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DESIGN, DEVELOPMENT AND FABRICATION OF A PROTOTYPE HYDRAULIC TRANSFORMER

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ABSTRACT

This paper discusses the design, development and fabrication 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 built to handle this combination of flows and pressures, unprecedented in such devices, has a weight of only 70 pounds for the first development model. The development of this unit is discussed and future development improvements are mentioned.

INTRODUCTION

The specification requirements of 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. A 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 with a weight of over 100 pounds.

Instead of the 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 exhibits 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 development unit has a weight of only 70 pounds.

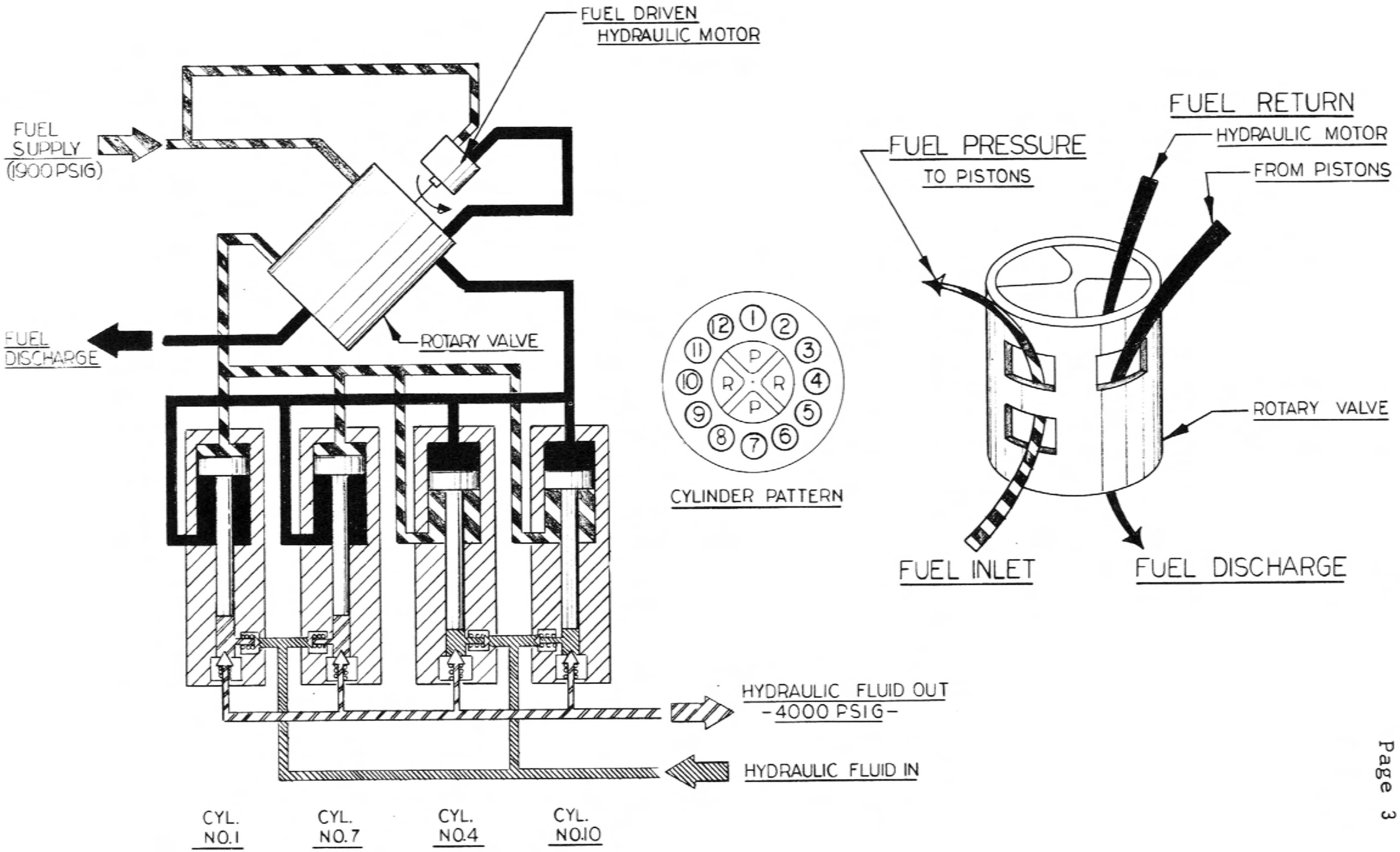

Figure 1 SCHEMATIC OF HYDRAULIC TRANSFORMER

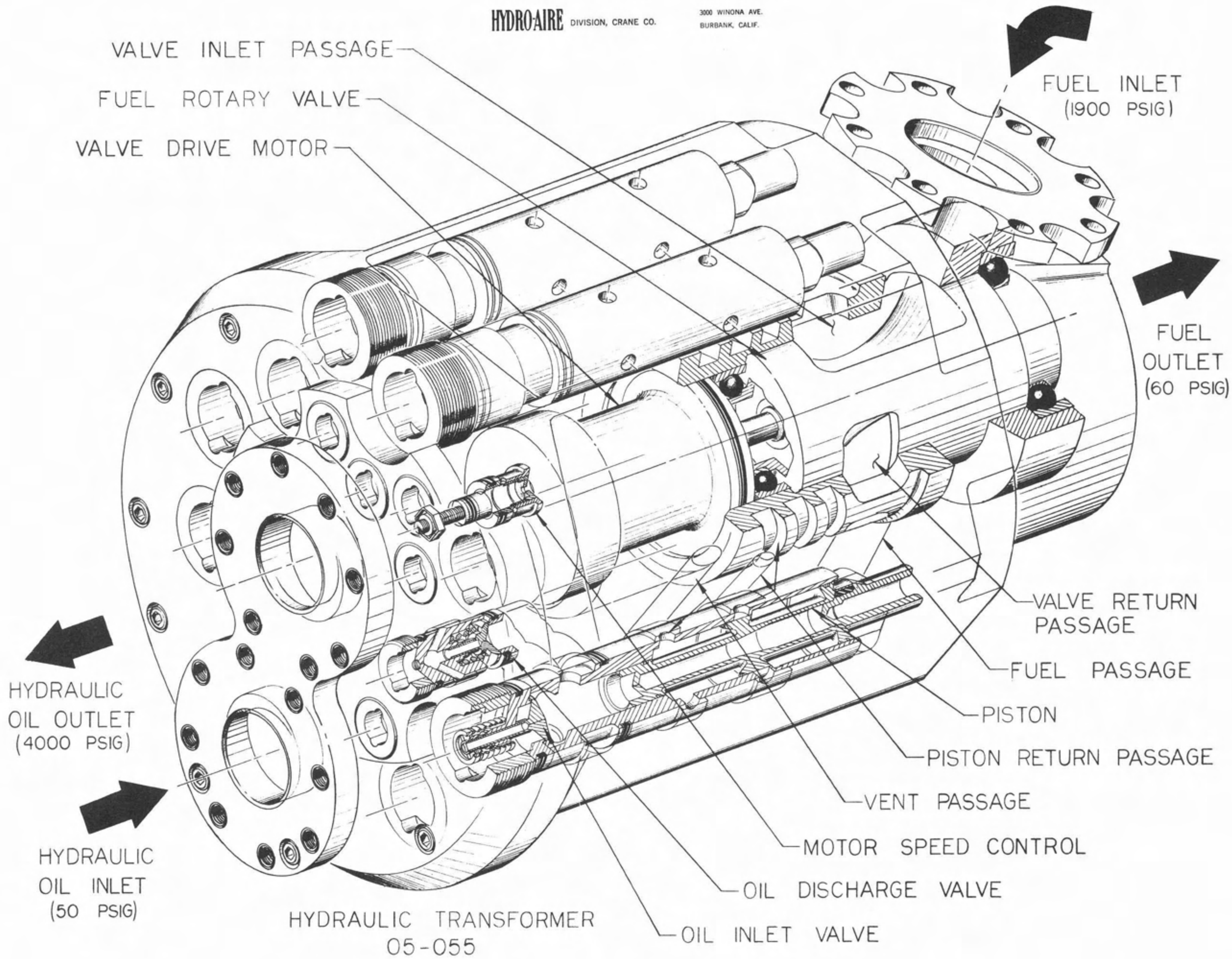

Figure 2 CUTAWAY VIEW OF HYDRAULIC TRANSFORMER

DESCRIPTION OF OPERATION

A schematic of the hydraulic transformer is shown in Figure 1, and a cutaway drawing is shown in Figure 2. 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 midpoint 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 12 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.

MECHANICAL DESIGN CONSIDERATIONS

Several features of the hydraulic transformer required special consideration on mechanical design. The rotary valve has a pressure differential of about 1850 psi between the fuel inlet and discharge passages, and thus must rotate within its surrounding ported sleeve with very small clearance, to prevent excessive leakage between inlet and discharge. In order to maintain this small clearance under operating conditions, the rotary valve must be very rigidly constructed to prevent deformation under the high differential pressure loads. Tolerances on dimensions and concentricities had to be carefully controlled.

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. This design also required maintaining very close tolerances and concentricities.

The main hydraulic transformer housing thus had to accommodate a number of parts to be assembled to very close tolerances. The complexity of the housing may be envisioned by noting the view shown on Figure 4. Machining the first housing unit out of bar stock, without special tooling, was for the machinist a traumatic experience.

DEVELOPMENT WORK

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, were found to be satisfactory with very little design modification.

Development of the fuel rotary valve required considerable development work because of the numerous parameters 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 is 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 best for initial development.

Several rotary valve configurations were tested to arrive at a design having low pressure drop and balanced pressure forces. The valve was also improved to give better rigidity and less deformation in operation. With these improvements the valve could be operated with smaller running clearances between the valve body and the surrounding ported sleeve, thus reducing leakage flow.

Another feature of the rotary valve requiring some modification was choice of valve bearings. Initially, roller bearings were used, but for several reasons the resulting starting torque at high fuel pressure was too great for the hydraulic drive motor. A change was made to ball bearings and the problem was eliminated.

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 return design was the reduction in the unit reliability. The pistons would be traveling at a rate of 5000 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. This 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. 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.

Another problem of the piston assembly was that of weight. To keep the inertia forces to a minimum, the weight of the piston must be as low as possible and still maintain the strength and rigidity required. To optimize the piston design, relationships between the various parameters, listed below, were studied:

a) Piston inertia forces.

b) Friction between the rubbing surfaces required to control the leakage flow.

c) Hydraulic springing given by the fluid compressibility.

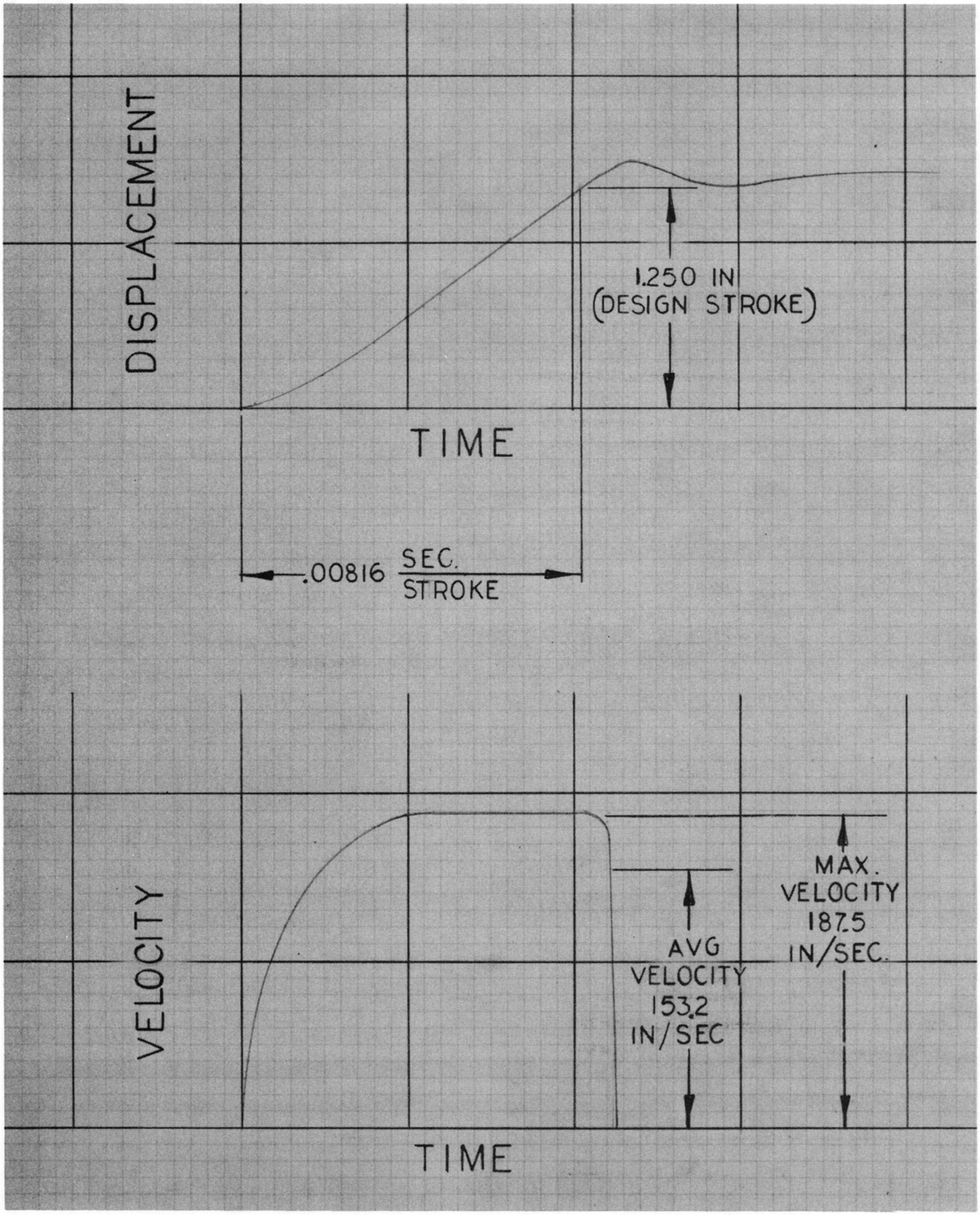

Figure 3 DISPLACEMENT AND VELOCITY VS. TIME IN HYDRAULIC TRANSFORMER PISTON TEST

d) Required area ratio under the different conditions of power stroke, return stroke and shutoff.

e) Rotary valve speed required to provide proper timing for the piston stroke.

A special test setup was made to record piston displacement vs. time, while piston velocity and acceleration were obtained by graphical differentiation. Results are shown in Figure 3 for one of the test pistons. Due to the operation of the discharge check valve, it may be seen that piston velocity, and hence oil flow, is nearly constant over most of the piston stroke. This is in contrast to the sinusoidal form of output given by a typical hydraulic piston pump.

HYDRAULIC TRANSFORMER DEVELOPMENT UNIT

The complete hydraulic transformer development unit is shown as a parts array in Figure 4 and fully assembled in Figure 5. The test setup required to supply the flows and pressures required for this unit is necessarily quite substantial and is shown partially in Figure 6. The purchased main fuel supply pump and motor drive is, unfortunately, not yet fully operable; and without the use of this special equipment, unit performance could only be

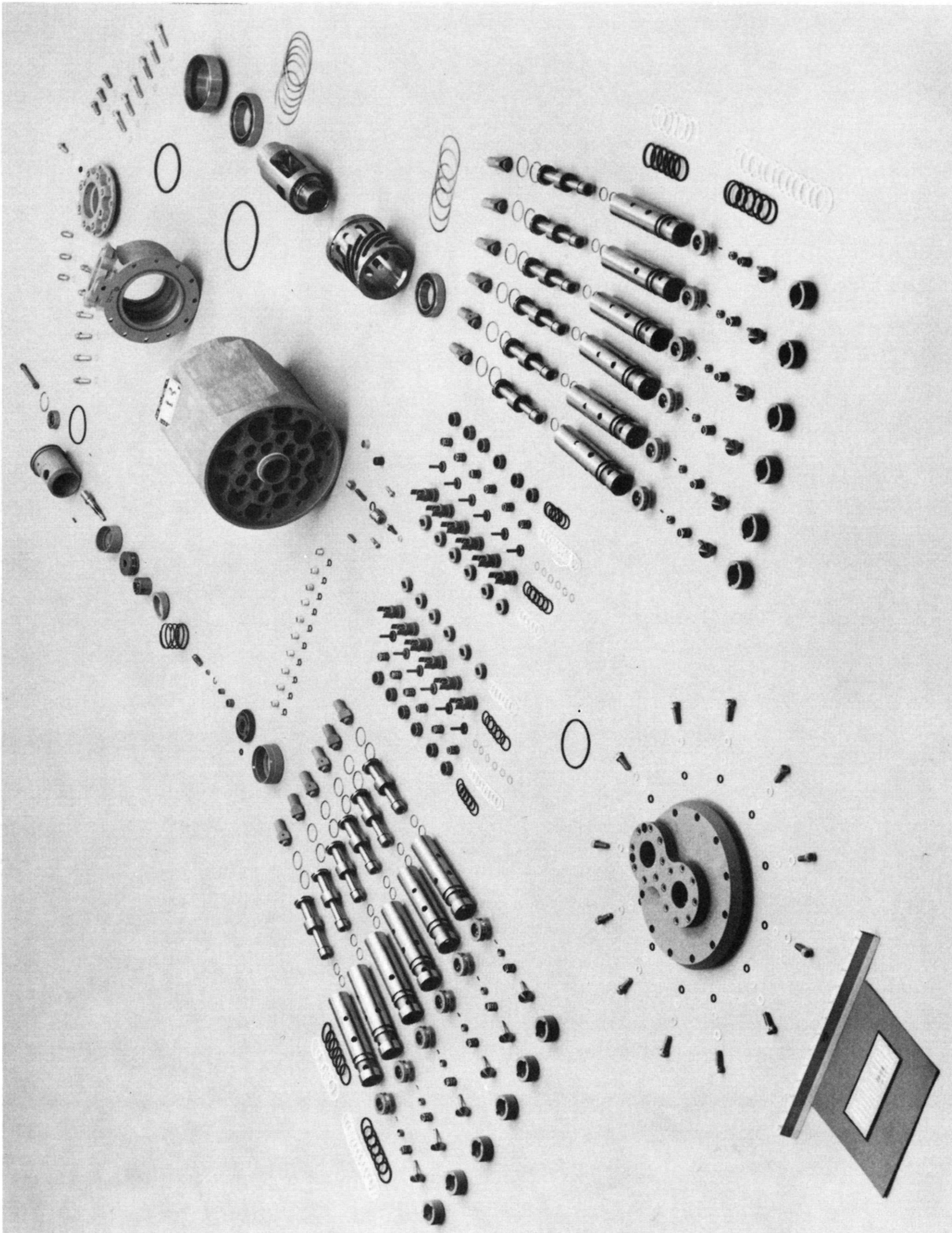

Figure 4   PARTS ARRAY OF HYDRAULIC TRANSFORMER

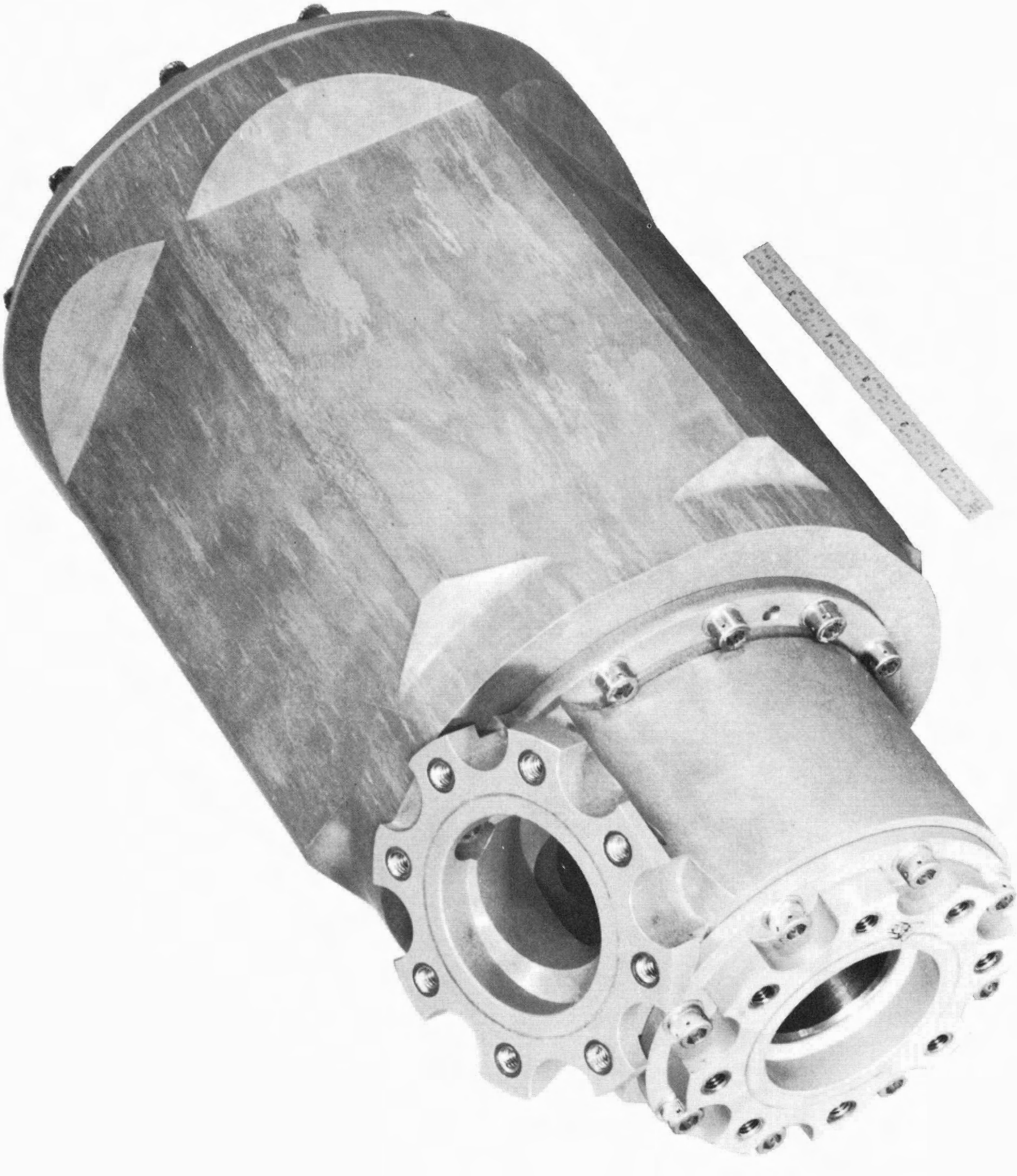

Figure 5 HYDRAULIC TRANSFORMER DEVELOPMENT UNIT

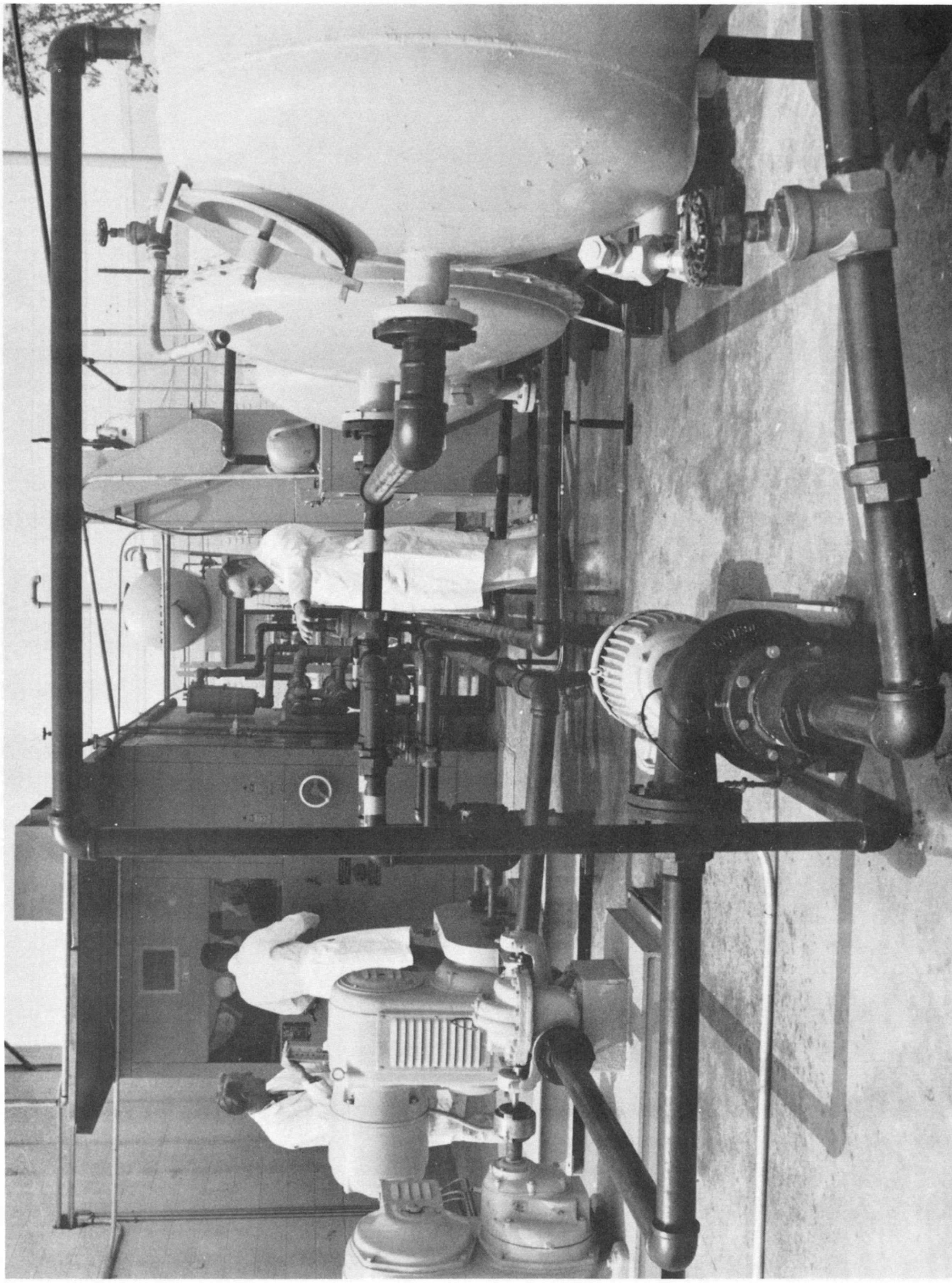

Figure 6 PARTIAL VIEW OF HYDRAULIC TRANSFORMER TEST SETUP

tested under very limited conditions. From such preliminary testing it appears that the unit is functioning properly and that overall efficiency should be about 69%.

DESIGN IMPROVEMENT

During the development program of the hydraulic transformer, many things were learned which could be applied to improvement of the unit. A number of these things, however, gave rise to ideas that could only be developed in a completely new design configuration, which was not possible in the time allowed in the original program. A new program has now commenced to produce an improved design.

One major design improvement is in making the pistons double-acting, so that they pump oil on both thrust and return strokes. The initial hydraulic transformer design used single-acting pistons to simplify development problems, but once the basic problems of piston action were solved, double-acting pistons presented several obvious advantages:

a) Pumping on both thrust and return strokes eliminates the wasted effort of returning the piston without any useful output.

b) The double-acting pistons can use oil-side dashpots on both thrust and return strokes. This leads to improvements both in damping, due to the higher viscosity of the oil, and in manufacture, due to the simplification of the design.

c) The improved piston configuration makes it possible to reduce the piston size. The smaller pistons in turn allow an overall reduction in the size of the hydraulic transformer.

d) The smaller and hence lighter pistons allow an increase in operating speed, improving efficiency.

e) The combined effect of double-action for the pistons, smaller pistons, and faster acting pistons, allows a reduction in the number of pistons. This allows a big reduction in size and weight of the hydraulic transformer.

Another design change which is contemplated is the replacement of the cylindrical rotary valve with a face-type rotary valve. The improvements anticipated with this type of fuel valve are:

a) Reduction of internal fuel leakage from the inlet to the outlet because the face-type valve can be operated with closer running clearances than the present cylindrical version.

b) Due to greater simplicity of construction, weight of the valve is reduced.

c) The greater simplicity of construction should also allow a reduction in cost of manufacture of the rotary valve.

The use of face-type rotary valves requires careful pressure balancing in order to maintain the very small running clearances contemplated. The operation of the valve must be carefully analyzed to achieve optimum pressure balancing if the full advantage of this type of design is to be utilized.

A further improvement in valving is considered by replacing the oil check valves with an oil-side rotary face valve. The operation of such a valve requires close synchronization with the fuel rotary valve so that proper piston travel is achieved. Such synchronization can be achieved by driving both the fuel valve and the oil valve with the same motor, using a common drive shaft.

CONCLUSION

The development of the high power hydraulic transformer has shown that the expectations for this type of unit are justified. Weight, size and performance are such that the hydraulic transformer has definite advantages over more conventional methods of obtaining high pressure hydraulic power by the utilization of a lower pressure hydraulic energy source. Furthermore, future developments should be able to give substantial improvement over that of the present development unit.