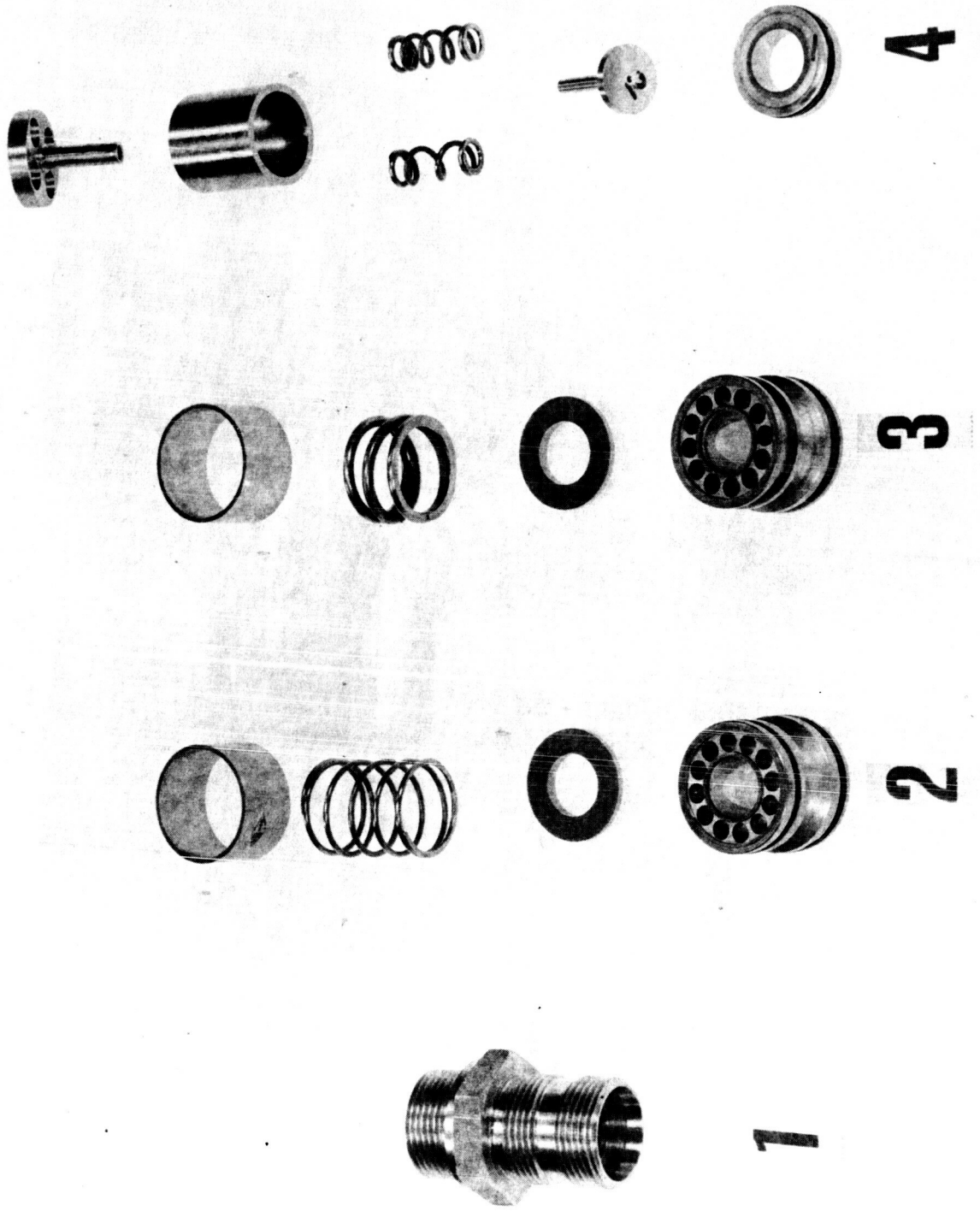


PART NO. Y05-05512A & 13A NEG. NO. _____ DESCRIPTION _____

NAME Piston-Plunger Assembly

Figure 4-6

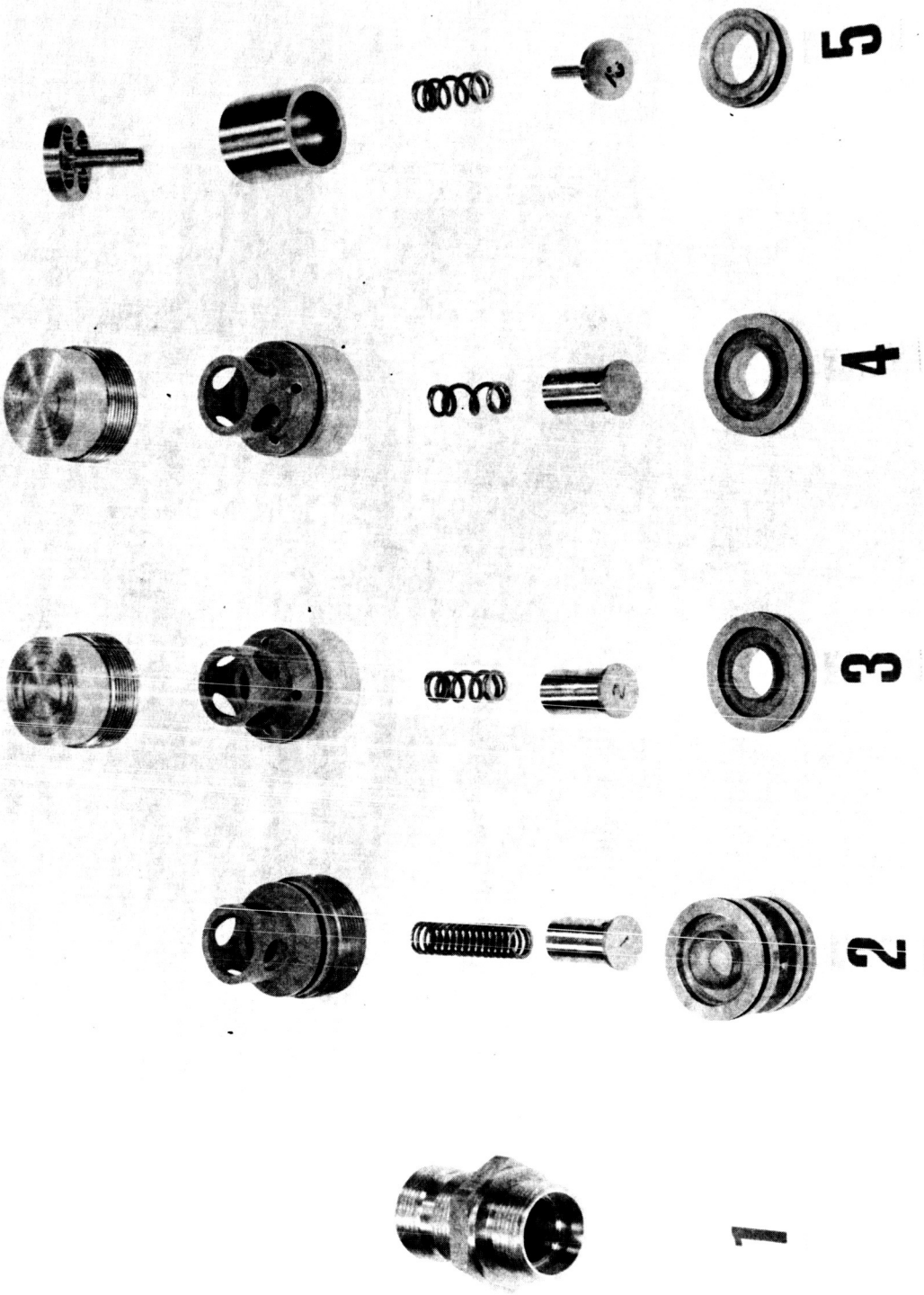
The initial designs of the hydraulic oil check valves were satisfactory as far as pressure drop flow characteristics were concerned, but it was found that dynamic response was not fast enough when used in the piston test fixture. The main feature of check valve operation is that their time of operation must be short, coinciding with the period that the pistons are at rest at the end of a stroke, so that the pistons will not waste power by discharging through the inlet port or pumping from the discharge port. Different valve springs were used, and performance was improved, but eventually a new design was adopted which was completely satisfactory. The valve modifications tested are shown in Figures 4-7 and 4-8.



PART NO. _____ NEG. NO. _____ DESCRIPTION _____

NAME Inlet Check Valve Configurations

Figure 4-7



PART NO. _____ NEG. NO. _____ DESCRIPTION _____

NAME Discharge Check Valve Configurations

Figure 4-8

4.3 Test Equipment

At the start of the program, efforts were directed toward securing a test facility for the performance testing of the prototype Hydraulic Transformer. The essential features of the test facility were that a variable capacity pump was needed to deliver a maximum 300 GPM flow of RP-1 fuel at a pressure of 2000 psig, having a prime mover drive of over 450 horsepower. The fuel circuit needed filters, heat exchangers to control fuel temperature, and control valves with all the necessary plumbing. In addition, an oil circuit with control valves was needed to regulate the hydraulic oil flow pumped by the Hydraulic Transformer. And finally, the supplementary equipment required included fuel and oil tanks, electrical supply, water supply, cooling towers, circulating pumps, and safety equipment.

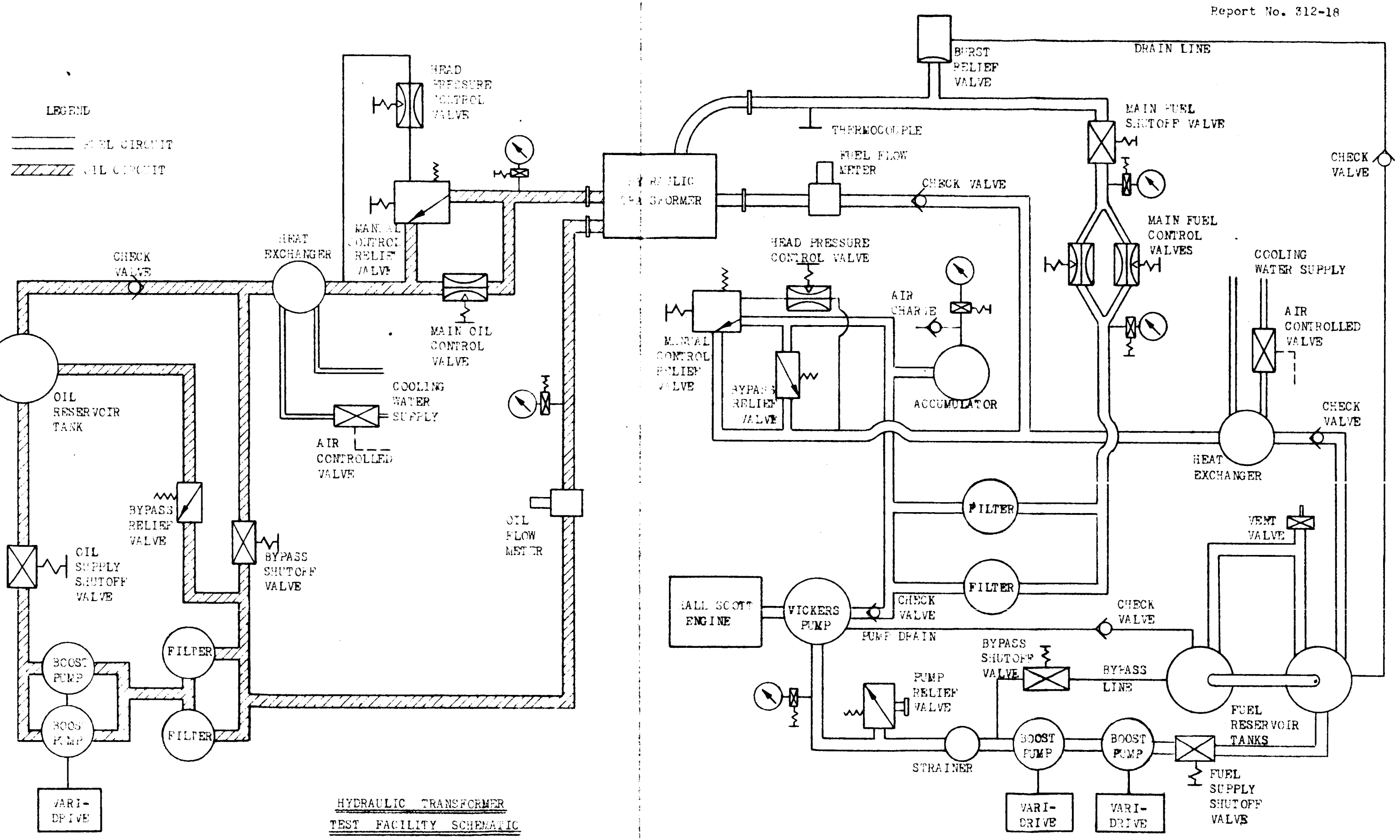
It was found that the equipment cost estimates used in preparing the initial proposal were not reliable, and also that pump vendors, particularly, were unwilling to verify their equipment for the use intended. It was necessary to make a new start. Indications were that test facility cost would considerably exceed the amount originally forecast.

In order to keep costs as low as possible, it was decided to invite bids from outside vendors to build the test facility using available Hydro-Aire equipment wherever possible. It was found that Hydro-Aire could supply much of the supplementary equipment, including tanks and booster pumps, thus limiting procurement to the main fuel pump, engine drive, and the controls for the fuel and oil circuits.

Bids were solicited for a test stand containing the essentials of the test facility. Auto Control Laboratories, Inc., of Los Angeles, submitted the lowest bid, and with NASA approval, received a contract for construction of the test stand. The contract was signed on 21 June 1963, with delivery promised for September 1963, but numerous delays postponed delivery until March 1964. Various difficulties were encountered with the equipment, including the failure of the main pump-drive engine. The engine was removed, and a replacement was finally installed in August 1964. The facility was tested and found to deliver the required fuel flow and pressure, although it was found to be difficult to control flow and pressure to any close degree.

With the start of full scale testing of the prototype Hydraulic Transformer, the difficulties in controlling the test facility

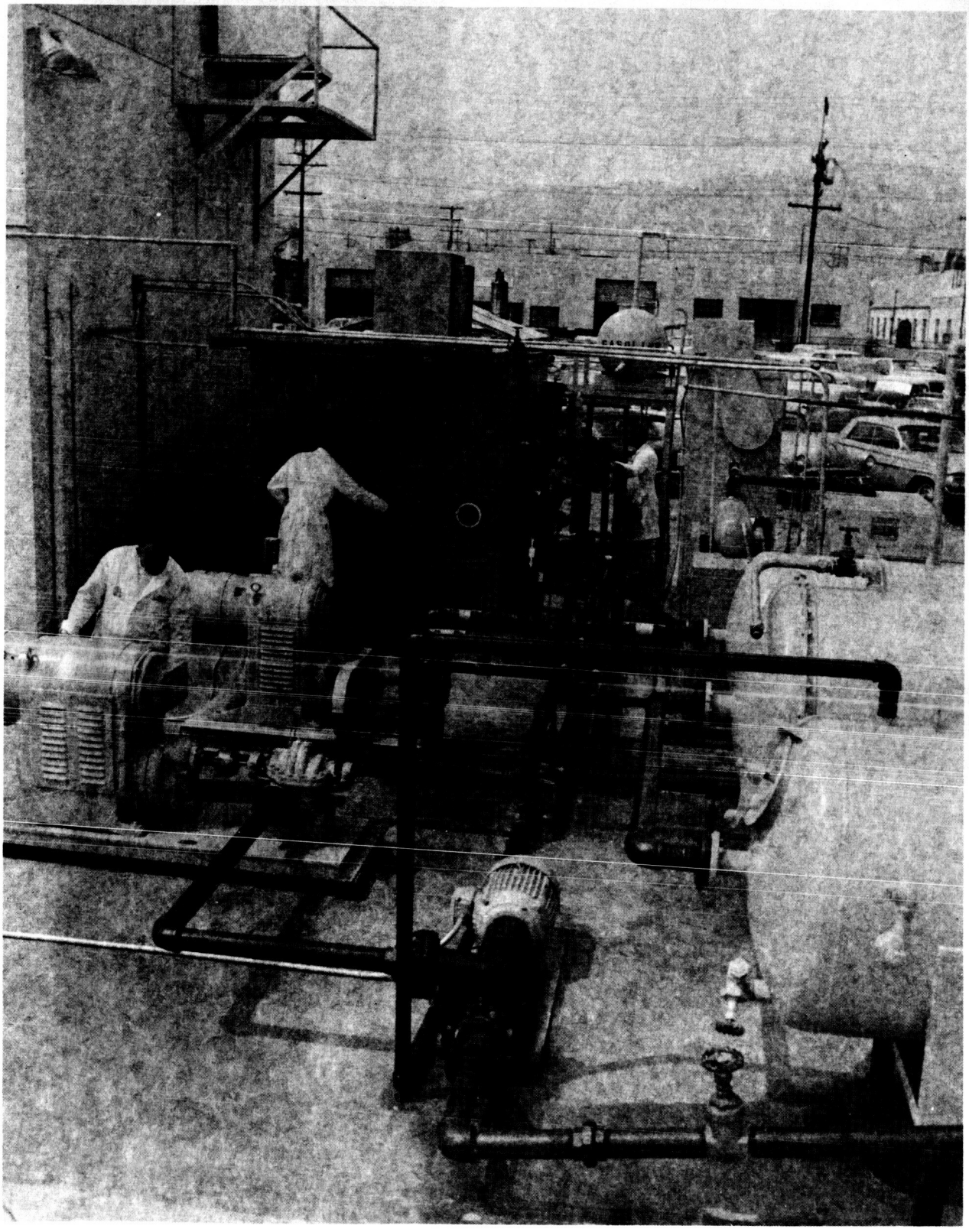
became quite critical. It became necessary to make a number of revisions in the flow control circuits to achieve stable operation. In spite of improvements in the test facility, it is believed that pressure surges during operation were a contributing factor in the stress failure of the housings of both of the prototype Hydraulic Transformers. The latest revisions to the test facility were made in adding a burst tube for protection against surges, since efforts were not completely successful to achieve surge-free control of the system itself. The final schematic of the test facility is shown in Figure 4-9, and a partial photograph is shown in Figure 4-10.



HYDRAULIC TRANSFORMER
TEST FACILITY SCHEMATIC

HYDRO-AIRE DIVISION, CRANE CO. 3000 WINONA AVE. BURBANK, CALIF.

Figure 4-9



PART NO. _____ NEG. NO. _____ DESCRIPTION _____

NAME Partial View of Hydraulic Transformer Test Facility

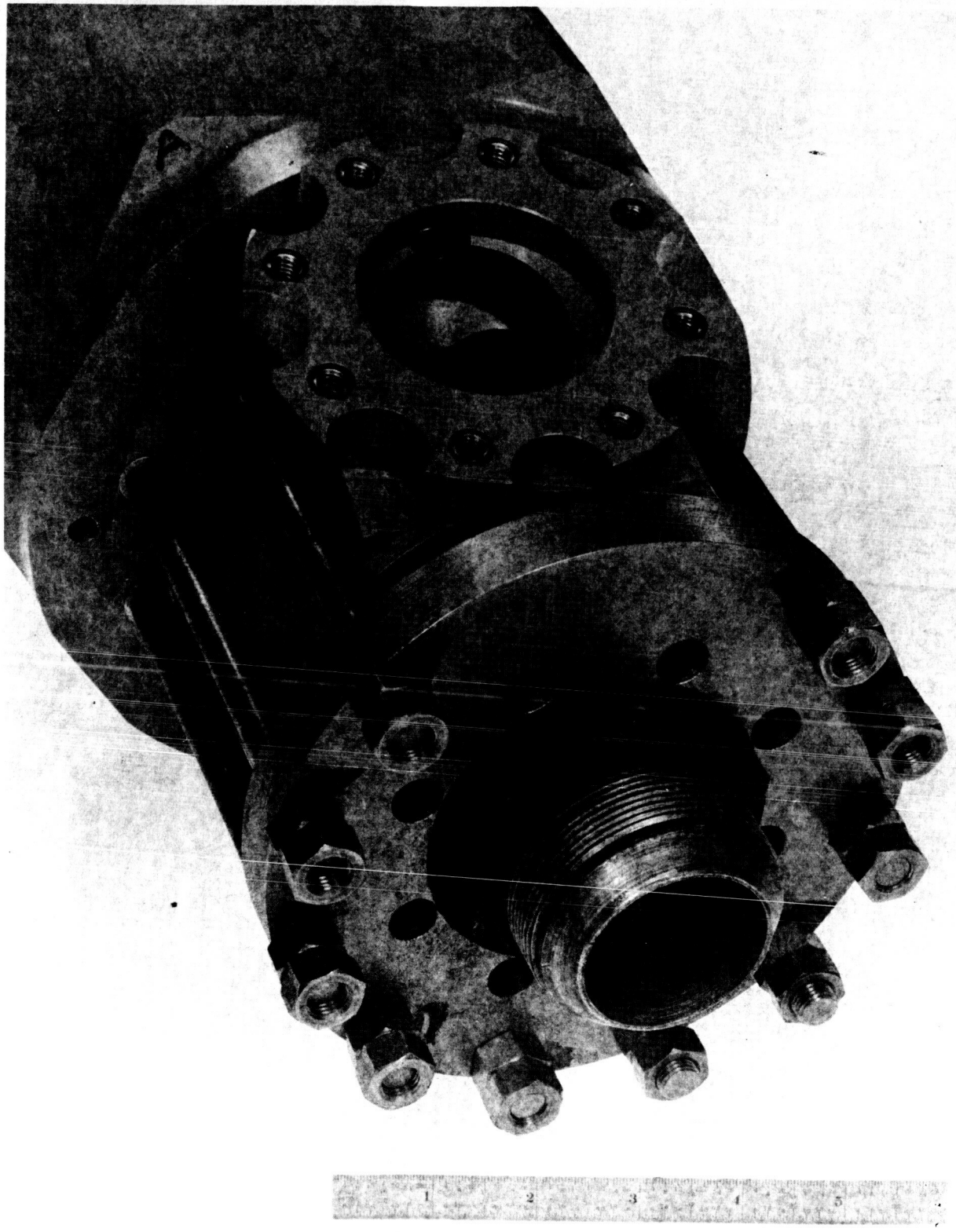
Figure 4-10

4.4 Prototype Fabrication and Testing

With the completion of tests on the individual Hydraulic Transformer components, the final design of the prototype unit was made. All the changes in individual components were incorporated which resulted from the previous testing.

Fabrication of the various components took a number of weeks, since some of the subassemblies also required shrink fits and finish-machining. Two prototype Hydraulic Transformer units were assembled, and preliminary tests made to be sure that the hydraulic motor would drive the rotary valve at the required speed. Further testing had to await completion of the main test facility.

When the test facility was operable, tests of the two prototype units were started. A considerable amount of difficulty was encountered in both units, with binding of the rotary valve stopping the unit from operating. The trouble was found to be pressure distortion of the housing and fuel cap, which was finally alleviated by using a clamping fixture to limit distortion. The installation of the clamping assembly is shown in Figure 4-11.



PART NO. SK-11411 NEG. NO. _____ DESCRIPTION _____
NAME Clamping Assembly Installed on Hydraulic Transformer Fuel Cap

Figure 4-11

In the initial tests, difficulties were already being encountered in control of the test facility. Finally, in one test a sudden 5000 psig surge in fuel pressure cracked the housing of one prototype unit. Improvements were subsequently made in the test system's controls before continuing performance tests with the remaining unit.

The second unit was tested at several fuel pressure levels, starting at only 500 psig, to avoid the difficulties which the test equipment previously encountered at high pressure. The Hydraulic Transformer was found to operate as expected, though at low fuel pressure, the flow and efficiency were lower than they would be as the pressure level increased. A number of performance tests were run at several fuel pressure levels, until the second Hydraulic Transformer also suffered a cracked housing. Although the general feasibility of the unit had been demonstrated, the main cause of failure is attributed to a housing which is not sufficiently strong for the maximum pressures encountered. The performance curves of the Hydraulic Transformer are presented in detail in Section 6.0.

5.0 THEORETICAL ANALYSIS

A theoretical analysis of the hydraulic transformer is presented to show the reasoning behind the design and aid in interpreting the performance test results. The analysis covers the three main component items which influence pressure and flow capacity: the pistons, the rotary valve, and the hydraulic oil check valves.

5.1 Piston Analysis

The pistons are the critical components of the hydraulic transformer which perform the pumping and pressure intensification function. The pistons are "free acting", without mechanical drive, acted on by pressure forces of the oil and fuel in addition to friction and inertia forces.

The differential equation that describes the piston dynamics is developed from the force balance on the free piston during the power stroke:

$$M \frac{d^2x}{dt^2} + B \frac{dx}{dt} + P_o A_o = P_f A_f \qquad (1)$$

Where:

$M = M_p + M_f$ = Accelerated Mass

M_p = Piston Mass

M_f = Fluid Mass

B = Damping coefficient

P_o = Oil pressure

A_o = Piston area, oil side

P_f = Fuel pressure

A_f = Piston area, fuel side

$\frac{dx}{dt}$ = Piston velocity

$\frac{d^2x}{dt^2}$ = Piston acceleration

The total output flow rate is given by:

$$q = ZA_o \frac{dx}{dt} \qquad (2)$$

Where:

q = Total output flow rate

Z = Number of pistons pumping simultaneously

From (2) we obtain:

$$\frac{dx}{dt} = \frac{q}{ZA_o} \qquad (3)$$

$$\frac{d^2x}{dt^2} = \frac{1}{ZA_o} \frac{dq}{dt} \qquad (4)$$

Introducing (3) and (4), equation (1) becomes:

$$\frac{M}{ZA_o} \frac{dq}{dt} + \frac{B}{ZA_o} q + P_o A_o = P_f A_f \qquad (5)$$

Substituting:

$$P_f A_f - P_o A_o = F \tag{6}$$

Using Laplace transform notation and re-arranging:

$$\left[\frac{M}{ZA_o} S + \frac{B}{ZA_o}\right] q_{(s)} = \frac{F}{S} \tag{7}$$

Solving for $q_{(s)}$:

$$q_{(s)} = \frac{F}{S\left[\frac{M}{ZA_o} S + \frac{B}{ZA_o}\right]} \tag{8}$$

$$q(s) = \frac{ZA_o F/M}{S(S + \frac{B}{M})} = \frac{K_1}{S} + \frac{K_2}{(S + \frac{B}{M})} \tag{9}$$

$$K_1 = -K_2 = \frac{ZA_o F}{B} \tag{10}$$

Taking the inverse Laplace transform of (10), we obtain the expression of the flow in the time domain:

$$q(t) = \frac{ZA_0F}{B}\left(1 - e^{-\frac{B}{M}t}\right) \tag{11}$$

The average flow with time is given by:

$$q_a = \frac{\int_0^{t_s} q(t)\,dt}{t_s} \tag{12}$$

where t_s is the stroke time required to obtain the specified flow.

$$t_s = \frac{ZA_0F}{Bq_a}\int_0^{t_s}\left(1 - e^{\frac{B}{M}t}\right)dt \tag{13}$$

With the factors Z, A_0, F and q_a remaining constant, equation (13) shows that in order to reduce the time required to complete a pumping stroke, a low damping coefficient and minimum piston mass are desirable.

From the results of the preceding analysis, a design philosophy is established to achieve maximum flow:

a) The damping factor should be reduced to a minimum value, avoiding any detrimental effect in the discharge pressure characteristic of the unit.

b) The piston weight should be reduced to a minimum commensurate with the required strength of the part.

c) The selection of the total number of pistons pumping simultaneously must be consistent with a rotary valve design which is pressure balanced.

For the present design, the rotary valve is constructed so that 4 pistons are always pumping simultaneously.

Other representative values of the unit parameters are selected as follows:

W_p = Piston weight = 1.302 pounds

W_o = Oil weight = 0.135 pounds

M_T = Total mass accelerated = $M_o + M_p$

M_o = Oil mass

M_p = Piston mass

A_R = Total friction area = $A_{RF} + A_{RO}$

A_{RF} = Fuel friction area = 3.20 $in.^2$

A_{RO} = Oil friction area = 1.15 $in.^2$

μ_F = Fuel viscosity = 1.5×10^{-6} lb-sec/in^2

μ_o = Oil viscosity = 3.12×10^{-6} lb-sec/in^2

A_o = Oil side piston area = 1.480 in^2

A_F = Fuel side piston area = 1.090 in^2

h = Piston radial clearance = .0004 in

P_F = Fuel pressure = 1850 psi

P_o = Oil pressure = 4000 psi

Thus:

$$M_T = M_p + M_o = 1.06 \times 10^{-3} \ \frac{LBS.\ SEC.^2}{IN}$$

$$F = A_F P_F - A_o P_o = 180 \ LBS$$

$$B = \frac{A_R \mu}{h} = \frac{A_{RF} \mu_F + A_{RO} \mu_o}{h} = \frac{(3.20 \times 1.5 + 1.15 \times 3.12) 10^{-6}}{4 \times 10^{-4}} =$$

$$B = 2.12 \times 10^{-2} \ \frac{LBS.\ SEC.}{IN.}$$

Replacing values in equation (11):

$$q(t) = \frac{4 \times 180 \times .460}{2.12 \times 10^{-2}} - \frac{4 \times 180 \times .460}{2.12 \times 10^{-2}} e^{-20t}$$

$$q(t) = 10,400 \left(1 - e^{-20t}\right) \Big/ \qquad (19)$$

$$0 < t < .0072 \text{ SEC.}$$

Which is the instantaneous flow from the 4 pistons which are pumping at any time, for a stroke duration of .0072 sec. The average oil flow for this period is then:

$$q_a = \frac{\int_{t=0}^{t=.0072} 10,400\left(1 - e^{-20t}\right) dt}{.0072} = 680 \frac{IN^3}{SEC}$$

$$q_a = 177 \text{ GPM}$$

This flow is the average flow realized during the period of time when the 4 pistons, which operate simultaneously, are pumping. Since the rotary valve is constructed so that 4 pistons are

always pumping simultaneously, this oil flow of 177 GPM represents the theoretical average output of the intensifier as a whole. The analysis indicates that the present design configuration is capable of much greater capacity than the specification requires. In operation, this excess capacity is limited by the action of the inlet and outlet check valves which need adequate time for operation. The rotary valve is thus constructed to allow a time gap between piston strokes, for check valve operation, and total flow is consequently reduced.

It will be realized that many of the values chosen for the parameters are semi-independent variables which are empirically established as a result of the tests on the individual components. The following analyses of the rotary valve and the check valves will also illustrate some of the rationale behind the choice of individual parameters.

5.7 Rotary Valve Operation

The function of the rotary valve is to direct the flow of the high pressure fuel to the pistons in the proper sequence. The valve is rotated by a hydraulic motor driving the valve through

a splined shaft. The rotary valve operating speed may be represented in the following manner:

$$N = \frac{q_o \ A_F/A_o}{N_p \ T \ A_F \ X \ \eta_v}$$

where:

N = Rotational speed = RPM

q_o = Oil flow = 100 GPM

q_F = Fuel flow

N_p = Number of pistons = 12

T = Strokes per revolution = 2

A_o = Oil side piston area = .460 in.^{2}

A_F = Fuel side piston area = 1.090 in.^{2}

X = Piston stroke = 1.25 in.

η_v = Volumetric efficiency = 95% assumed

The relationship between oil flow and fuel flow is:

$$q_F = q_o \ \frac{A_F/A_o}{\eta_v}$$

The flow losses represented in the volumetric efficiency are mostly due to fuel leakage in the rotary valve and the fuel used to drive the pistons on the return stroke. Under the above conditions, the rotational speed is:

N = 1760 RPM

Based on the values above, the rotary valve design must fulfill the following requirements:

a) Provide equal flow area for both the fuel pressure and return passages in order to avoid fluid hammering.

b) Provide the piston with the appropriate time required to complete the pressure and the return stroke.

c) Provide the check valves (inlet and outlet) with a sufficient no-flow time period to allow them to close before initiating the piston displacement in either direction.

At the rated speed of the rotary valve, the angular velocity is given by:

$$\omega = \frac{N \times 360^\circ/R}{60 \text{ SEC/MIN}} = 10,600^\circ/\text{SEC}$$

The check valve closing times have been set at t = .0012 second (minimum) which is equivalent to a displacement angle of

$$\alpha = \omega \times t = 10,600 \times .0012 = 12.7^\circ \text{ (MINIMUM)}$$

The piston stroke time has been set at T = .007 second (minimum), which in turn represents a displacement angle of

$$\beta = W \times T = 10,600 \times .007 = 74.2° \text{ (MINIMUM)}$$

The valve elements are arranged as shown in Figure 5-1, to provide complete pressure balance. Then:

$$\delta = \frac{360°}{N_p} - \alpha$$

$$\beta = 90° - \frac{360°}{N_p} + \delta = 90° - \alpha$$

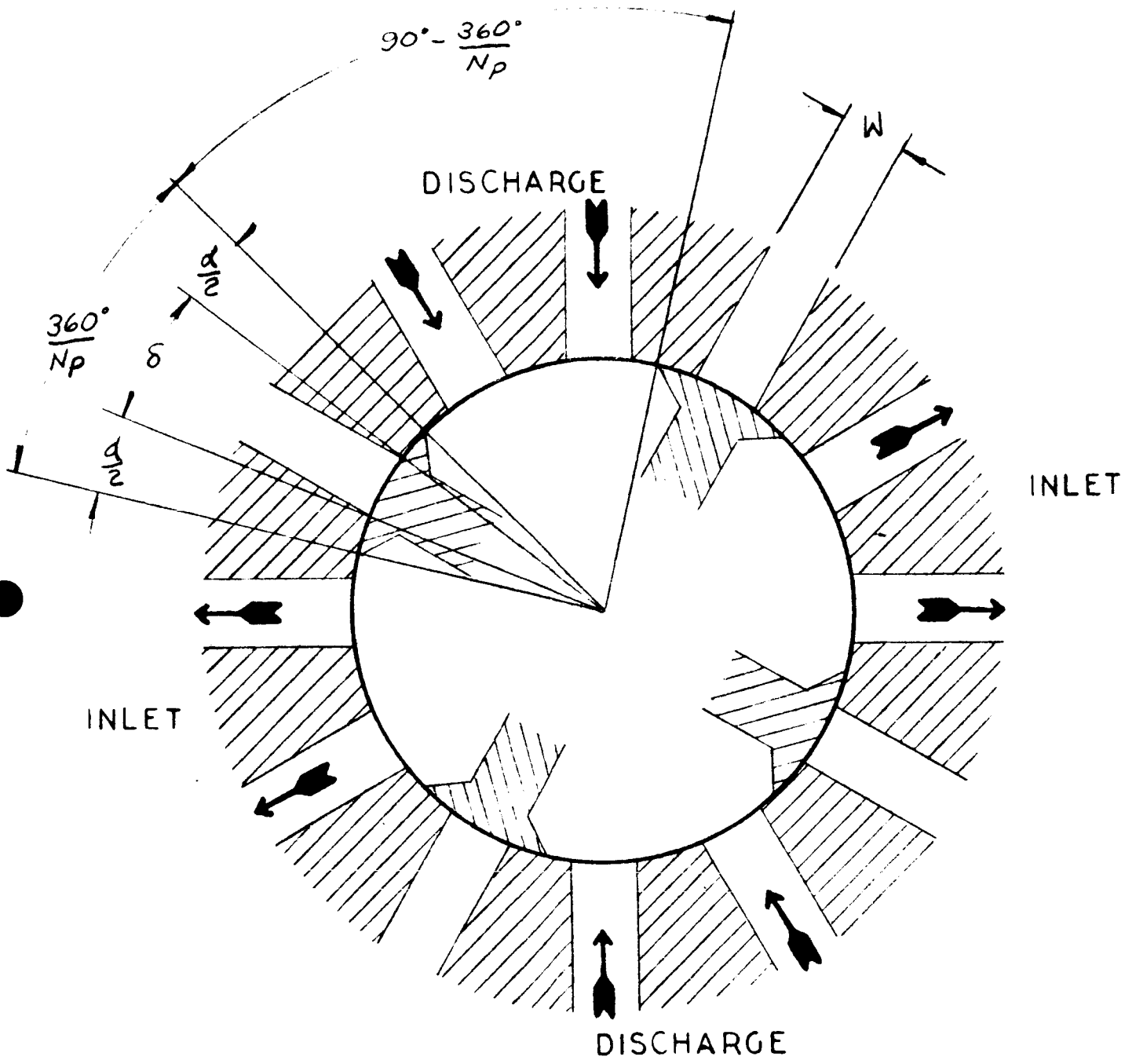


Figure 5-1

To satisfy these relationships, the following values are adopted:

$$\gamma = 14^\circ \qquad \beta = 76^\circ \qquad \delta = 16^\circ$$

which allows for the following operating time periods:

t = .00132 to .0012 sec.

T = .00725 to .0070 sec.

After the angular values are established, the dimension of the cylinder bore window width is related directly to the valve diameter. The size available leads to a convenient choice of window width:

W = .45 in.

All that remains to be determined is the window length, L, and hence the window area A_w. It is most convenient to establish the window area first, on the basis of a flow velocity low enough to give reasonably small flow pressure loss. Since fuel has a much lower viscosity than oil, we can allow higher velocities than we would with oil without incurring prohibitive pressure loss. Using a fuel flow velocity of

$$V_f = 45 \text{ ft/sec}$$

Associated with a flow per window of

$$q_F = \frac{q \, A_F}{z \, A_o(\eta_v)} = \frac{100}{4} \, \frac{1.090}{.460\,(.95)} = 62.5 \text{ GPM}$$

Then

$$A_W = \frac{q_F}{V_F} = \frac{62.5 \times 231}{45 \times 60 \times 12} = .445 \text{ IN}^2$$

and

$$L = \frac{A_W}{W} = \frac{.445}{.45} = .99 \text{ IN}^2$$

5.3 Check Valve Operation

In the preceding analysis, it was emphasized that the piston remains at rest during the period of check valve operation, and the rotary valve shuts off piston flow for the required period. The time of check valve operation thus represents time lost for pumping action, and hence should be as brief as possible. To accomplish this objective, the following criteria are established:

a) The mass of the valve poppet should be as low as possible consistent with the required structural demands.

b) The rate of the valve spring should be as high as possible consistent with the required stroke and actuating forces.

c) The poppet stroke should be as short as possible consistent with a reasonable poppet diameter.

In order to minimize pressure drop through the valve, it was desirable to keep flow velocity through the valve to a reasonably low value, in this case 20 ft./sec. Then the required valve open area may be established:

$$A_v = Q/V_c = \frac{17 \times 231}{20 \times 12 \times 60} = .273\ IN^2$$

Where:

A_V = Valve flow area

q = Rated oil flow per piston per stroke = 17 GPM

V_o = Oil flow velocity = 20 ft./sec.

Selecting a convenient poppet diameter:

D_P = .780 in.

The minimum valve stroke is given by:

$$S = \frac{A_V}{\pi D_P} = \frac{.273}{3.14 \times .780} = .111 \text{ IN.}$$

The combination poppet-spring will have a period of oscillation given by:

$$T_V = \frac{2\pi}{\sqrt{K/M}\ \sqrt{1-\xi^2}}$$

where:

T_V = Oscillating period of valve

M = Oscillating mass of valve

K = Spring rate

ξ = Damping ratio

The damping ratio used is defined as the ratio of the actual damping coefficient B to that value of B which will give critical damping.

$$\xi = \frac{B}{B_c} = \frac{B}{2\sqrt{KM}} = .30 \text{ (ESTIMATED)}$$

The expected performance from the check valve is based on an operating time of t = .0012 second.

Since the valve is supposed to travel a distance equal to S, in a time t, which is less than half of T_v, the full period. Let us choose:

$$T_v = 4t = .0048 \text{ SEC}$$

Then solving for the necessary spring rate K, using a mass:

$$M = 1.06 \times 10^{-4} \text{ LB-SEC}^2/\text{IN}$$

Thus,

$$K = \left(\frac{2\pi}{4t\sqrt{1-\xi^2}} \right)^2 M = \left(\frac{6.28}{4 \times 1.2 \times 10^{-3} \times .955} \right)^2 1.06 \times 10^{-4}$$

$$K = 1.888 \times 1.06 \times 10^{4} = 200 \frac{\text{LBS}}{\text{IN}}$$

For an inlet pressure P_i = 50 psig, the valve stroke will then be

$$S = \frac{P_i \times \frac{\pi}{4}(D_p)^2}{K} = \frac{50 \times \frac{\pi}{4}(.78)^2}{200} = .120 \text{ IN}$$

The actual valve opening stroke is thus larger than the minimum required, .111 in. stroke. The flow restriction of the valves has been minimized, while the time of operation coincides with the timing allowances of the rotary valve.

6.0 PERFORMANCE TEST RESULTS

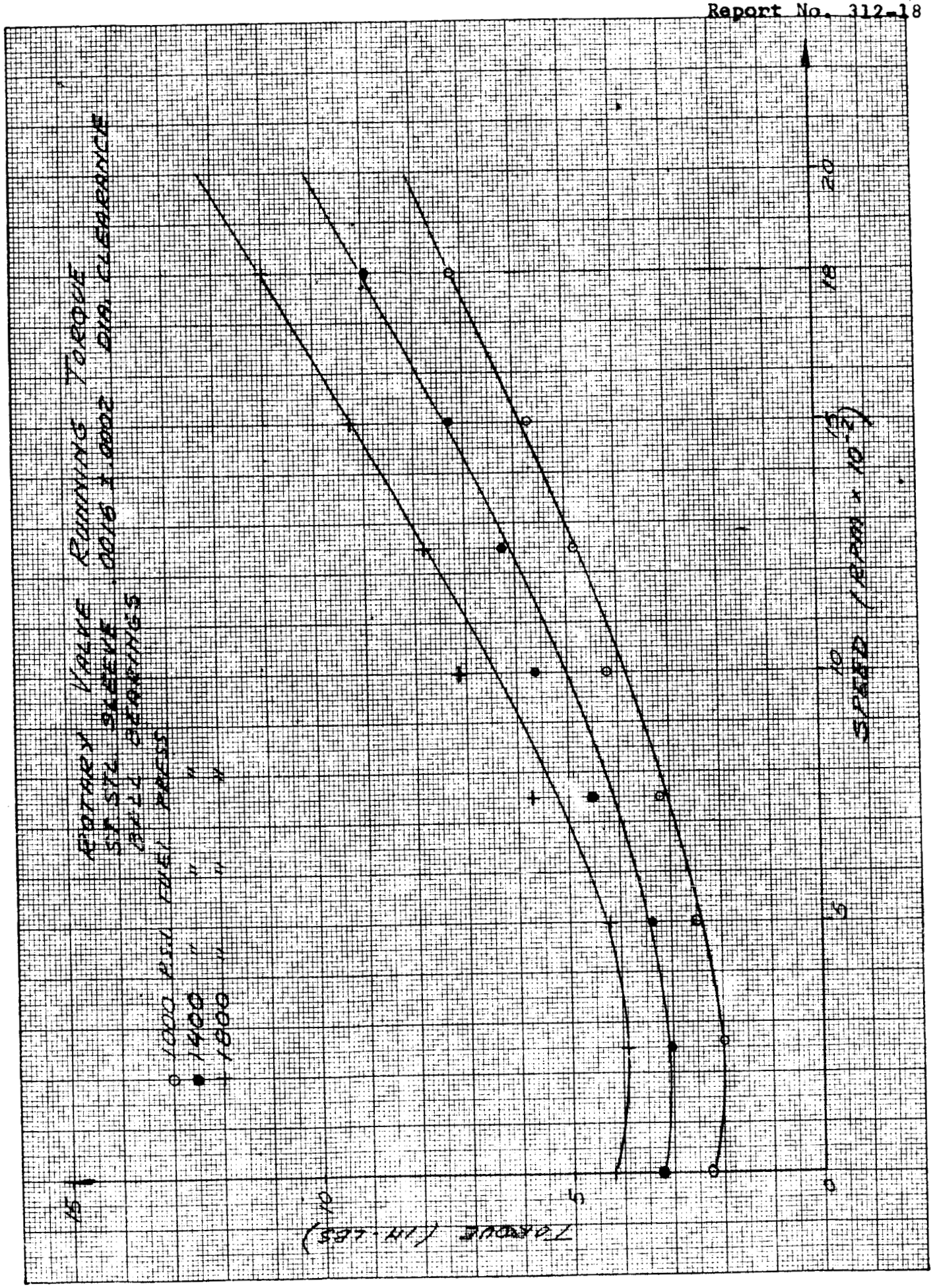
A great many tests were run in the course of the development program of the Hydraulic Transformer. Many tests were simply to aid in design or improvement of individual components. The tests described in this section are those which most directly relate to the operation of the final prototype Hydraulic Transformer units.

6.1 Rotary Valve Tests

The results of the rotary valve tests which are of main interest are those which concern torque requirements and fuel leakage flow. Earlier tests with the rotary valve had indicated that torque requirements, particularly for starting torque, were very high. It was subsequently found that the high torque was mainly due to unbalanced pressure forces on the valve, and also due to excessive torque in the bearings. The valve was redesigned to give better pressure balance and was mounted to run on ball bearings. The resulting torque requirements at various speeds and pressures are shown in Figure 6-1.

These torque results were obtained in the rotary valve test fixture, which did not have the pressure distortion problems that caused trouble in the prototype Hydraulic Transformer, and thus represent only normal operating torque loads. The general parabolic shape of the curves is in agreement with the theory of windage torque for a cylinder rotating in a viscous medium. The increase of torque with fuel pressure indicates some slight remaining pressure area unbalance, though the increase is small compared to the total value of torque.

The leakage flow of the rotary valve is shown in Figure 6-2 for various speeds and pressures. There is a rapid increase of leakage up to speeds of 300 RPM after which the flow increase is small as speed increases. Leakage flow is much more dependent on pressure and rotary valve diametrical clearance than on rotational speed.



K&E 10 X 10 TO THE CM. 359T-14G
KEUFFEL & ESSER CO. MADE IN U.S.A.
ALBANENE

Figure 6-1

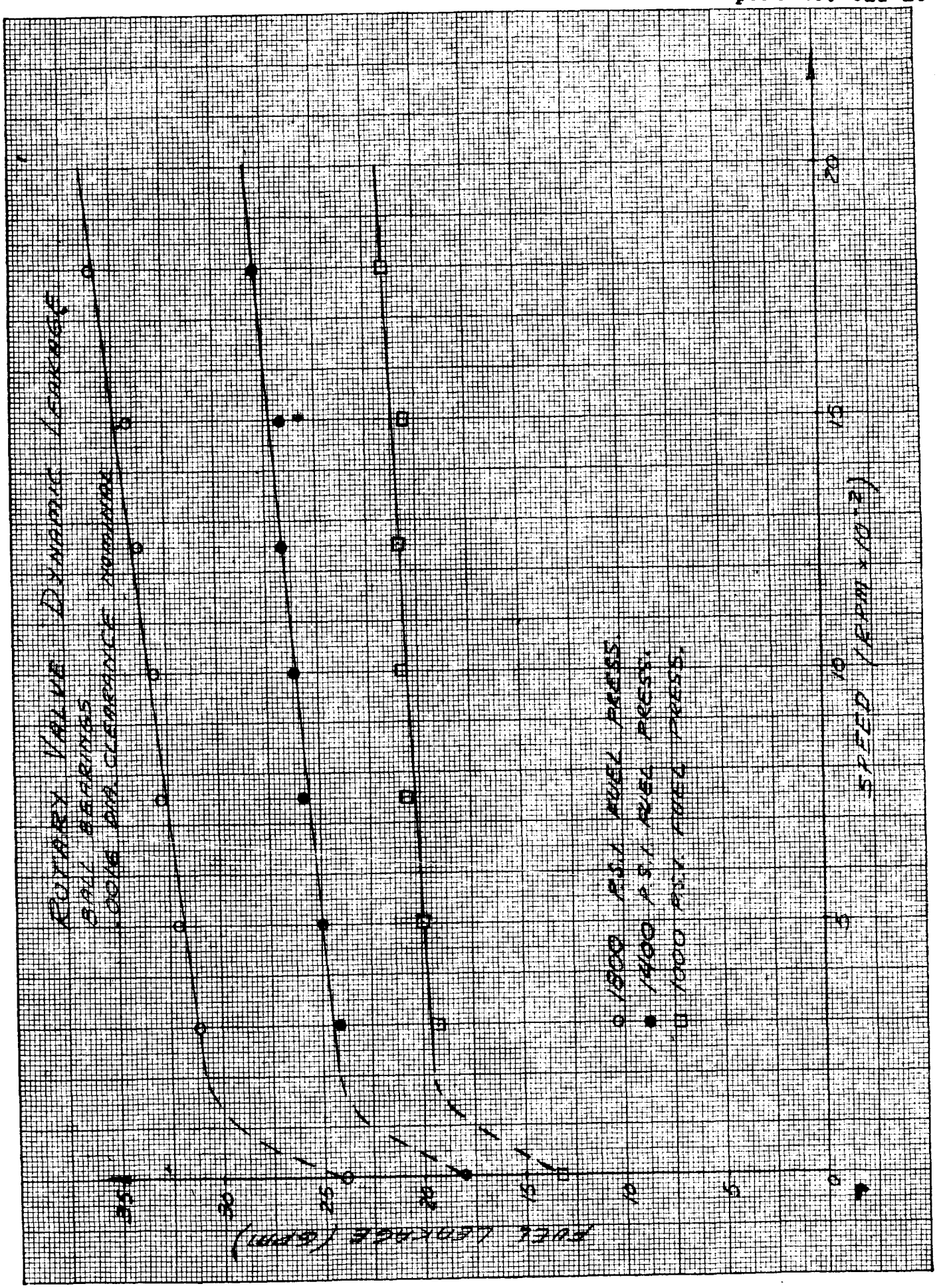


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Figure 6-2

6.2 Piston Tests

The single-piston test fixture was used to measure the dynamic behavior of a free piston to see if it conformed with the performance analysis. A special setup was made to record piston displacement directly as a function of time, and the instrument trace is shown in Figure 6-3.

The elapsed time for the selected stroke of 1.250 inches was .00816 seconds, somewhat over the design time of .0072 seconds. However, the increased time is mostly due to the additional weight of the apparatus attached to the pistons, a weight increase of approximately 14%. Graphical integration of the results as recorded give a maximum piston velocity of 187.5 in./sec. and an average velocity of 153.2 in./sec. The free piston travels at nearly constant maximum velocity over a large portion of its stroke, with rapid acceleration and deceleration at the ends of the stroke. Consequently, oil flow is also nearly constant over most of the piston stroke, in contrast to the sinusoidal form of output given by a typical hydraulic piston pump.

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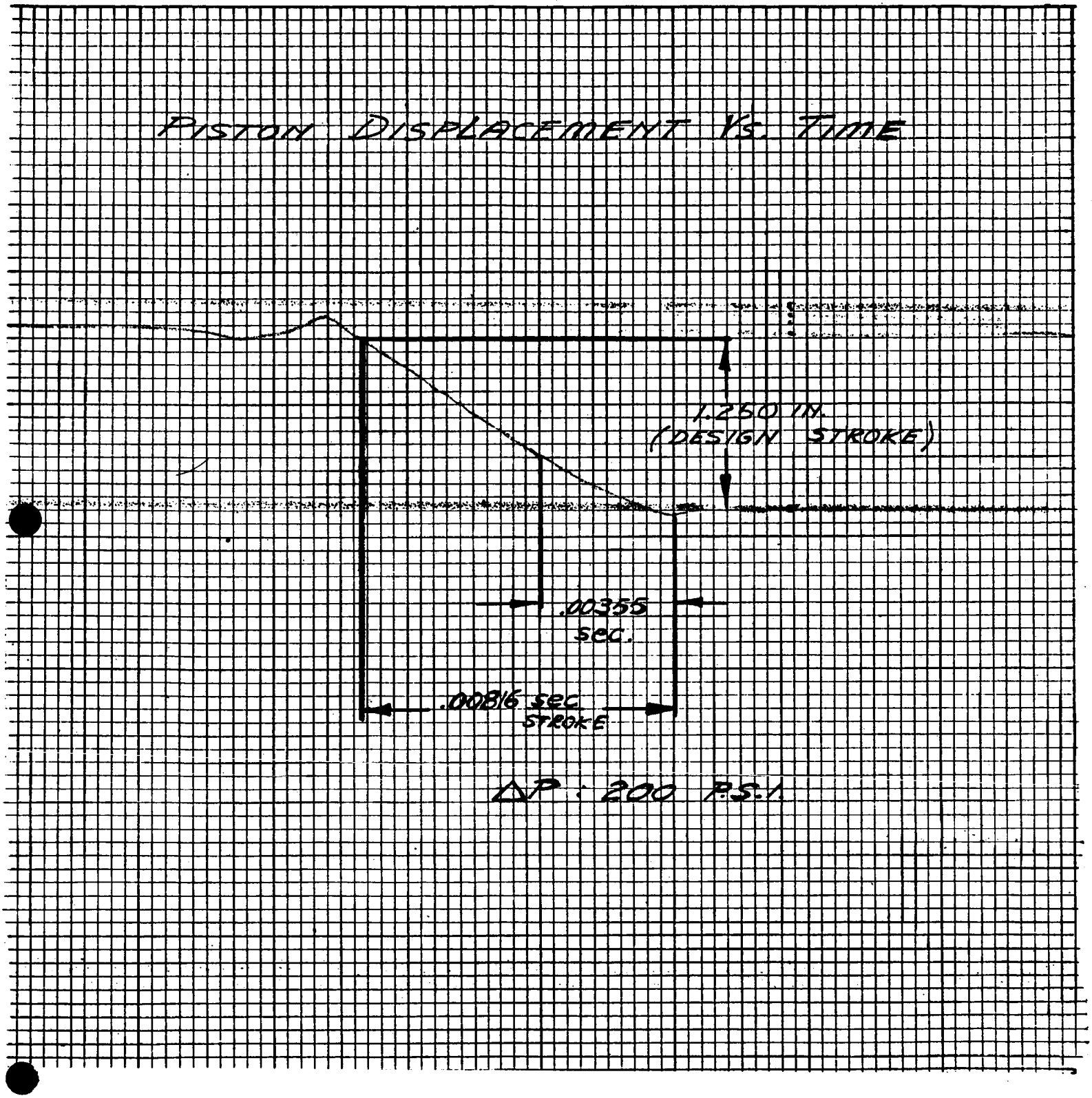

PART NO. _____ NEG. NO. _____ DESCRIPTION _____
NAME Piston Displacement Trace

Figure 6-3

6.3 Prototype Unit Tests

Much of the initial testing of the prototype units was directed toward diagnosing and preventing the binding of the rotary valve. This binding was found to be primarily due to distortion of the housing and fuel cap under pressure, and this condition was largely alleviated by use of the clamping fixture described in Section 4.4.

With the clamping assembly installed, performance tests could be run on the Hydraulic Transformer. However, one unit had already suffered a fractured housing due to a 5000 psi fuel pressure surge in the test facility. Although work was done to improve the test facility, subsequent tests on the remaining unit were run at reduced fuel pressure, starting at 500 psig, in an attempt to avoid further pressure damage.

Test results for tests at fuel pressures of 500 psig and 700 psig are presented in Figures 6-4 and 6-5. Two aspects of these test results warrant special attention: the great variation in rotary valve speed and the sharp drop of hydraulic oil pressure at high hydraulic flows.

DATE OF TEST 12-16-65

DATA PAGE NO. 41599

ENGINEER O. MODIG

HYDRO-AIRE COMPANY
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PART NO. Y05-055 UNIT NO. B-1

PAGE 6-8

REPORT NO. 312-18

DATE DRAWN 2-9-65

DRAWN BY V. TILENIUS

INLET FUEL PRESSURE 500 P.S.I.G.

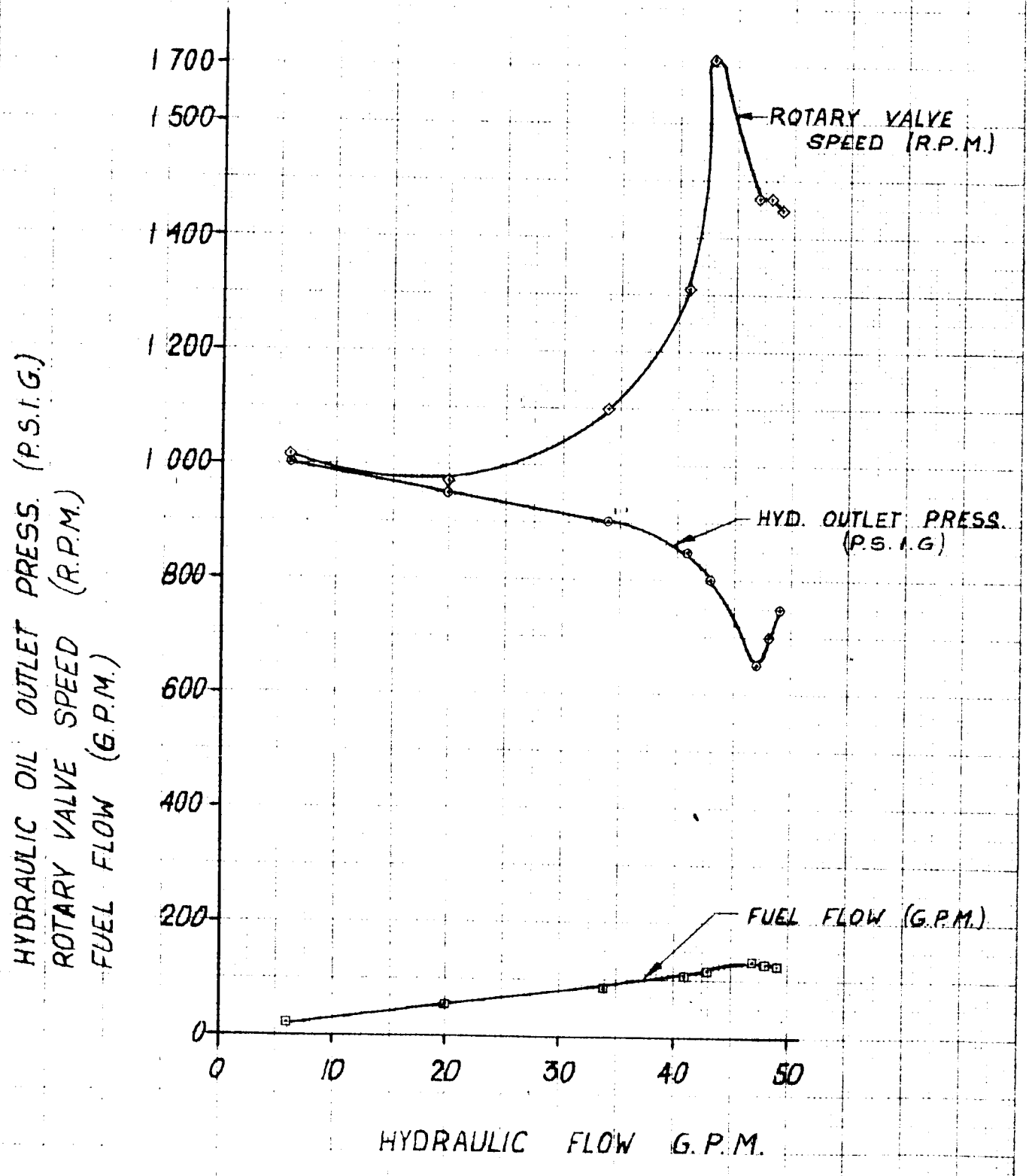

Figure 6-4

DATE OF TEST: 1-7-65

DATA PAGE NO.: 41608

ENGINEER: O. MODIG

HYDRO-AIRE COMPANY
A DIVISION OF CRANE CO.

PART NO.: Y05-055 UNIT NO.: B-1

PAGE 6-9

REPORT NO. 312-18

DATE DRAWN: 1-8-65

DRAWN BY: U. TILENIUS

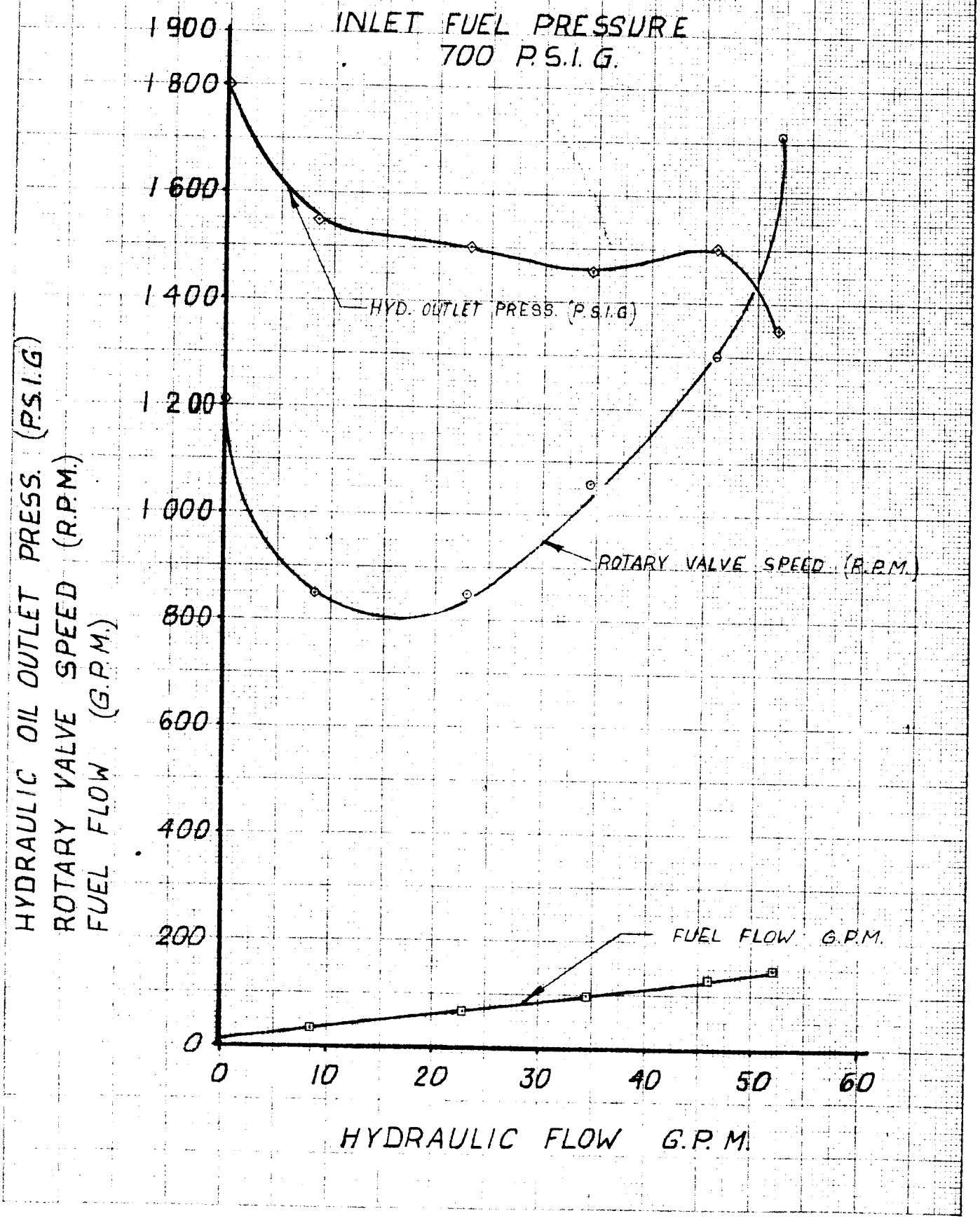

Figure 6-5

The rotary valve of the Hydraulic Transformer is driven by a hydraulic motor with a speed control, designed to run at constant speed. The fact that the rotary valve speed fluctuates so greatly indicates that the motor was not developing sufficient torque to keep the speed constant, or conversely, the rotary valve torque loads are overloading the motor and slowing it down. This effect is most prominent at low fuel flows, while at higher fuel flows the fuel tends to drive the rotary valve like a turbine, and thus at least partially alleviates the torque load on the motor.

These tests were run at low fuel pressure which directly reduces the motor torque available much below the design operating conditions. Nevertheless, torque loads were not expected to vary to such a degree. The reason for the excessive torque needed to drive the rotary valve has been explained by the fact that when the Hydraulic Transformer was disassembled there was still some evidence of rubbing between the rotary valve and the main sleeve as in earlier tests. Although rubbing marks were slight, it did appear that the clamping assembly was not completely successful in preventing deflections of the unit which caused contact under pressure.

With respect to the sharp drop of hydraulic oil pressure at the highest hydraulic oil flows, this defect was due to the fact

that the oil supply flow was limited by excessive leakage through a bypass line due to failure of a check valve in the test facility. Thus it was not possible to increase oil flow beyond 50 GPM even at reduced oil discharge pressure.

In order to correct the shortcomings of the previous tests, the oil supply circuit was revised and two changes were made to the Hydraulic Transformer:

a) The rotary valve was ground down to increase the diametrical clearance from the existing value of .0020 in. to a new nominal value of .0080.

b) The internal drive shaft of the rotary valve was removed and an external drive shaft was used, so that the rotary valve could be driven by an external hydraulic motor having more torque than the internal motor.

By grinding down the rotary valve, which thus gives increased diametrical clearance with the surrounding main sleeve, it was anticipated tnat any remaining binding problems would be eliminated, or at least substantially reduced. Of course, there was also the penalty of increased fuel leakage flow with some corresponding loss of efficiency. Nevertheless, this loss is known and can be accounted for in the performance evaluation.

By using an external motor drive for the rotary valve, it was possible to drive the rotary valve at constant speed in spite of changing torque loads. Further, by using a calibrated motor drive, it was possible to know the actual torque required by measuring the motor fluid supply pressure. The use of the external drive did not require any modification of the Hydraulic Transformer test unit, as the external drive shaft extended through the center of the rotary valve and was brought out through the side of the fuel supply pipe elbow.

As in previous tests, runs were first made at 500 psig fuel pressure and then increased to 700 psig with results shown in Figure 6-6 and 6-7. In these tests, rotary valve speed was maintained nearly constant, though it was necessary to vary the torque of the drive motor by varying the hydraulic fluid supply pressure as indicated. The reduced torque at high fuel flow is again due to the turbine effect of the rotary valve at large flow rates.

DATE OF TEST 1-11-65

DATA PAGE NO 43009
43011

ENGINEER O. MODIG

HYDRO-AIRE COMPANY
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PART NO. Y05-055 UNIT NO. B-1

REPORT NO. 312-18

DATE DRAWN 1-12-65

DRAWN BY U. TILENIUS

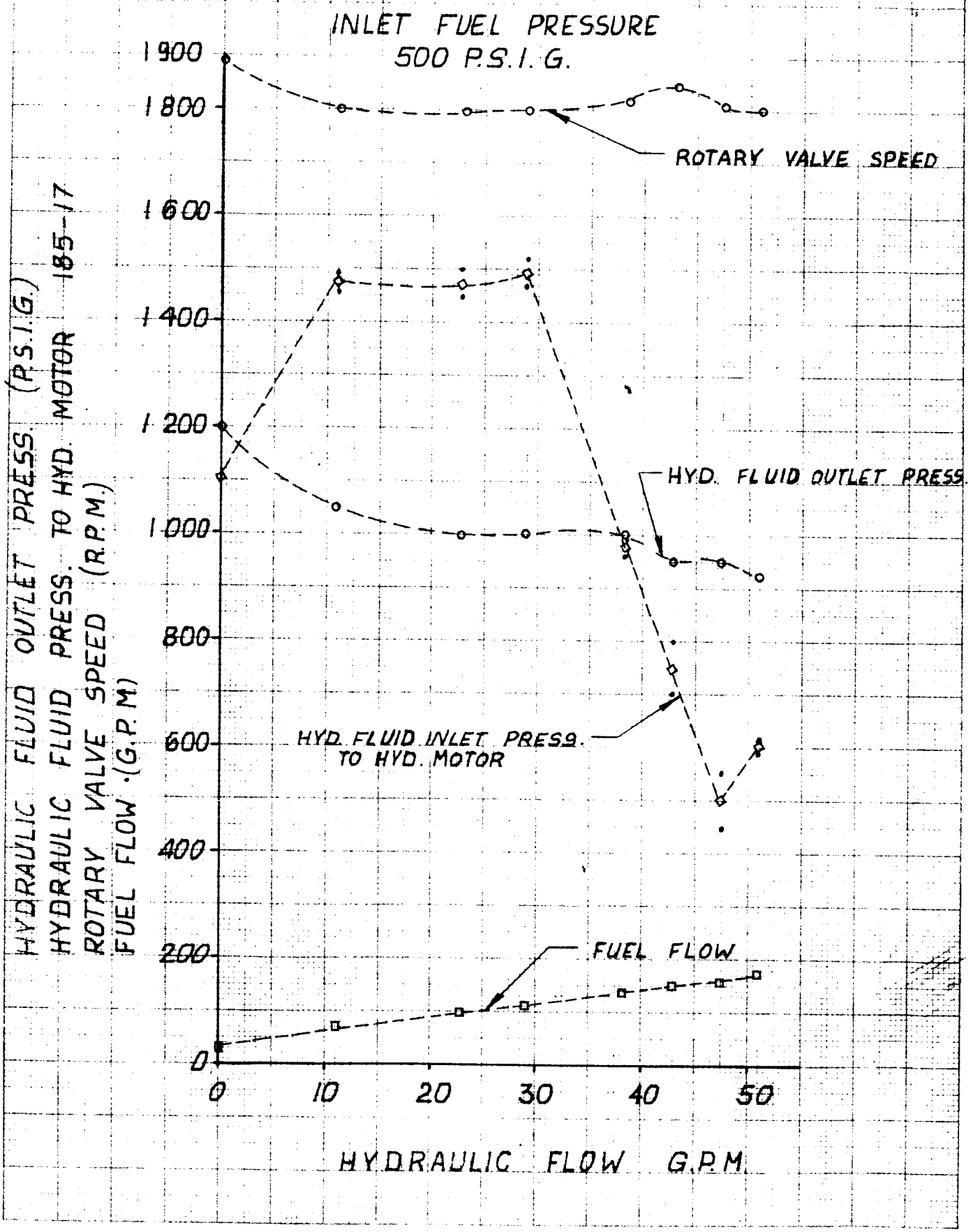

Figure 6-6

DATE OF TEST: 1-11-65

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PAGE 6-14

DATA PAGE NO.: 43009
43011

REPORT NO. 312-18

DATE DRAWN: 1-13-65

ENGINEER: O. MODIG

PART NO.: Y05-055 UNIT NO.: B-1

DRAWN BY: U. TILENIUS

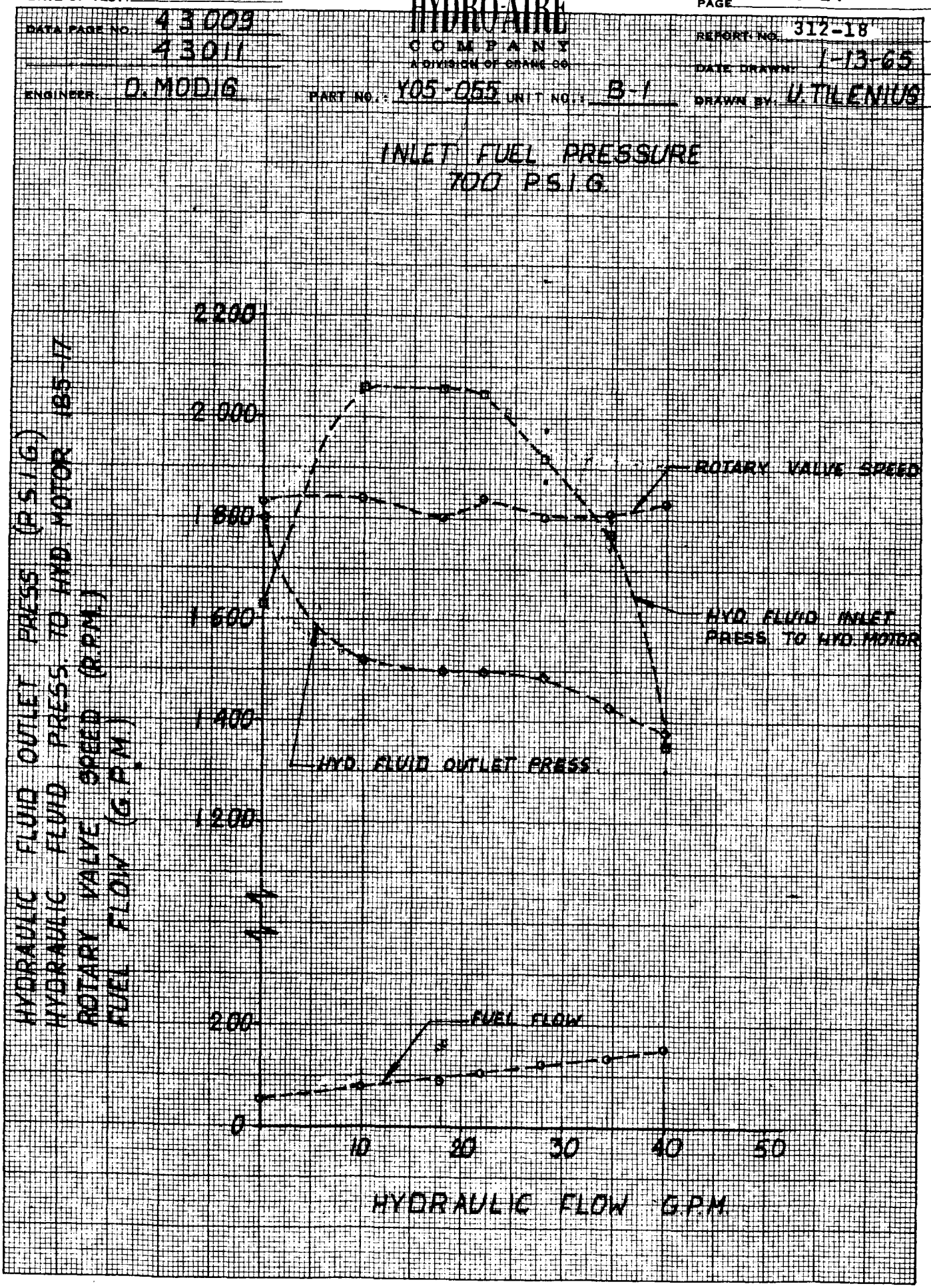

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Figure 6-7

The final test of the prototype Hydraulic Transformer was made at a fuel supply pressure of 1000 psig, and results are shown in Figure 6-8. Use of the ground-down rotary valve resulted in a fuel leakage flow of 65 GPM which limited the maximum Hydraulic Transformer overall efficiency to 59 percent for this test. If it had been possible to use the larger rotary valve, with smaller leakage flow, the efficiency should have been approximately 75 percent. At the conclusion of this test it was found that the unit would no longer develop hydraulic pressure and subsequent attempts to operate the unit were unsuccessful. The unit was disassembled and found to be damaged, apparently by a pressure surge during the last performance test.

DATE OF TEST: 1-13-65

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PAGE 6-16

DATA PAGE NO: 43 015
43 017

REPORT NO. 312-18

ENGINEER: O MODIG

PART NO.: Y05-055 UNIT NO.: Br 1

DATE DRAWN: 1-16-65

DRAWN BY: U TILENIUS

HYDRAULIC INLET PRESSURE: 100 P.S.I.G.
FUEL OUTLET PRESSURE: 80 P.S.I.G.
ROTARY VALVE SPEED: 1800 R.P.M.

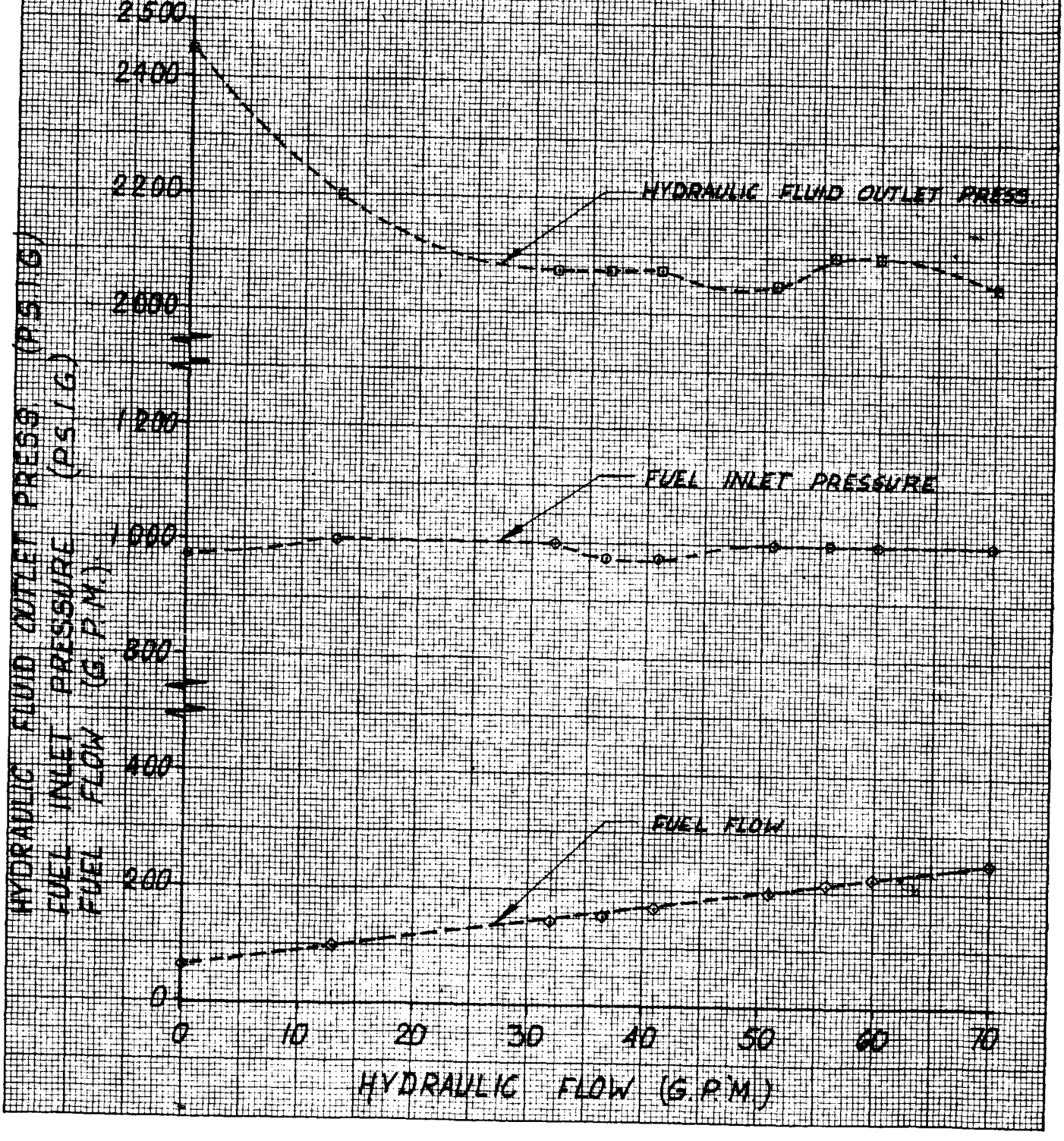

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Figure 6-8

6.4 Final Disassembly and Inspection of Prototype Unit

Upon removal of the hydraulic end cap, it was found that the housing had numerous cracks. All cracks appeared to originate at sharp corners in the hydraulic outlet cavity. Most cracks started in the section between the hydraulic piston bore and hydraulic outlet check valve, as this section contained the sharpest corners. Figure 6-9 is a typical illustration showing the crack between the No. 2 discharge bore and the No. 3 inlet bore.

There was also damage to other components, as ten of the pistons exhibited galling with their sleeves. The rotary valve had circumferential scribe marks, and the hydraulic motor showed evidence of a heavy side loading. These other areas of damage were considered to be a direct consequence of the housing failure and subsequent distortion.

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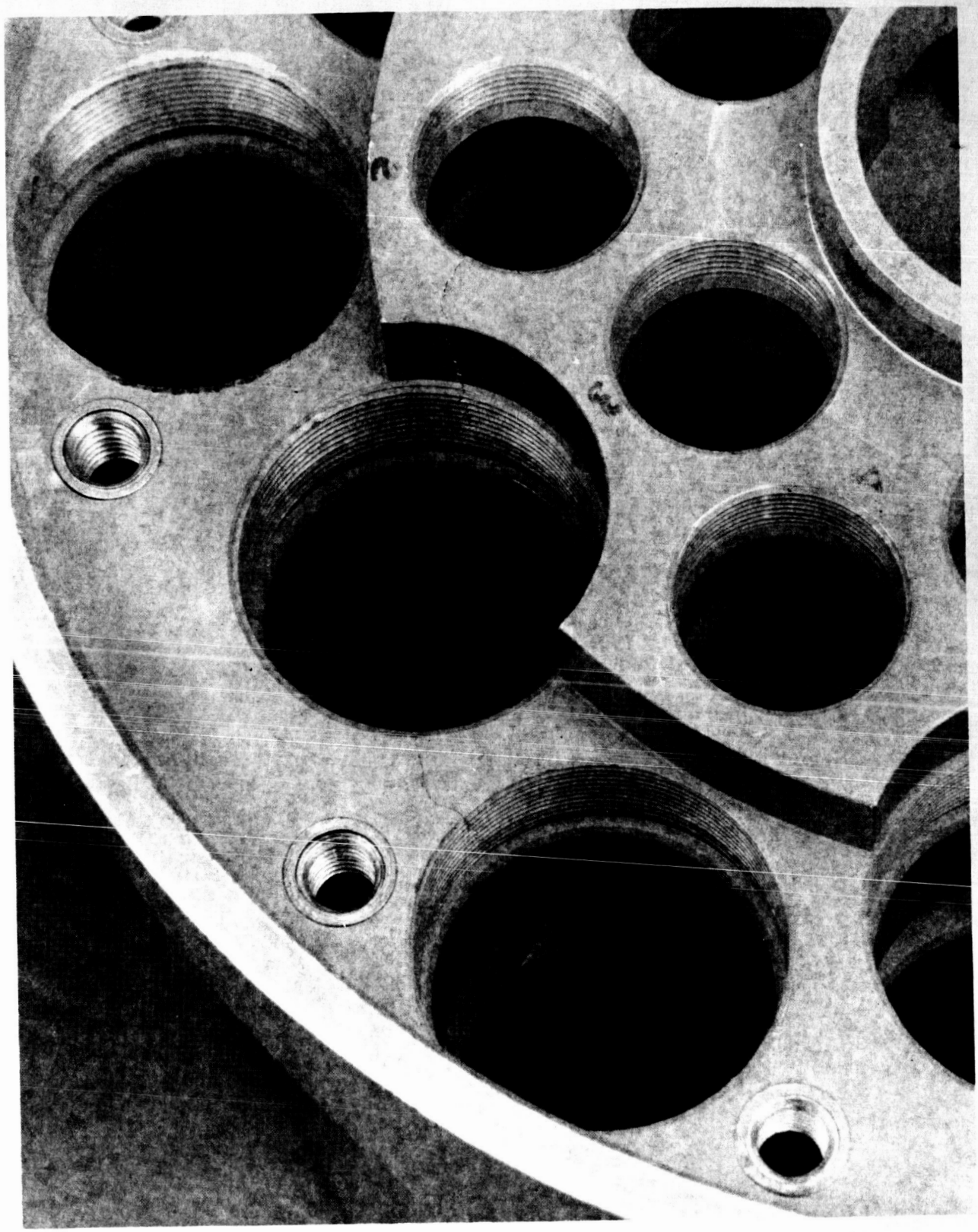
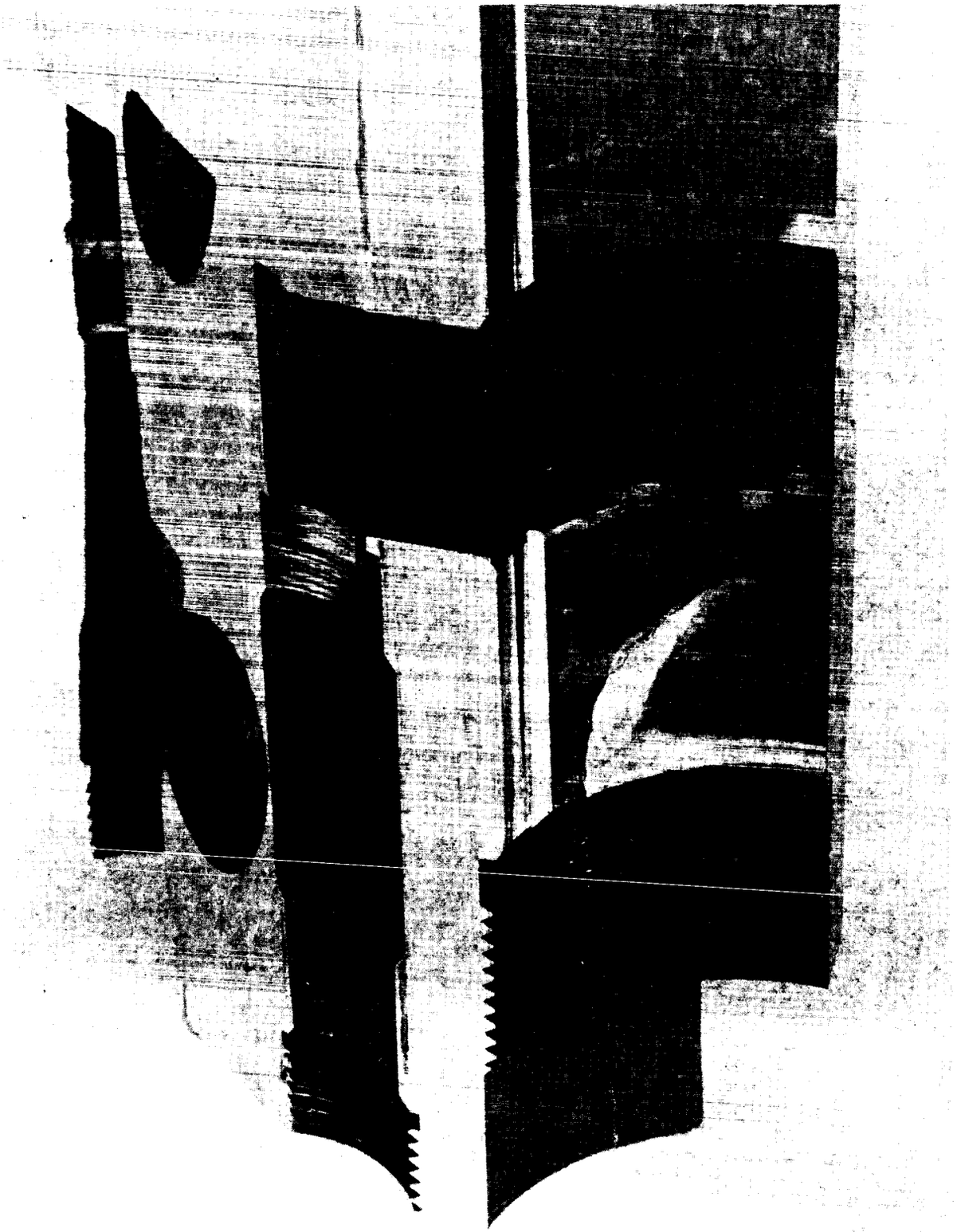


PART NO. 05-055 NEG. NO. 13034 DESCRIPTION Close-Up of Cracks in Housing
NAME Main Housing

Figure 6-9

The complete housing was sent to a materials testing laboratory. In summary, their report indicates that the housing material met the specifications as indicated on the drawings and as quoted by the material vendor. No evidence of stress corrosion or material strength decay due to heating of the housing for assembly of the rotary valve main sleeve was prevalent. Upon dissecting the housing, cracks were found to start at sharp corners, and to then lead off into heavier sections. Cracks were not longitudinal as would be expected in a high pressure burst, although displacement of fractured areas indicated longitudinal stresses were present. The major limiting factor in the design was found to be knife-edge sections resulting from intersecting holes.

A typical example of cracks in a dissected section is shown in Figure 6-10.



PART NO. 05-055 NEG. NO. 13058 DESCRIPTION Cracks at Hydraulic End of Main Housing
NAME Dissection of Main Housing

Figure 6-10

An analysis of the stresses induced in the hydraulic section of the main housing indicated that these stresses are beyond the strength capability of the material used. The high stress risers, in the form of sharp machined corners, together with the hydraulic loads applied, caused these stresses to be excessive. Due to the fact that this test unit underwent numerous test phases, it is believed that the fatigue factor contributed to the stress failure.

It is impossible to completely disassociate the various factors leading to failure. Nevertheless, it should be noted that the unit was certainly adequate for the rated operating pressure levels, and had in fact been subjected to static pressure testing of 4000 psig in the hydraulic outlet passage and 2000 psig in the fuel circuit prior to performance testing. With respect to pressure surges and fatigue, the stress risers were undoubtedly a critical shortcoming. The sharp corners were much more severe than might have been expected since the corner radii were .005 in. or less, while the drawing indicated a normal required tolerance of .015 to .040 in. corner radius.

The test during which 70 GPM hydraulic flow was obtained with 250 GPM fuel flow, was the last test where satisfactory performance resulted. It is suspected that at the time of high fuel flow, a

pressure pulsation occurred that was amplified by the test facility bypass relief valve, and transmitted into the Transformer. This pressure was subsequently amplified by the Transformer itself to cause an over-pressure on the hydraulic side. Evidence as to the fuel pressure surge is found in the fact that the safety pressure relief burst tube was deformed, indicating experience of pressure slightly below the 2230 psig burst pressure, and nearly double the fuel operating pressure for the tests being conducted.

The tests after the time of suspected failure served to cause unequal loading on the hydraulic pistons, which resulted in distortions and subsequent galling of the pistons and mating sleeves.