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ROTATING AND POSITIVE-DISPLACEMENT PUMPS FOR LOW-THRUST ROCKET ENGINES

FINAL REPORT
VOLUME II
FABRICATION AND TESTING

ROCKETDYNE DIVISION
ROCKWELL INTERNATIONAL
CANOGA PARK, CALIFORNIA

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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16. Abstract

Rotating and positive displacement pumps of various types were studied for pumping liquid fluorine for low thrust high performance rocket engines. Included in the analysis were: centrifugal, pitot, Barske, Tesla, drag, gear, vane, axial piston, radial piston, diaphragm and helirotor pump concepts. The centrifugal and gear pumps were carried through detail design and fabrication. After preliminary testing in Freon 12, the centrifugal pump was selected for further testing and development. It was tested in Freon 12 to obtain the hydrodynamic performance. Tests were also conducted in liquid fluorine to demonstrate chemical compatibility.

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FOREWORD

This technical report presents the results of an investigation of rotating and positive displacement pumps for low thrust rocket engines. The program conducted by Rocketdyne, a Division of Rockwell International, during the period of July 1968 to March 1971, was authorized by the Lewis Research Center of the National Aeronautics and Space Administration (NASA) under Contract NAS3-12022. The NASA project manager was Mr. W. Britsch.

This report is submitted as Rocketdyne report number R-8494, in two volumes:

I. Pump Evaluation and Design

- II. Pump Fabrication and Testing

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Overall program direction was provided by Mr. H. G. Diem, the Rocketdyne Program Manager.

Dr. O. E. Balje provided valuable consultation, in the preliminary analysis of the pitot, drag, diaphragm and helirotor pump concepts.

The evaluation of various pump concepts, as well as selection, design and fabrication of the rotating and positive displacement pumps was accomplished under the technical direction of Mr. C. A. MacGregor.

Testing of the selected pump concepts and preparation of Volume II were accomplished under the technical direction of Mr. A. Csomor.

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INTRODUCTION

Studies of future space missions show the requirement for relatively low thrust (1000 to 10,000 lb, 4448 to 44,480 N), high specific-impulse rocket engines in upper stages for orbital maneuvering and instrument package landers. For long-duration missions, or for vehicles having limited tankage volume, the use of high energy, space-storable propellant combinations such as methane-FLOX (fluorine-oxygen mixture) is desirable.

To realize the full potential of the high-energy propellants, it is necessary to achieve the lowest practicable values for engine size and weight and tank weight, thereby maximizing payload. A means for optimizing the performance, size and weight of the vehicle is the use of a high-pressure, pump-fed engine cycle.

At the thrust level of 1000 lb (4448 N) and a chamber pressure of 1000 psia (689.5 N/cm^2) the FLOX flowrate is 12 gpm (7.57×10^{-4} m^3/s) assuming an attainable specific impulse of 385 sec (3775 Ns/kg) at a mixture ratio of 5.75. A chamber pressure of 1000 psia will require a pump discharge pressure of approximately 1500 psia (1030 N/cm^2). Since the use of a separate low-speed inducer was not considered practical for these small engines, the maximum speed of the pump was limited primarily by cavitation requirements. The combination of flowrate, pressure rise, and fluid density together with the allowable range of speeds places the pump in the low specific speed regime where the predicted attainable efficiencies for several types of centrifugal pumps are below those of the positive displacement types.

The predicted efficiencies of centrifugal pumps available in the general literature are results obtained on pumps having flowrates in excess of 100 gpm (6.31×10^{-3} m^3/s). Extrapolation of these data to the flowrate required inherently assumes both geometric and hydrodynamic similarities are maintained. In these small size ranges, geometric similarity cannot be maintained. In contrast, the low viscosity of fluorine makes it possible to maintain a high Reynolds No.

despite the small size of the impeller. However, surface roughness and clearances that are practicably attainable become a significant percentage of the characteristic dimensions and can noticeably affect the efficiency. The inability to maintain geometric similarity can also make it difficult to attain the necessary cavitation performance. Because of these uncertainties in predicting the efficiencies and suction performance, test data were needed to establish the performance characteristics of rotating pumps operating under the hydrodynamic conditions imposed by these low-thrust rocket engines using high-energy propellants.

The capability of operating centrifugal pump impellers in fluorine mixtures was previously established (Ref. 1). In those tests, the impeller cavity and the bearing cavity were separated by a three-element seal to prevent mixing of the organic lubricant and the FLOX mixtures. The desire to alleviate critical speed problems in the 12-gpm (7.57×10^{-4} m^3/s) pump led to the decision to operate the bearings in liquid fluorine, thus eliminating one seal. Although bearings have been tested in fluorine, pump bearings have not previously been subjected to the fluorine. Tests to determine the feasibility of this design approach are necessary.

The scope of the program consisted of the following four areas of effort:

1. Evaluation of various concepts of small, lightweight, efficient rotating and positive displacement pumps, and on the basis of this evaluation select the best single design of positive displacement pump and the best single design of rotating pump. (Task I and Task V)
2. Design the two selected concepts. (Task II and Task VI)
3. Fabricate and assemble two rotating and positive displacement pumps. (Task III and Task VII)
4. Test the two pump designs in both Freon 12 and LF_2 to establish the mechanical integrity, and determine the pump performance and operating characteristics, and the fluorine compatibility of the drive elements (Task IV and VIII)

Volume I of this report describes the work accomplished in areas (1) and (2) above. Volume II describes the effort of (3) and (4).

For purposes of identification, the centrifugal pump was designated as "Mark 36 Pump" and the gear pump as "Mark 37 Pump."

SUMMARY

Rotating and positive displacement pumps of various types were studied for pumping fluorine mixtures for low thrust, high specific impulse rocket engines, with the following specific requirements:

• Inlet Pressure	35 psia	(2.41×10^5 N/m^2)
• Inlet Temperature	159 R	(88.4 K)
• Discharge Pressure	1500 psia	(1.03×10^7 N/m^2)
• Flow	12 gpm	(7.57×10^{-4} m^3/s)
• Fluid	Liquid Fluorine	

Analysis and preliminary layouts were made of the following rotating pump concepts: centrifugal pump, pitot, Barske, Tesla, and drag pumps. Comparison of the different concepts was made on the basis of performance, weight, reliability, cost and life. The centrifugal pump was selected as the rotating pump concept most suitable for this application. In a similar manner, positive displacement pumps consisting of gear, vane, axial piston, radial piston, diaphragm and helirotor types were evaluated. The evaluation criteria indicated that the gear pump and the vane pump were equally suitable candidates for this application. The gear pump was selected because the rubbing velocities at the points of contact were lower than those encountered in the vane pump.

Detailed analysis and design of the centrifugal and the gear pump were made and two sets of hardware were fabricated for each of these pump types.

During the initial test in Freon-12, the mechanical difficulites encountered with the gear pump precluded further test with it, and no further development of this pump was undertaken. The remainder of the contract effort was limited to the centrifugal pump.

The centrifugal pump was tested in Freon-12 for a total of 2.94 hours at speeds up to 80,000 rpm with a maximum discharge pressure and flow of 2000 psig and 14.5 gpm, respectively. Thirty start sequences were conducted during these tests. The overall efficiency at design flow was 52 percent. A maximum suction specific speed of 36,500 was achieved.

Subsequently, the centrifugal pump was tested in liquid fluorine for an accumulated time of 387 seconds under steady-state conditions at speeds up to 40,000 rpm. Some brief transient speed excursions to 104,000 rpm were encountered. During these fluorine tests, the ball bearings experienced excessive wear rates. These diametral wear rates were 0.004 in./hr (0.012 cm/hr) on the front bearing and 0.012 in./hr (0.0305 cm/hr) on the rear bearing. (This is in contrast to the Freon tests where no ball wear was experienced.) These wear rates are believed to be the cause of rotor instabilities which precluded tests above 40,000 rpm.

Funding limitations did not permit resolution of the bearing wear problems. No other deleterious effects due to the liquid fluorine were observed during these limited tests.

FABRICATION AND ASSEMBLY

Most of the major parts of the centrifugal pump were machined from wrought INCO 718 alloy, heat treated to obtain a minimum room temperature ultimate strength of 180,000 lb/in.^{2} (1.24 x 10^9 N/m^2). Included in this group were the inlet housing, volute housing, inducer, shaft, and dynamic seal housings. The impeller was cast from INCO 718 by the lost wax process as opposed to sand casting, which is used with conventional size impellers. In addition to the size consideration, the lost was process was also necessary to maintain tight casting tolerances and, thereby, reduce part-to-part performance variation.

Figure 1 illustrates the detail parts of the centrifugal pump prior to initial assembly. The drive turbine components are not included in this photograph.

Balancing of the rotor assembly was accomplished on a Gisholt Type U-11-1113 balancing machine at a speed of 2950 rpm. Figure 2 and 3 illustrate the installation of the rotor in the balance machine. Components of the rotor assembly which are not fixed in angular position relative to the shaft, such as the inducer and impeller retaining nut, were individually check balanced and corrected as required. Final two-plane balance correction for the complete rotor assembly was effected by removing material from the impeller, inboard of the rear wear ring, and from the tips of the turbine disk vanes. The residual unbalance was reduced to the machine limit which, for a rotor of this size, is approximately 0.01 gm-in. (0.0249 x 10^{-6} N-m).

Assembly of the centrifugal pump was accomplished in the following sequence: Before assembly was initiated, certain critical dimensions on the detail parts were measured and recorded, for the purpose of establishing nominal clearances and fits and dynamic seal compressions. Also, the impeller was installed temporarily on the shaft, and the runout of the front and rear wear-ring surfaces was recorded. To determine the impeller front and rear axial clearance, the inlet housing, impeller, and volute housing were assembled initially, without the shaft or bearings.

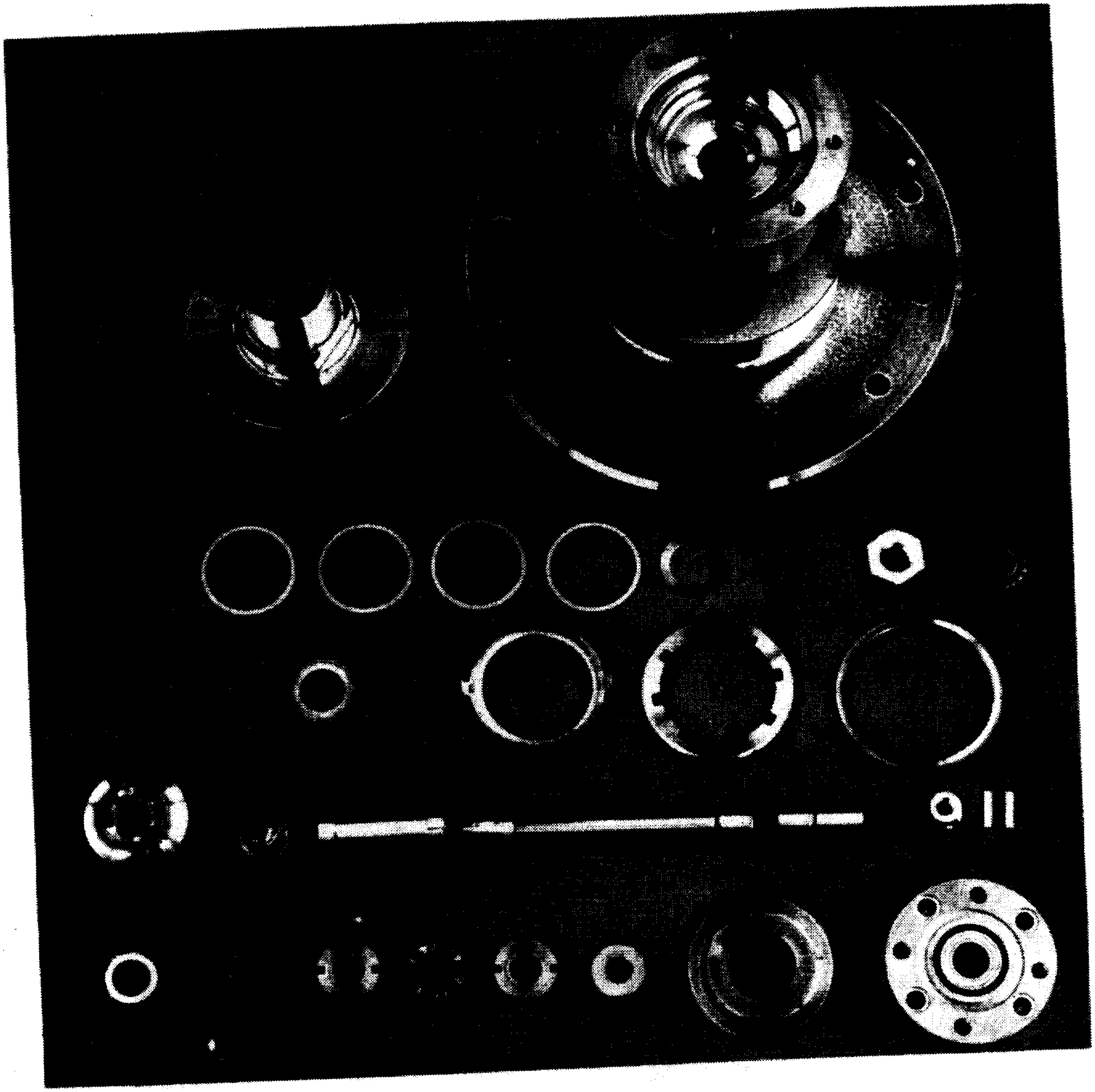

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Figure 1. Mark 36 Centrifugal Pump

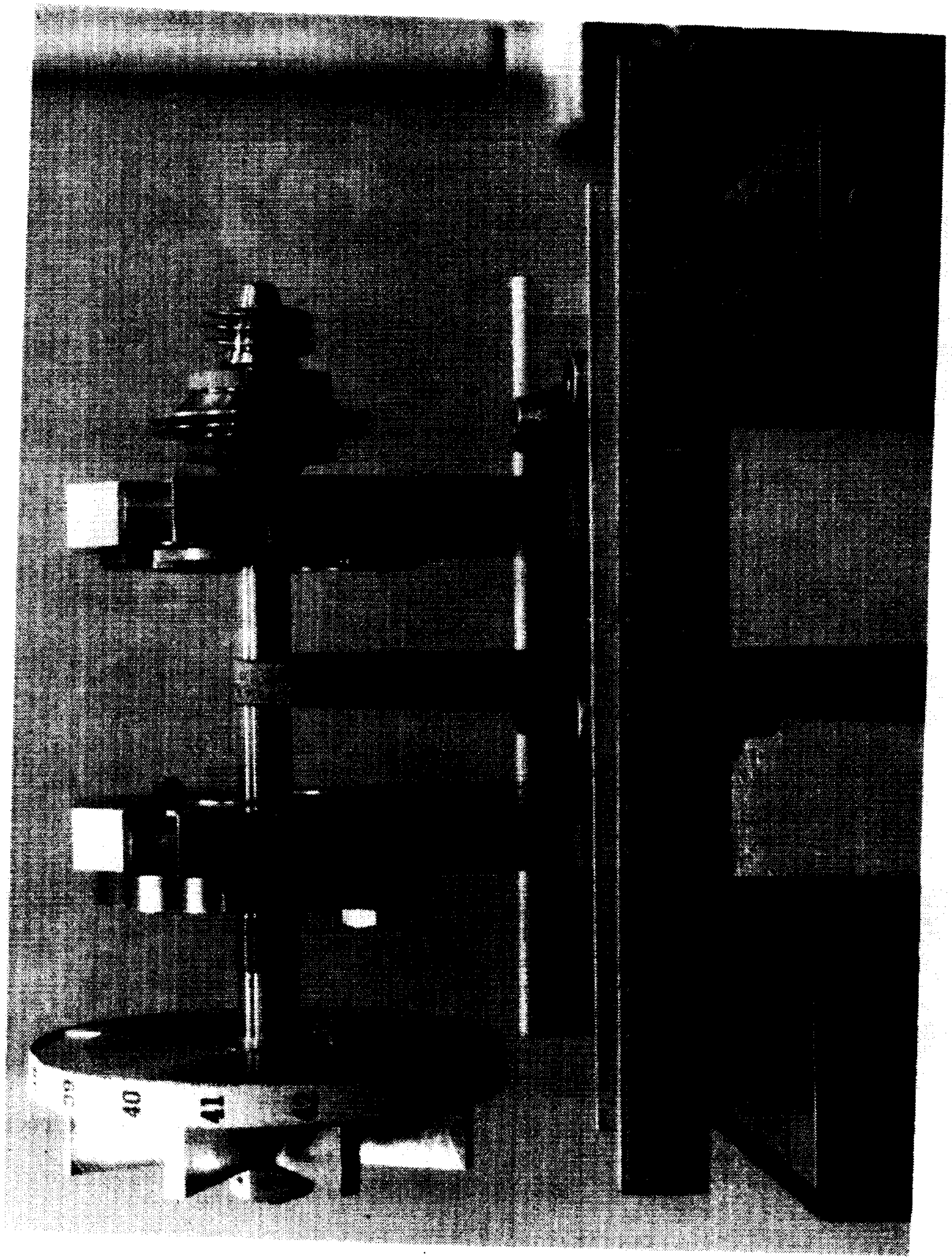

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Figure 2. Mark 36 Pump Balance Assembly

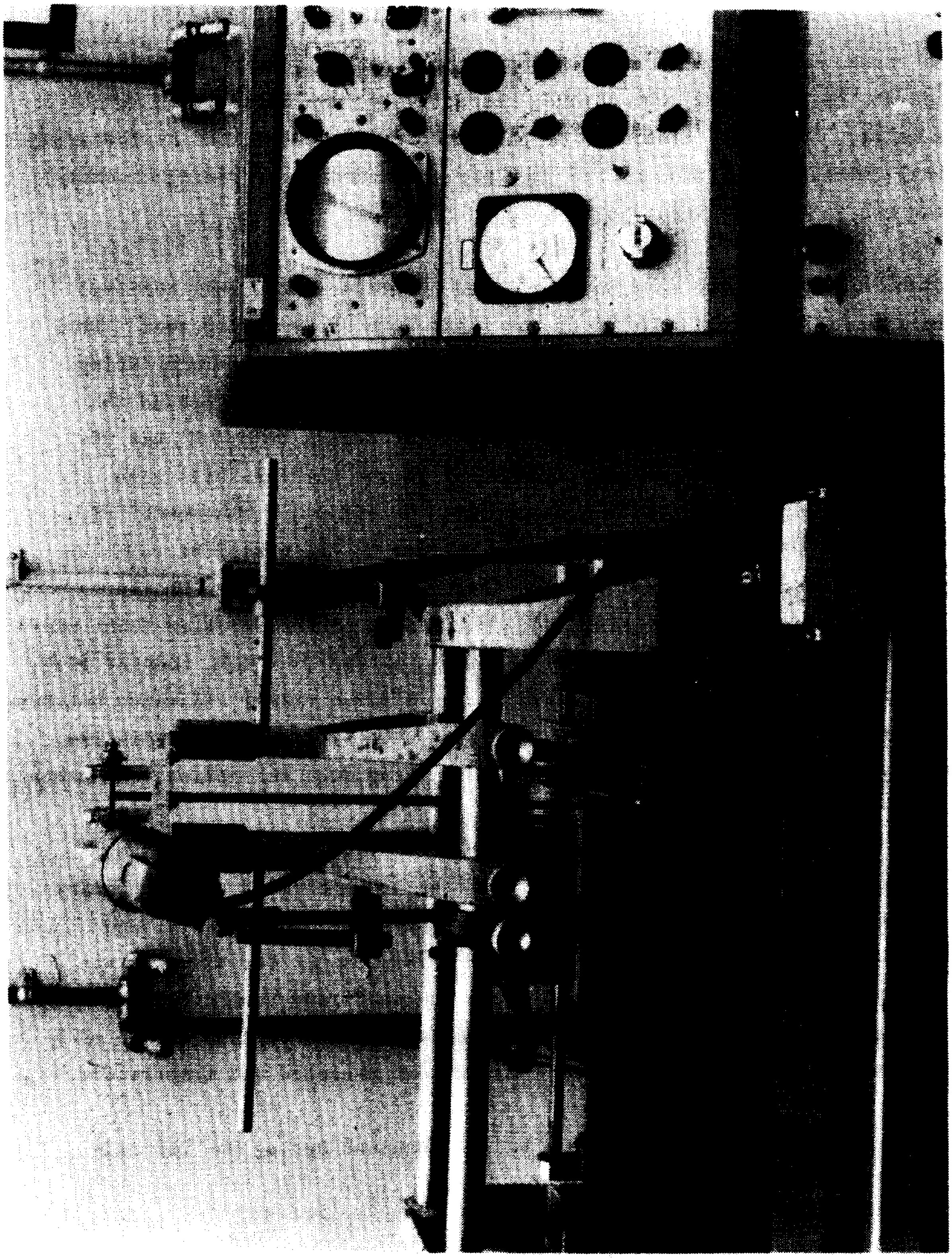

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Figure 3. Mark 36 Pump Balance Assembly

Then, a "push-pull" was performed in which the position of the impeller, when bottomed at the front and rear shroud, respectively, was established relative to the inlet. The axial clearance of the impeller was calculated by comparing the final installed position of the impeller (recorded later during assembly) with the two values obtained above.

The actual buildup of the pump was initiated by pressing the rear bearing, bearing spacer, and front bearing on the shaft with an hydraulic tool. This rotor subassembly was then installed with the front bearing preload spring in the volute housing (Fig. 4), and the thickness of the shim behind the spring was adjusted until a bearing preload of 60 pounds (266.5 N) was obtained. Subsequently, the primary, intermediate and turbine seals were installed, with appropriate flange seals (Fig. 5 and 6). The runout of the volute wear-ring inner diameter, with respect to the center of rotation of the shaft, was measured, after which the impeller was pressed on the shaft with an hydraulic tool (Fig. 7). The runout of the impeller front wear ring was verified, followed by installation of the impeller nut, inducer lock, inducer (Fig. 8), and the inlet housing. The minimum radial clearance between the inducer tip and inlet housing was measured with wire gages; the minimum wear-ring clearances were calculated by subtracting from the nominal clearance the measured run-outs.

Ambient leak checks were conducted in which the leakage rates of the primary seal, intermediate seal, and turbine seal were measured when pressurized with gaseous helium to 30 psig ($.2 \times 10^6$ N/m^2) (Fig. 9). The assembly was completed by installing the turbine disk on the shaft and fastening the turbine manifold on the pump housing. The breakaway and turning torque of the rotor was measured after assembly, at ambient pressure and temperature.

The significant clearance and leakage values obtained during the initial assembly of the pump are included in Table I.

The assembled pump minus the turbine manifold is shown in Fig. 10. The weight of the pump, excluding the turbine components and external fittings, was established at 10.4 pounds (46.2 N).

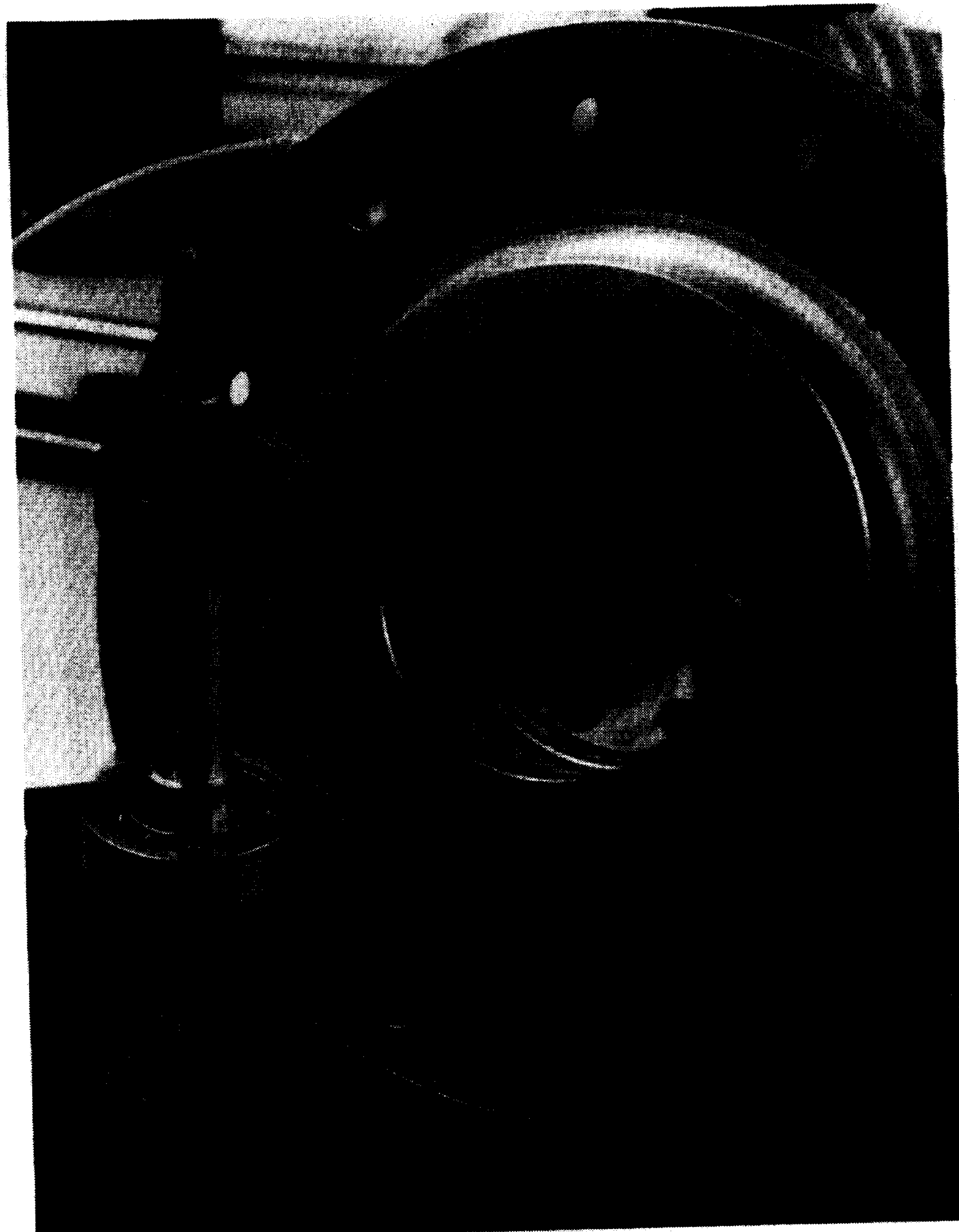

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Figure 4. Mark 36 Centrifugal Pump Installation of the Shaft-Bearing Subassembly

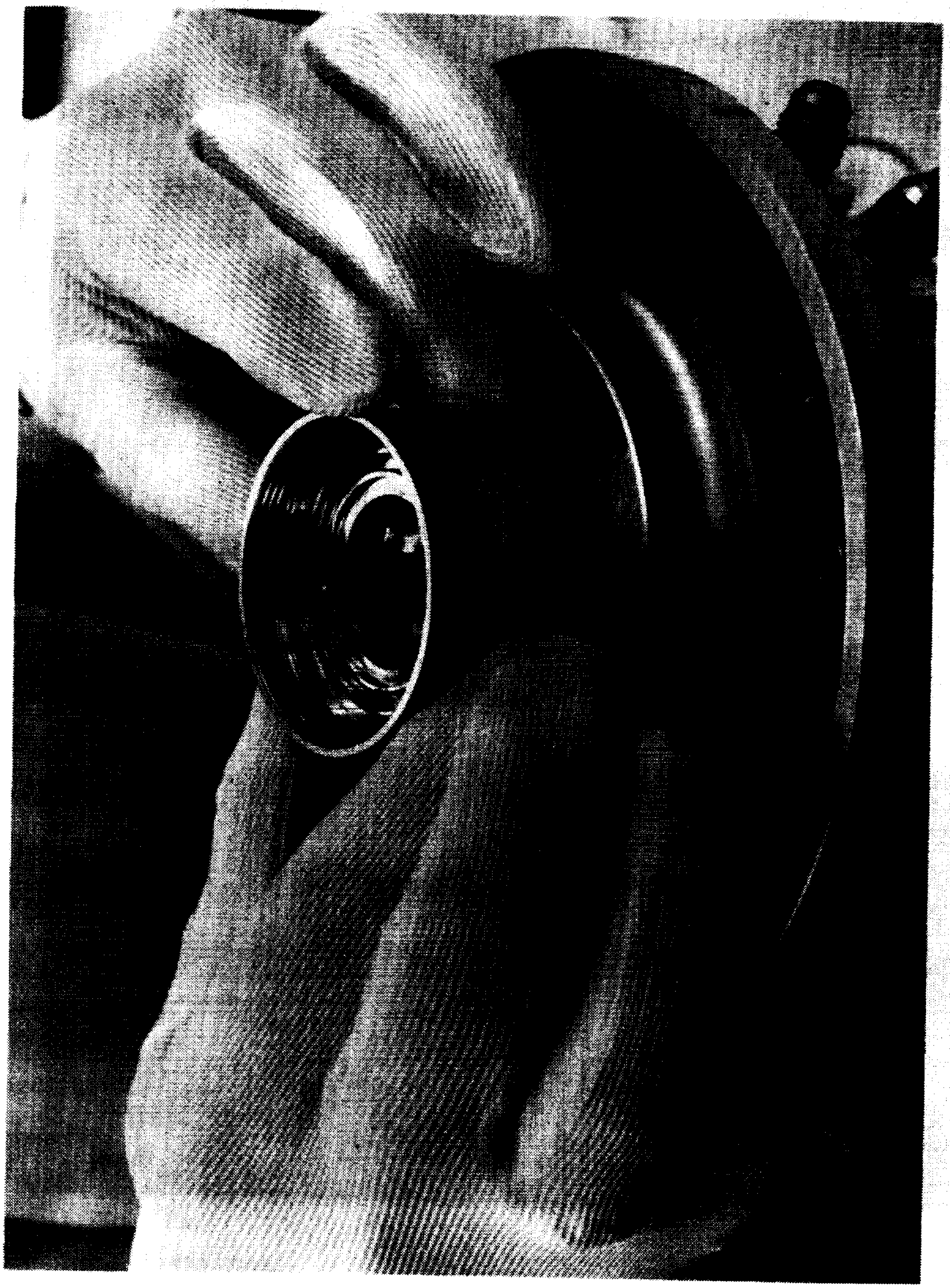

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Figure 5. Mark 36 Centrifugal Pump Primary Seal Installation

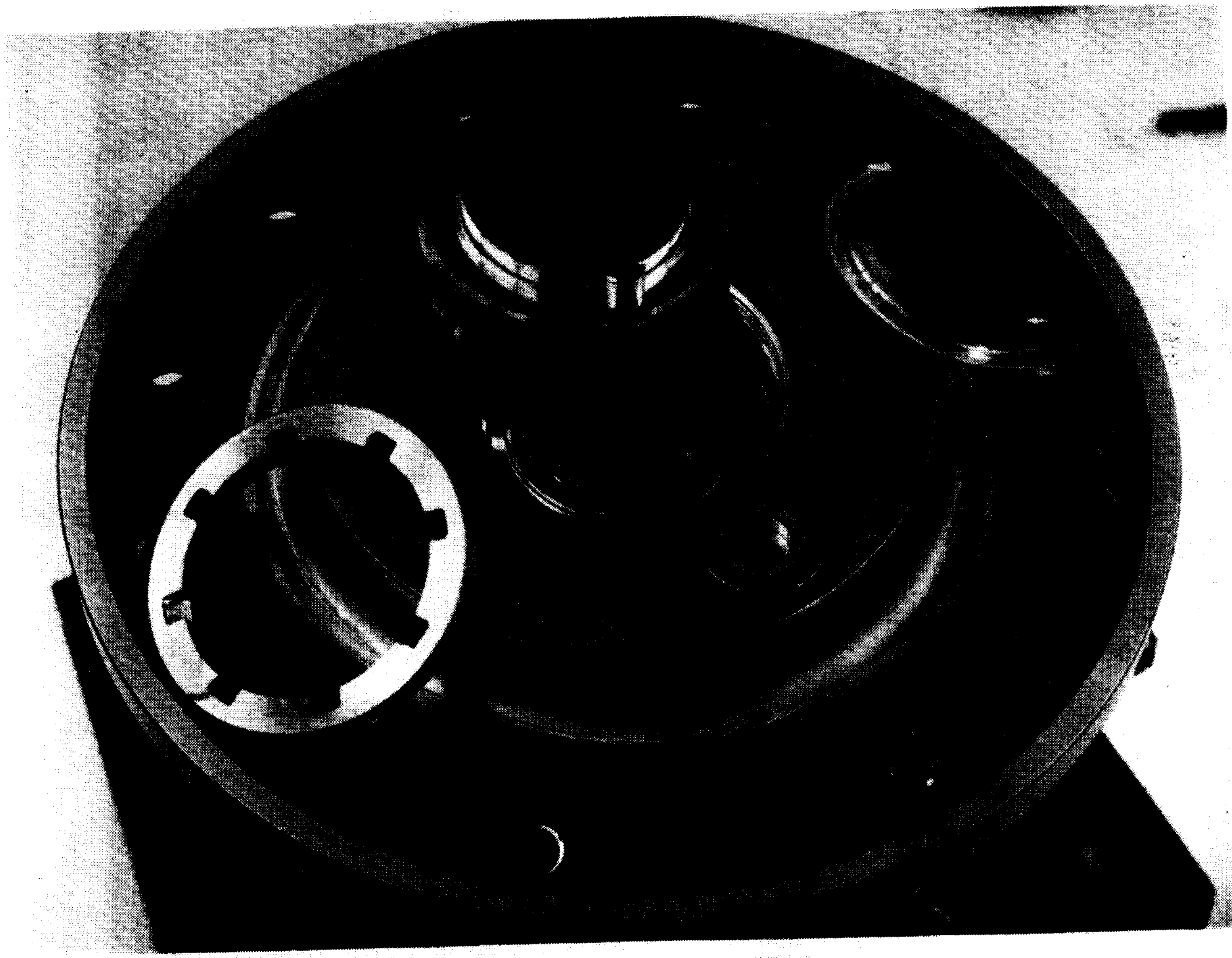

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Figure 6. Mark 36 Centrifugal Pump Intermediate Seal Rings Installed; Turbine Seal Ready for Installation

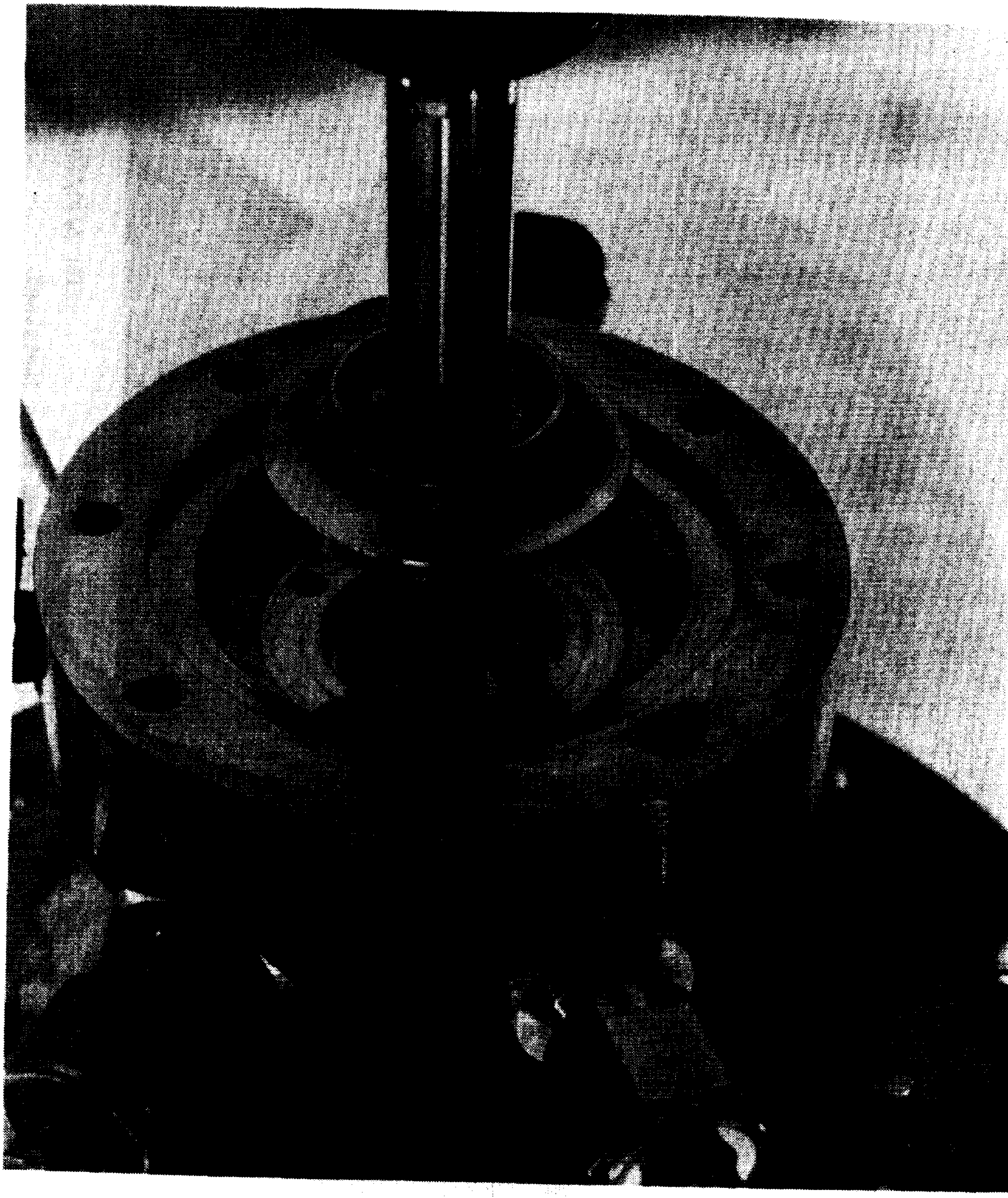

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Figure 7. Mark 36 Centrifugal Pump Impeller Installation

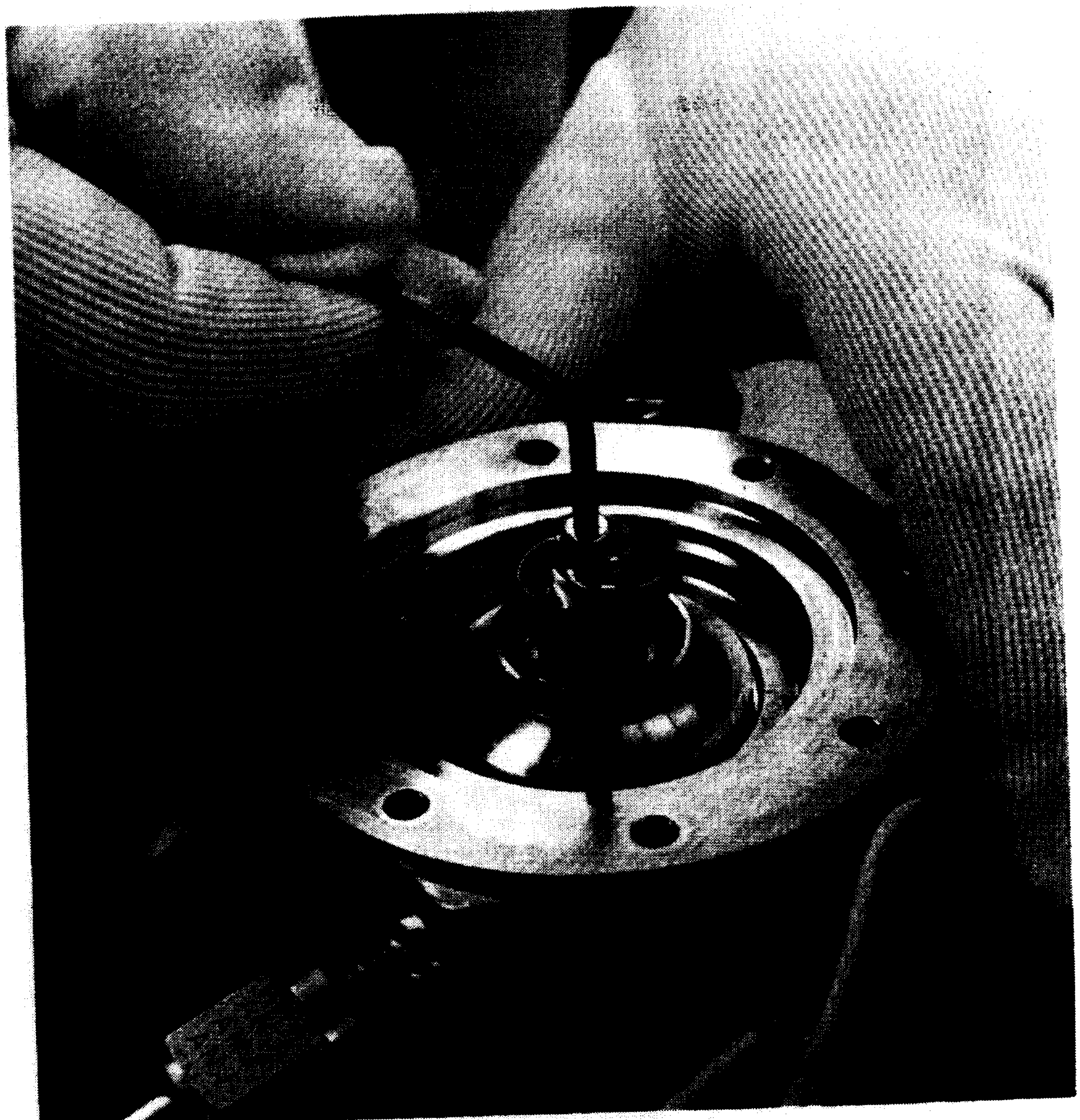

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Figure 8. Mark 36 Centrifugal Pump Inducer Installation

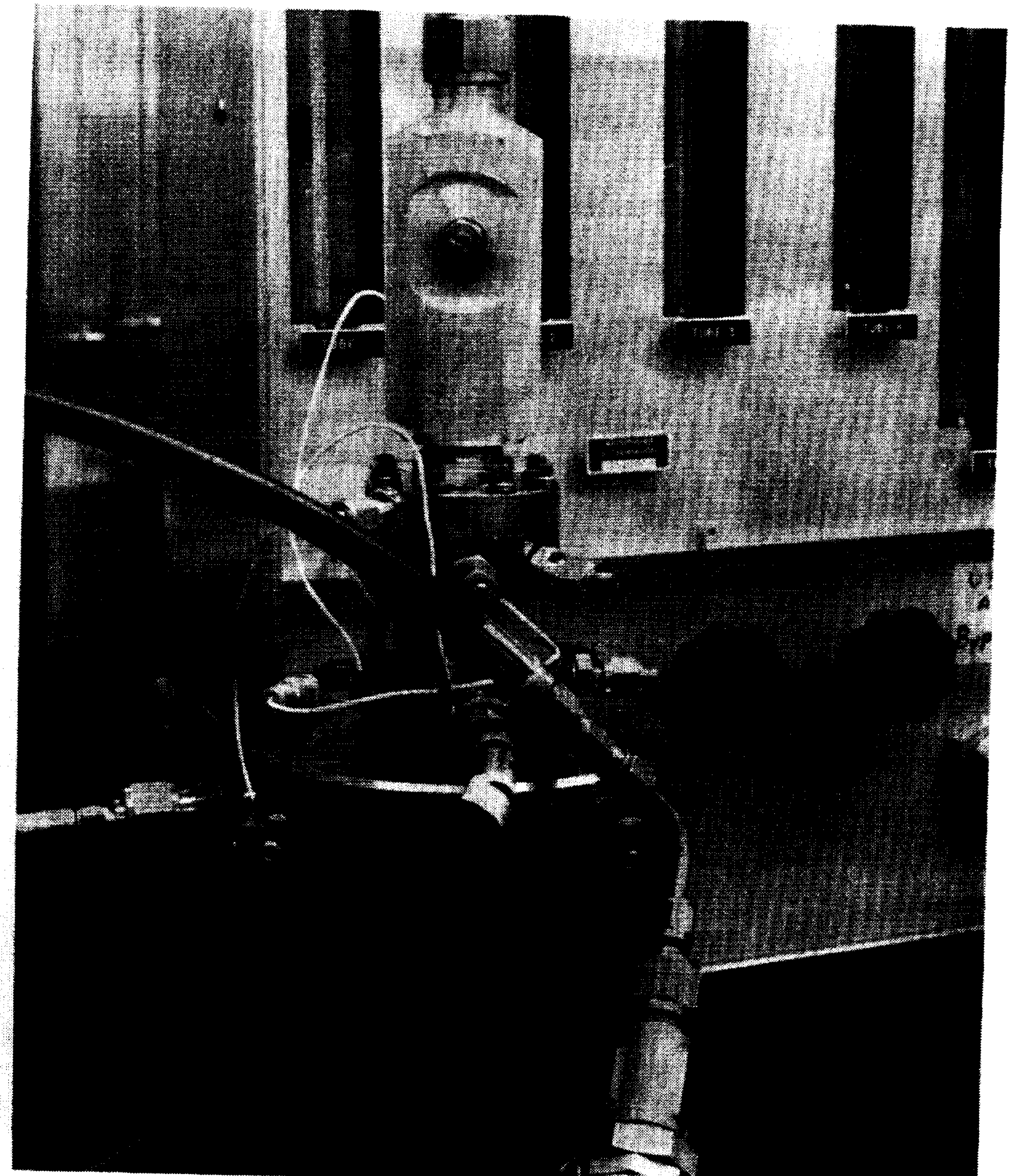

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Figure 9. Mark 36 Centrifugal Pump Seal Ambient Leak Check

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Figure 10. Mark 36 Centrifugal Pump With Turbine Drive

TABLE I

MK-36 CENTRIFUGAL PUMP

INITIAL ASSEMBLY MEASUREMENTS

INDUCER RADIAL CLEARANCE	NOMINAL	=	.0091 INCH	(.231 mm)
	MINIMUM	=	.009 INCH	(.229 mm)
FRONT WEAR-RING CLEARANCE	NOMINAL	=	.0055 INCH (RAD)	(.140 mm)
	MINIMUM	=	.0053 INCH (RAD)	(.135 mm)
REAR WEAR-RING CLEARANCE	NOMINAL	=	.0030 INCH (RAD)	(.076 mm)
	MINIMUM	=	.0028 INCH (RAD)	(.071 mm)
FRONT SHROUD CLEARANCE	MINIMUM	=	.029 INCH	(.789 mm)
REAR SHROUD CLEARANCE	MINIMUM	=	.017 INCH	(.432 mm)
PRIMARY SEAL LEAKAGE	@ 30 PSIG	=	5 SCIM	(1.36×10^{-6} m^3/SEC)
INTERM. SEAL LEAKAGE, PRIM. SEAL SIDE	@ 80 PSIG	=	7600 SCIM	(2.08×10^{-3} m^3/SEC)
INTERM. SEAL LEAKAGE, TURBINE SEAL SIDE	@ 80 PSIG	=	6700 SCIM	(1.83×10^{-3} m^3/SEC)
TURBINE SEAL LEAKAGE	@ 30 PSIG	=	0 SCIM	
SHAFT TORQUE		=	10 IN. OZ	(.071 Nm)

FREON TEST FACILITY

Facility Description

Testing of the centrifugal pump with Freon-12 was accomplished in Rotary Test Cell #2 of Rocketdyne's Engineering Development Laboratory at Canoga Park, California. A schematic of the facility configuration is presented in Fig. 11. The pump installation in the test cell is shown in Fig. 12, 13 and 14.

The pump fluid system was of a closed-loop type, in which Freon-12 was supplied to the pump from a 40-gallon (.15 m^3) capacity run tank. From the pump discharge, the freon was returned to the tank, after passing through a throttle valve, which was used to regulate the system resistance.

The temperature of the freon was maintained at the desired level by submerging the run tank in a tank of ethylene glycol-water mixture, which in turn was kept chilled by allowing a controlled amount of LN_2 to bubble through it.

An important consideration in establishing the facility configuration was the requirement to minimize the amount of noncondensible gases, such as GN_2 or air, in the Freon 12. This was desirable particularly for the suction performance tests, because noncondensible gases coming out of solution at the lower pressures could give an erroneous appearance of cavitation. To avoid this problem, instrumentation grade Freon 12 was procured, which has a maximum concentration of 50 ppm in the liquid phase. Before filling the run tank with freon from the shipping containes ("K"-bottles), the tank as well as the pump inlet and discharge lines were evacuated and, from that point on, the system was maintained free of foreign gases. Pressurization of the tank, to provide the desired pump inlet pressure, was effected by bleeding liquid freon from the bottom of the run tank and passing it through a heat exchanger and returning it into the tank ullage in the vapor phase.

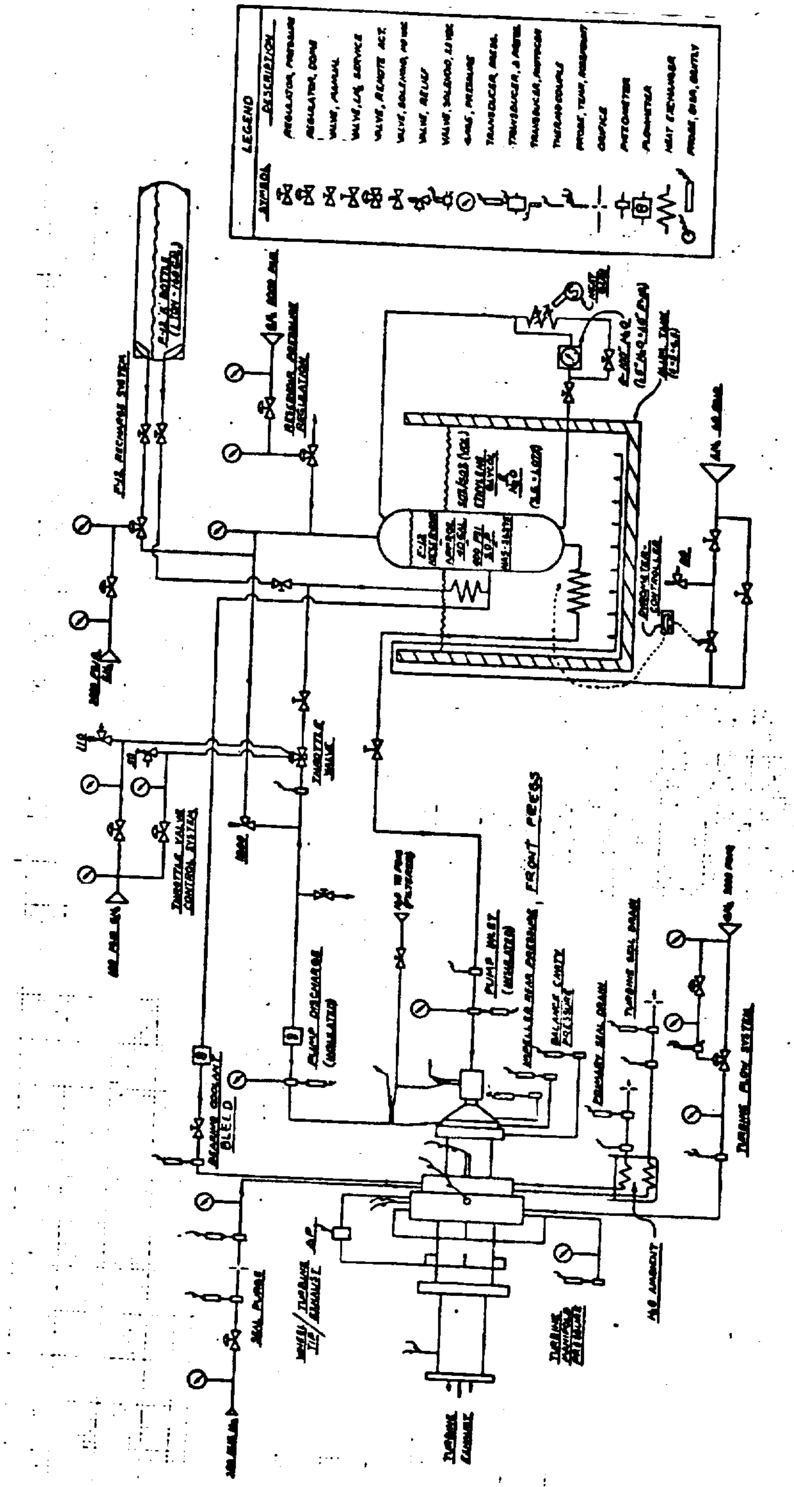

Figure 11. Schematic F-12 Flow System, MK-36 Pump

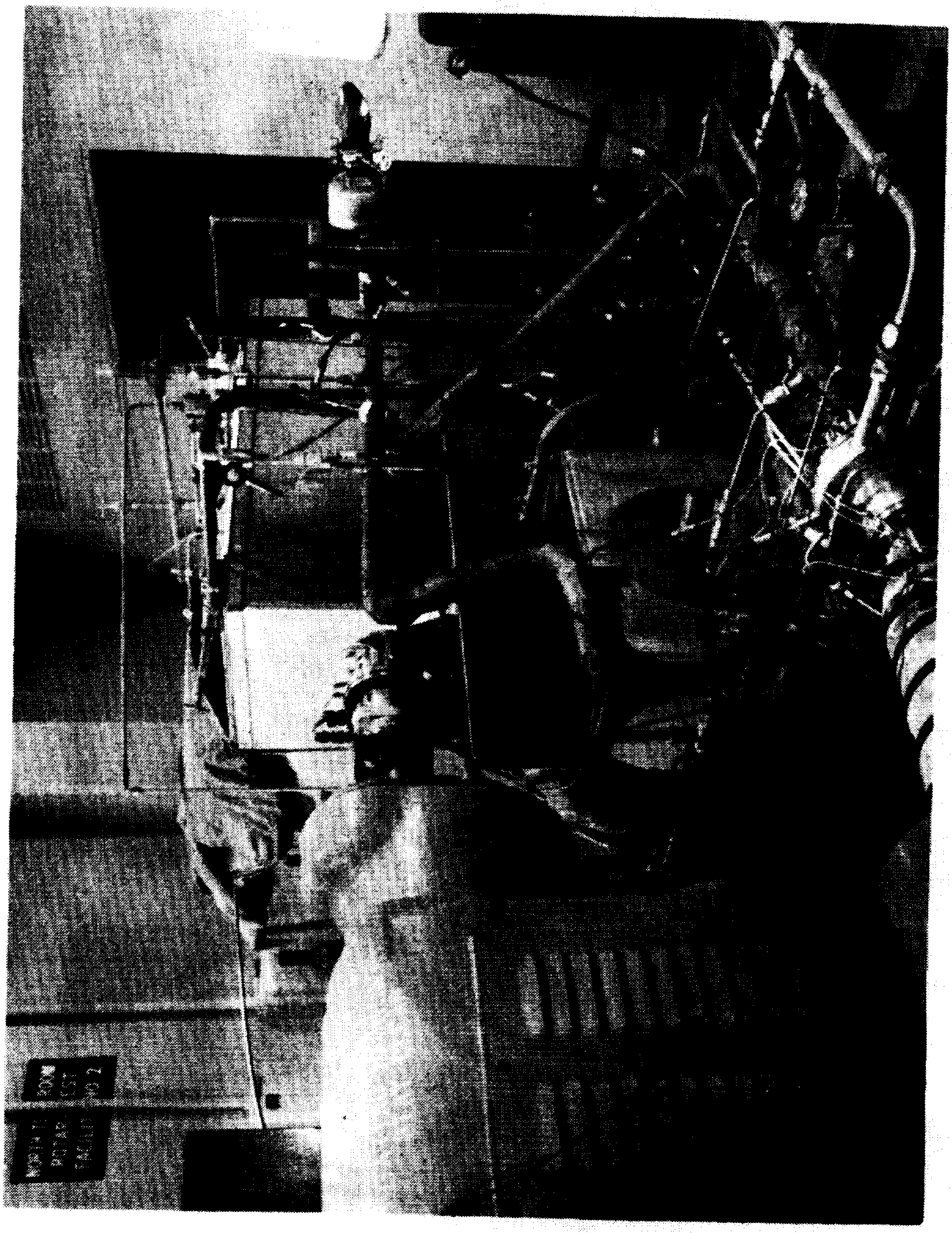

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Figure 12. Mark 36 Centrifugal Pump Freon 12 Test Facility

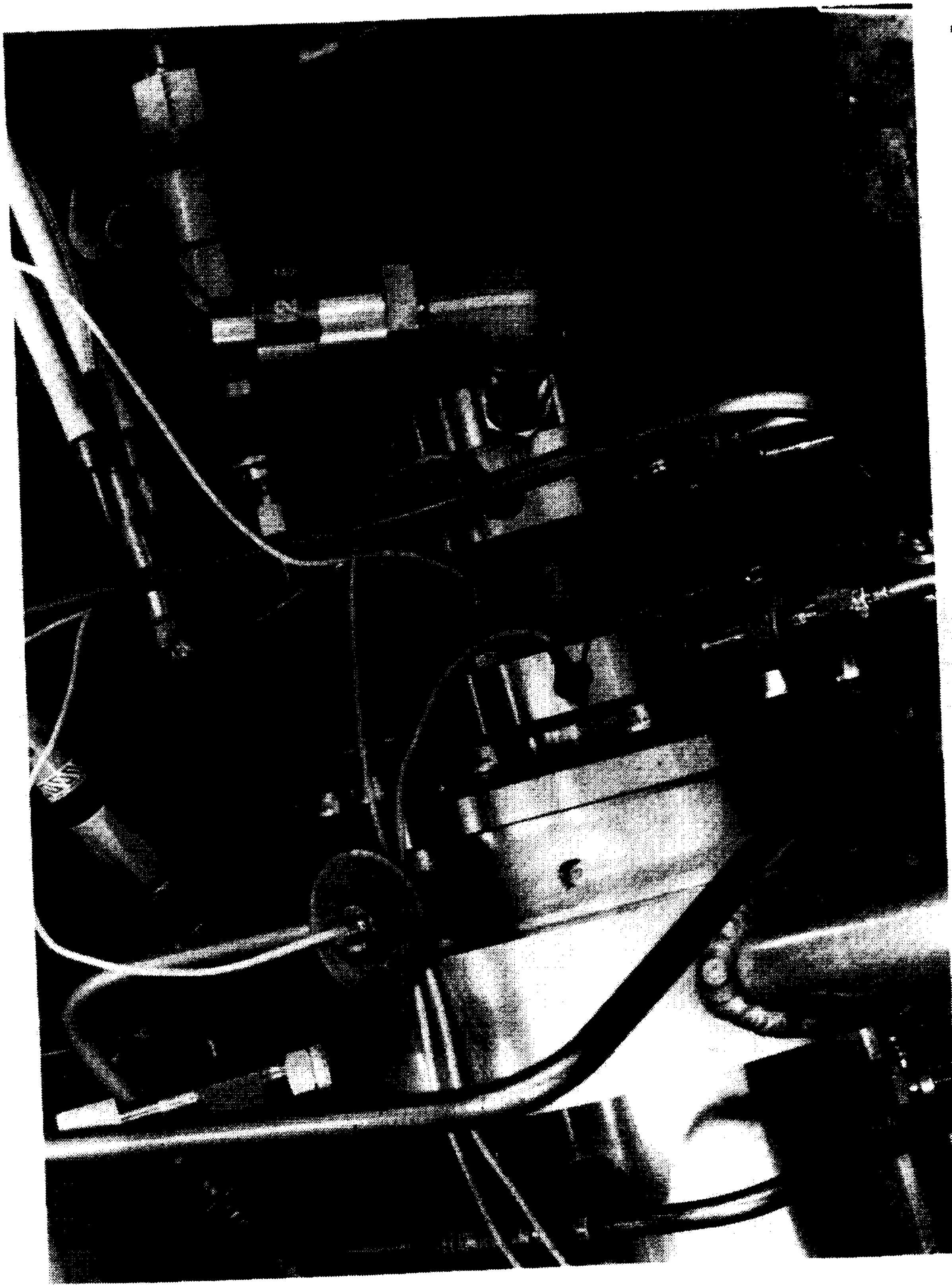

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Figure 13. Mark 36 Centrifugal Pump Installed for Testing With Freon 12

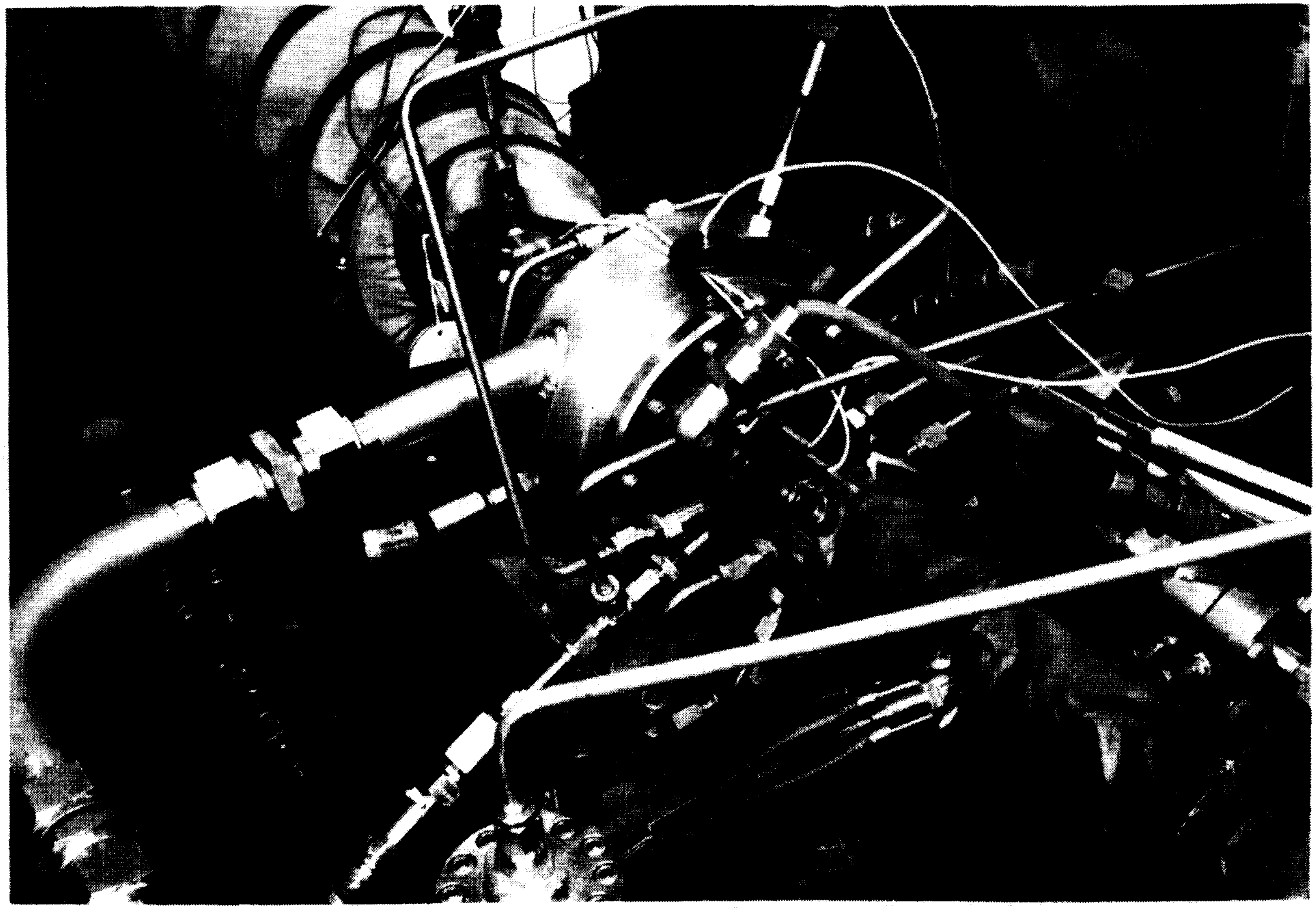

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Figure 14. Mark 36 Centrifugal Pump Installed for Testing With Freon 12

The pump inlet and discharge lines were insulated to reduce heat absorption and to provide a better quality liquid to the pump inlet.

Instrumentation

Figure 15 presents a schematic of the parameters which were measured and recorded on pump tests with freon. Pump inlet temperature and pressure, discharge pressure, speed, pump flow, and turbine manifold pressure were recorded on a Brush recorder, which facilitated monitoring these parameters while the test was in progress. The other parameters were recorded on an oscillograph, with the exception of high-frequency pressure measurements, speed, accelerometer outputs, and shaft radial motion indications which were recorded on tape.

Statham and Norwood transducers were used to measure steady-state pressure levels; the dynamic pressures at the pump inlet and discharge were recorded with photocon transducers. Turbine fluid temperatures were measured with Rosemount transducers, whereas freon temperatures were recorded with exposed tip iron-constantan thermocouples. In addition to the individual temperature probes in the inlet and discharge lines of the pump, the temperature differential between those two locations was directly recorded.

The seal leakages from the primary and turbine seal drain lines were passed through heat exchangers to ensure that the fluid was in the gaseous state; flowrates were then measured with orifices downstream of the heat exchangers.

Pump delivered flowrate was measured with a turbine-type flowmeter located in the discharge return line.

The radial motion of the shaft in the plane of the turbine disk was measured with a Bently inductive-type noncontacting displacement measuring device (Bently Nevada Corp., Minden, Nevada). In addition to recording the Bently output on tape, it was also displayed on an oscilloscope and continuously monitored during the tests. A Bently transducer was also used to obtain a shaft speed indication. This was accomplished by

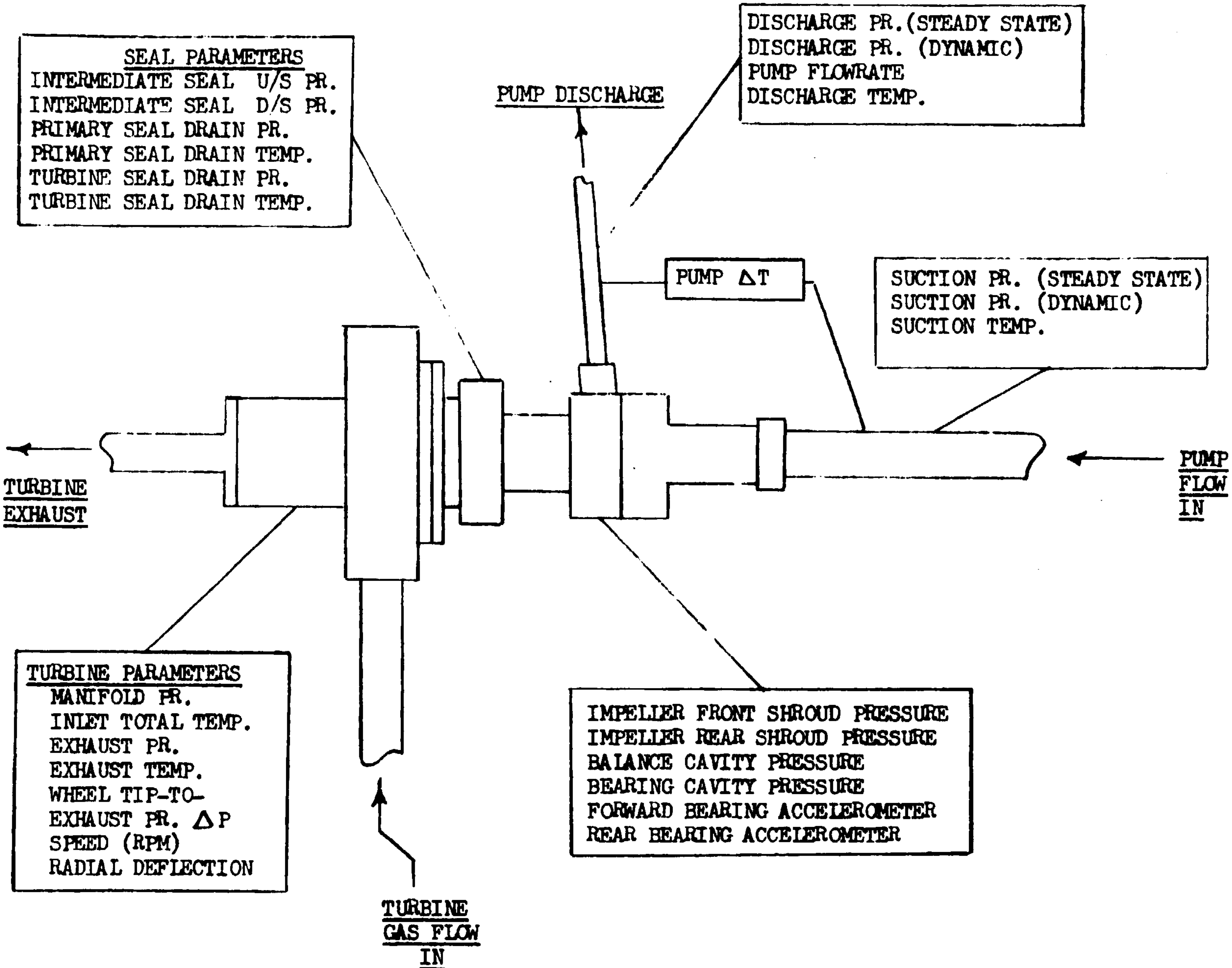

Figure 15. Mark 36 (12 gpm) Centrifugal Pump Instrumentation

machining a spotface in the turbine disk opposite the Bently transducer, which then produced a countable electronic signal every revolution.

Drive Turbine Calibration

To provide a means of determining the pump overall efficiency with the integral turbine drive system, a calibration was conducted in which the turbine power output was established with a dynamometer.

Figure 16 shows the mechanical configuration used for calibration. The volute, bearings, shaft, and turbine components of the standard pump assembly were used. A spline coupling, which connected the shaft to the dynamometer, was mounted in place of the inducer and impeller. The shaft seal package was replaced by a labyrinth seal to minimize the torque absorbed between the turbine and the dynamometer. With this configuration, the only power dissipation between the turbine and the dynamometer was due to the bearings, which is considered negligible. Bearing cooling was accomplished by introducing a small quantity of Freon 21 between the labyrinth seal and the rear bearing and allowing it to flow through both bearings, after which it evaporated into the atmosphere.

The dynamometer used for calibration was a Vortec Model A-20 air dynamometer in which power is absorbed through a rotor, which imparts angular momentum to air drawn in from the atmosphere. This angular momentum is then absorbed by a cradled stator, the reaction torque of which is measured and recorded. The power absorption rate can be adjusted by varying the flow area through the rotor.

Calibration was performed in Rotary Test Cell #2 of Rocketdyne's Engineering Development Laboratory at Canoga Park, California. The installation is shown in Fig. 17.

Seven dynamic tests and two stall torque tests were conducted in the speed range 0 to 71,700 rpm and turbine pressure ratio 2.3 to 5.0. The torque range was limited to a maximum of 11 in. lb (1.24 Nm) and the power level to a maximum of 10 hp (7457 watts) by the dynamometer.

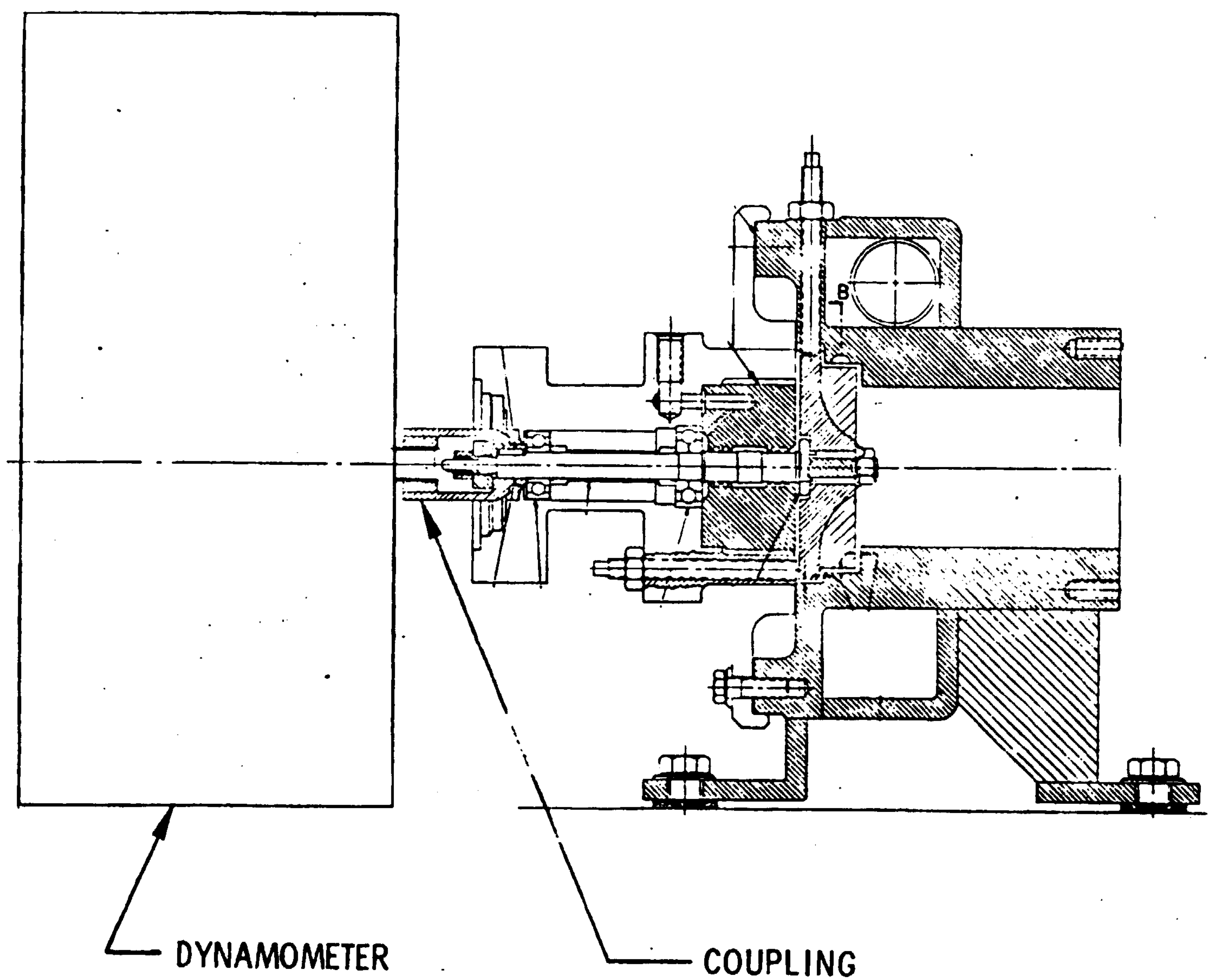

Figure 16. Mark 36 Gas Turbine Calibration Fixture

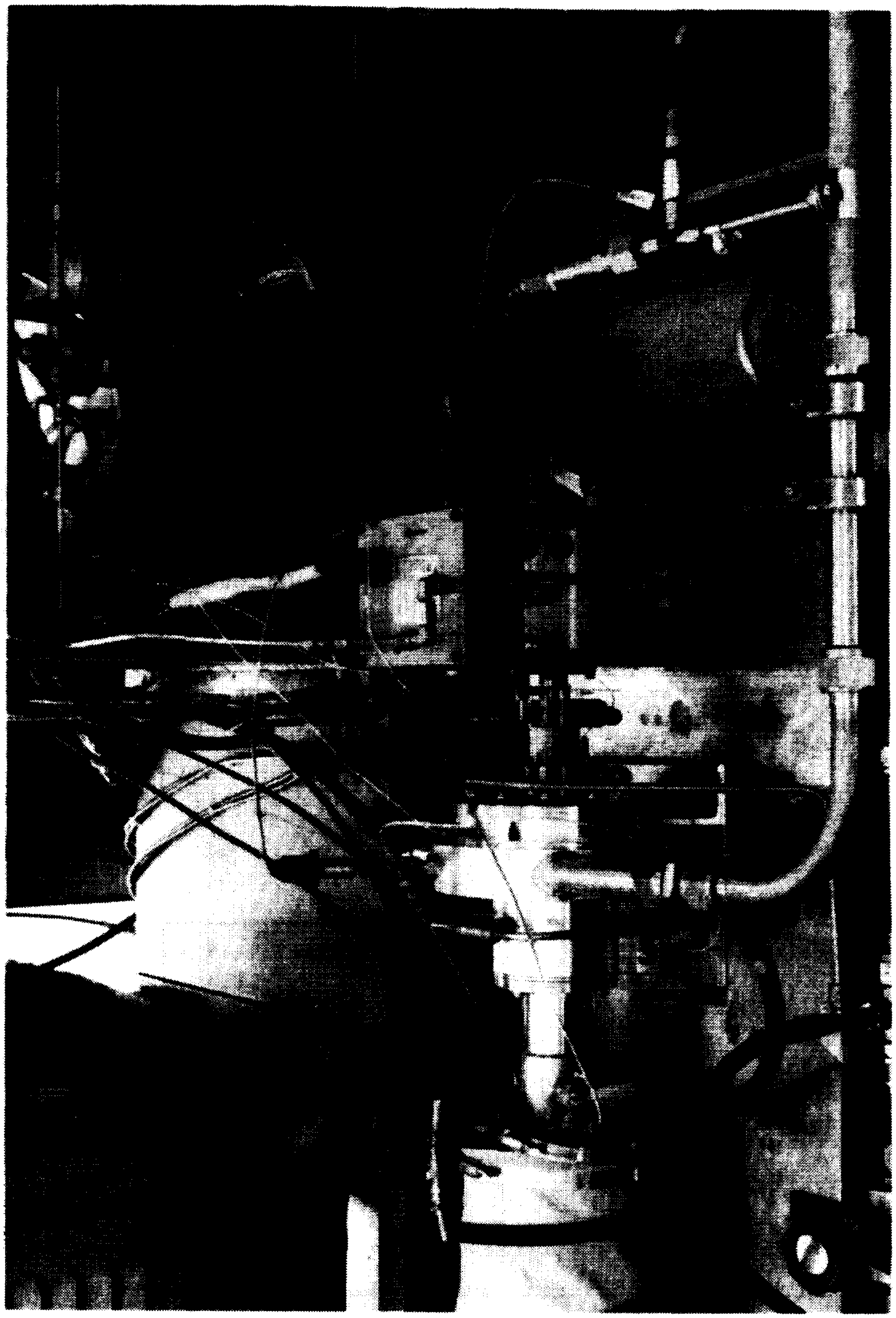

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Figure 17. Mark 36 Turbine Calibration Installation

Figure 18 presents the turbine performance map which was developed from the calibration data. A conformance check of the map with the valid dynamometer test points indicated an average difference of 1 percent, with a 2 σ deviation of 6 percent. A dynamometer loading point was repeated at constant turbine inlet pressure and the power measured was within 3 percent, indicating repeatability.

On pump tests, turbine inlet pressure, pressure ratio, speed, and turbine inlet temperature are measured, and torque applied to the pump shaft is calculated by using the performance map of Fig. 18.

The turbine calibration revealed that the pump shaft first critical speed was at approximately 28,800 rpm, compared to the predicted value of 25,650 rpm. Turbine disk radial motion was monitored with a Bently transducer. It was found that the shaft was experiencing subsynchronous whirl at speeds above 45,000 rpm. Radial motion amplitude of 0.006-inch (1.52×10^{-4} m) peak to peak were observed above that speed range typically, compared to a maximum of 0.011 inch (2.8×10^{-4} m) noted when passing through the first critical speed. One test was terminated when the turbine radial motion reached approximately 0.015-inch (3.8×10^{-4} m) peak to peak at 68,000 rpm and appeared to be increasing. Subsynchronous whirl can originate from fluid film effects or rotor internal friction effects. Since the rotor used during turbine calibration was of interim configuration, which differed from the operational rotor (inducer and impeller were replaced by coupling), no effort was expended to pinpoint the source of the whirl. All test objectives were achieved in that the performance of the turbine was defined over the desired range.

Disassembly after calibration revealed that all hardware was in excellent condition. The bearings, in particular, which were lubricated with freon, showed no sign of wear or distress.

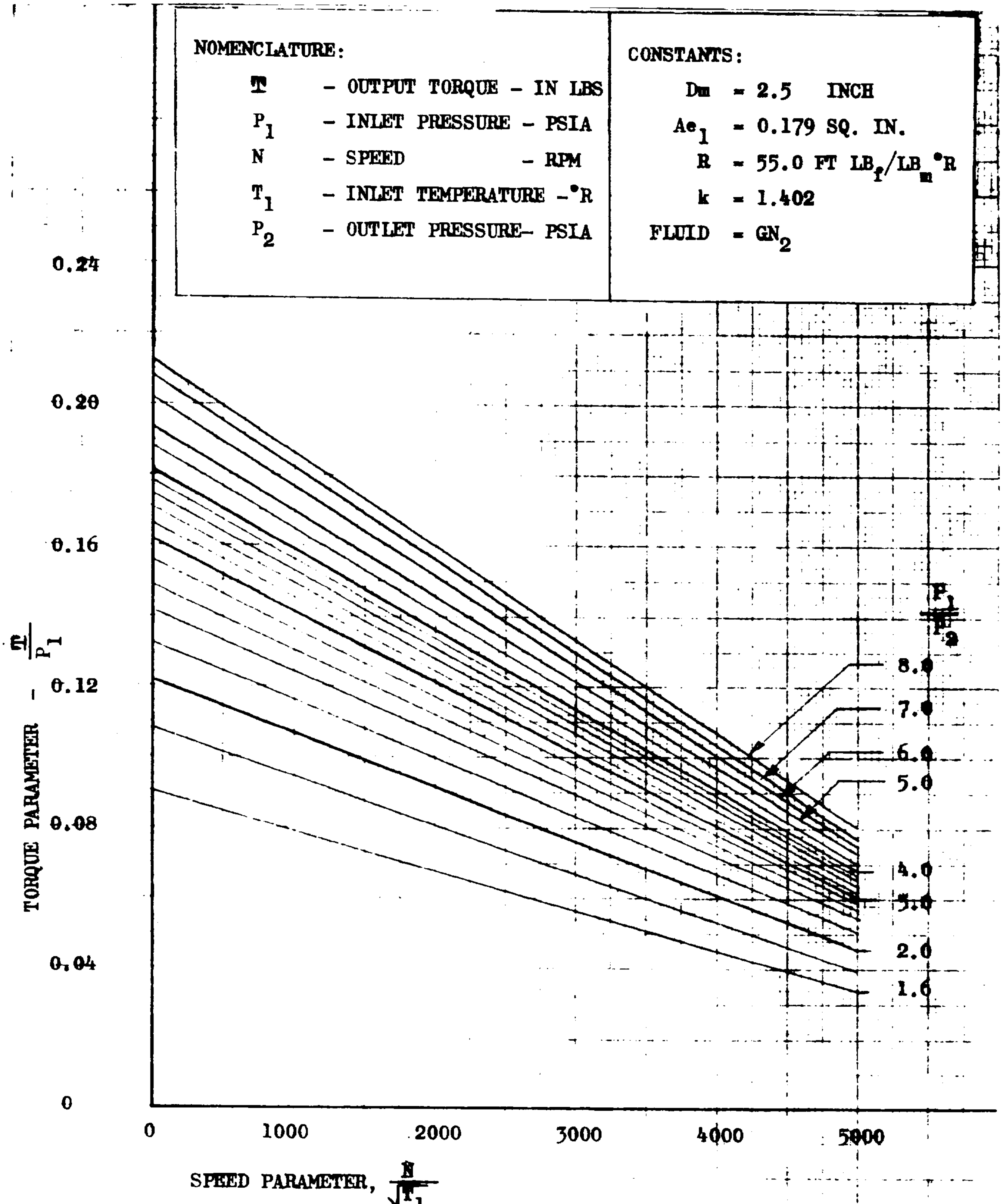

Figure 18. MK 36 Turbine Performance Map (Based on Analysis of Turbine Calibration Tests With Vortec - A-20 Dynamometer)

PUMP TESTS WITH FREON

Initial testing of the centrifugal pump was performed using Freon-12 as the pumped fluid, with the objective of determining the hydrodynamic performance and evaluating the mechanical operation of the pump. A total of 30 tests were conducted, at speeds ranging up to 80,000 rpm, accumulating a total of 2.94 hours of operation in the process. All contract objectives were achieved.

Freon Performance Tests

The objective of the freon H-Q tests was to define the hydrodynamic performance and mechanical operating capability of the pump, using Freon 12 as the pumped fluid. Eleven tests were conducted in the H-Q series, accumulating a total time of 1 hour and 12 minutes.

Table II presents the pump configurations which were used during the test series. The pump was completely disassembled after tests #004, #006, and #011; a partial disassembly to change the impeller nut was performed after test #010.

The raw pump and turbine data and seal leakage data obtained during the test series is presented in Tables III and IV, respectively. A summary of the reduced data at actual test speeds and scaled to the design speed of 75,000 rpm is included in Table V.

Tests No. 01 through 04. The purpose of the initial series of tests was to observe the mechanical operating capability of the pump and obtain H-Q data at speeds ranging from 10 percent to 80 percent of nominal speed. The actual speeds covered ranged from 0 to a maximum of 62,400 rpm. On each test, the flowrate was regulated from essentially shutoff to 120 percent of the nominal Q/N by varying the resistance of the valve located in the discharge line.

On tests #001 and #002, satisfactory operation in the speed range 0 to 20,000 rpm was achieved and low-speed H/Q data was obtained. Test #003

TABLE II. SUMMARY OF MARK 36 CENTRIFUGAL PUMP CONFIGURATIONS DURING FREON 12 H-Q TESTING IN ENGINEERING LABORATORY

RUN NO.	PUMP CONFIG. NUMBER	INDUCER DIAMETRAL CLEARANCE (IN)	FRONT WEAR RING DIAMETRAL CLEARANCE (IN)	REAR WEAR RING DIAMETRAL CLEARANCE (IN)	IMPELLER NUT SHOULDER RELIEVED	BEARING CAVITY PR. TAP & BLEED LINE INSTALLED	BEARING CAVITY BLEED LINE OPEN OR CLOSED	PRIMARY SEAL NOSE MATERIAL	TURBINE SEAL WITH OR WITHOUT DAMPENER	BEARING SPACER ID RELIEVED	REMARKS
01 THRU 04	I	0.0182	0.011	0.006	NO	NO	N/A	Al_2O_3	W/O	NO	TURBINE SEAL BELLOWS FAILED. PRIMARY SEAL LEAKING HEAVILY.
05 & 06	II	0.0182	0.011	0.006	YES	NO	N/A	CARBON	W/O	YES	USED FAILED TURBINE SEAL HOUSING (W/O BELLOWS) FROM TESTS 01 - 04.
07 1ST HALF	III	0.0182	0.011	0.0034	YES	YES	CLOSED	CARBON	W/O	YES	USED NEW TURBINE SEAL. INSTALLED NEW PRIMARY SEAL TO REPLACE ONE THAT FAILED BELLOWS DURING TEST 06.
07 2ND HALF	IV	0.0182	0.011	0.0034	YES	YES	OPEN	CARBON	W/O	YES	
08 THRU 10	III	0.0182	0.011	0.0034	YES	YES	CLOSED	CARBON	W/O	YES	
11	V	0.0182	0.011	0.0034	NO	YES	CLOSED	CARBON	W/O	YES	

TABLE III. SUMMARY OF RAW DATA OBTAINED DURING FREON 12 H-Q TESTING IN ENGINEERING LABORATORY MARK 36 CENTRIFUGAL PUMP

RUN NO	DATE	TIME SLICE (SEC)	PUMP SPEED (RPM)	PUMP FLOW (GPM)	PUMP INLET PR. (PSIG)	PUMP DISCH. PR. (PSIG)	PUMP INLET TEMP (F)	MEAS. PUMP ΔT (F)	IMPELLER FRONT SHROUD PR. (PSIG)	IMPELLER REAR SHROUD PR. (PSIG)	BALANCE CAVITY PR. (PSIG)	PUMP CONFIG. NUMBER (SEE TABLE II)
01	5/25/70	786	20,000	4.4	32	110	-71	3.5	20*	0*	50	I
		837	20,000	3.2	33	120	-68.4	4.1				
02	5/26/70	56	18,000	4.9	21	80	-60	10.8	25*	20*	25	I
		116	20,200	2.9	24	130	-59	8.0	50*	10*	65	
03	5/26/70	141	44,400	5.9	19	500*	-55	9.6	120*	DNR	230	I
		231	43,100	6.4	20	470*	-53	8.7	110*	DNR	220	
		329	42,000	7.9	19.5	410*	-51	7.4	90*	DNR	190	
		380	42,000	8.6	19.5	340*	-50	7.3	80*	DNR	150	
		462	42,300	7.0	19.5	440*	-50	8.0	100*	DNR	210	
		554	44,000	5.7	17	500*	-50	9.5	150*	DNR	235	
		603	47,000	2.9	18.5	590*	-50	16.0	200*	DNR	280	
		659	48,500	1.4	17	630*	-51	24.5	210*	DNR	310	
		803	44,000	1.0	19	520*	-52	27.2	180*	290	270	
		848	45,500	2.9	20	560*	-51	17.0	180*	290	275	
		891	45,500	4.2	21	550*	-49	13.0	150*	240	265	
		964	43,500	6.3	19	470*	-46	11.0	120*	220	230	
04	5/26/70	96	44,100	7.2	15.3	470*	-57	10.4	200	280	240	I
		156	60,500	9.1	8.8	750*	-54	14.5	250	375	320	
		226	60,500	8.2	8.0	830*	-50	15.2	350	490	380	
		258	61,000	7.5	8.0	900*	-49.7	16.1	410	540	423	
		325	62,000	5.5	9.8	960*	-48	20.1	500	580	455	
		383	61,500	3.7	10.3	960*	-48.5	25.8	520	610	470	
		443	62,400	1.6	12.6	1010*	-50	0/R	590	680	500	
		480	61,000	3.8	7.0	960*	-48	27.8	510	600	460	
		527	60,200	5.6	0.5	840*	-45.5	19.9	390	480	380	
05	6/4/70	135	61,500	8.9	33.5	1150	-38	17.5	DNR	575	475	II
		280	60,000	7.3	24	1100	-35	18.0	DNR	550	450	
		619	77,500	9.8	22.5	1800	-22	28.0	DNR	900	700	
06	6/4/70	DATA IN PARTIAL CAVITATION DUE TO INSUFFICIENT INLET PRESSURE										II

DNR - DID NOT RECORD
0/R - OVER CALIBRATION RANGE
* - MEASUREMENT APPARENTLY ERRONEOUS

TABLE III. (Concluded)

RUN NO	DATE	TIME SLICE (SEC)	PUMP SPEED (RPM)	PUMP FLOW (GPM)	PUMP INLET PR. (PSIG)	PUMP DISCH. PR. (PSIG)	PUMP INLET TEMP (F)	MEAS. PUMP ΔT (F)	BEARING CAVITY PR. (PSIG)	IMPELLER FRONT SHROUD PR. (PSIG)	IMPELLER REAR SHROUD PR. (PSIG)	BALANCE CAVITY PR. (PSIG)	PUMP CONFIG. NUMBER (SEE TABLE II)
07	6/16/70	114	46,000	8.2	47	660	-56	10.2	107	340	390	225	III
		145	46,500	7.2	50	720	-54	10.3	110	360	410	235	
		166	48,000	6.0	51	780	-54	11.8	117	410	440	255	
		214	48,500	2.6	46	850	-53	23.4	127	490	500	280	
		188	50,500	4.3	49	880	-53	16.8	127	480	500	280	
		232	48,500	2.6	46	830	-53	17.0	105	480	480	260	IV
		252	46,500	3.9	48	770	-52	12.2	100	420	430	230	
		272	47,500	5.8	47	770	-52	9.9	100	400	430	230	
		290	46,000	7.2	46	690	-51	7.9	90	350	400	215	
		308	48,000	7.8	46	740	-50	7.8	90	360	410	225	
		324	47,500	8.2	47	720	-49	7.5	90	360	400	220	
08	6/17/70	53	62,000	11.5	46	1070	-42	15.5	160	550	650	360	III
		70	63,000	10.4	50.5	1170	-40	16.2	175	600	680	385	
		85	64,000	9.4	50	1250	-40	18.0	180	650	700	400	
		106	60,500	8.0	48	1150	-38	17.0	170	610	640	365	
		129	63,000	5.4	47	1280	-37	24.5	205	760	725	415	
		157	63,000	3.1	46	1310	-38	O/R	230	850	760	435	
		178	65,000	1.6	47	1360	-38	O/R	245	920	800	440	
09	6/18/70	70	73,000	14.1	46	1180	-38	19.5	180				III
		100	73,500	12.0	54	1620	-36	21.0	235				
		125.5	73,000	10.0	62	1710	-35.5	23.5	245				
		154	73,500	7.2	67	1770	-36	30.5	280				
		182	74,500	4.1	51	1800	-38.5	O/R	305				
		200	73,500	2.1	51	1750	-40	O/R	305				
10	6/19/70	65	79,500	14.2	49	1770	-39	23	245				III
		73.5	80,500	12.3	57	1980	-37	25	280				
11	6/19/70	34	46,500	8.1	60	650	-48	15*	135	375	400	260	V
		62	45,000	7.2	58	640	-46	14*	130	370	375	250	
		82.5	47,000	5.9	58	740	-45	14.5*	140	420	415	265	
		106.5	49,500	4.1	56.5	860	-45	20*	165	540	491	300	
		130.5	44,000	2.7	56	710	-45	24*	150	468	415	265	
		145.5	45,500	1.5	55	760	-45	27.5*	160	514	455	273	

* MEASUREMENT APPARENTLY ERRONEOUS

TABLE IV. SUMMARY OF RAW DRIVE TURBINE DATA AND SEAL LEAKAGE RATES DURING FREON 12 H-Q TESTING IN ENGINEERING LABORATORY WITH MARK 36 CENTRIFUGAL PUMP

DRIVE TURBINE RAW DATA AT TEST SPEED						DYNAMIC SEAL LEAKAGE DATA										
						PRIMARY SEAL			INTERMEDIATE SEAL				TURBINE SEAL			
RUN NO.	TIME SLICE	TURBINE MANIFOLD PR. (PSIG)	TURBINE EXHAUST PR. (PSIG)	TURBINE INLET TOTAL TEMP. (F)	TURBINE WHEEL TIP-TO-EXHAUST PR. ΔP (PSI)	DRAIN PRESS (PSIG)	DRAIN TEMP. (°F)	LEAKAGE (FREON 12) (SCFM)	PURGE PRESS U/S ORIF (PSIG)	PURGE PRESS D/S ORIF (PSIG)	PURGE TEMP. U/S ORIF (°F)	PURGE TOTAL FLOW (GN_2) (SCFM)	DRAIN PRESS (PSIG)	DRAIN TEMP (°F)	LEAKAGE (GN_2) (SCFM)
01	786	8.5	0.5	32	1.0	0.8	53	.9	75	48	52	3.8	1.7	52	.3
	837	8.0	0.5	32	1.0	0.8	53	.9	75	49	52	3.7	1.6	52	.3
02	56	8.0	0.4	51	0	0.2	51	.3	74	48	53	3.6	1.5	52	.3
	116	7.5	0.5	51	0	3.5	43	2.3	76	54	53	3.4	1.8	52	.3
03	141	37	2.2	DNR	3.0	8.5	-14	-	57	48	54	1.7	4.8	51	1.4
	231	36	2.2	DNR	3.2	8.0	-15	-	61	48	54	2.2	3.8	51	1.2
	329	35	2.1	DNR	3.0	7.5	-19	-	65	48	53	2.7	-	51	-
	380	34	2.1	DNR	3.6	7.0	- 2	-	73	48	53	3.5	2.5	51	.7
	462	34	2.1	DNR	3.0	7.0	-18	-	66	49	53	2.7	3.1	51	1.0
	554	34	2.1	DNR	3.2	7.5	-15	-	59	49	53	1.9	5.0	50	1.5
	603	34	2.3	30	3.5	9.0	-12	-	52	52	42	-	25.0	16	-
	659	34	2.3	29	3.7	10.3	-10	-	56	57	32	-	22.0	16	-
	803	25	1.7	19	3.0	9.5	-10	-	51	50	35	-	19.0	8	-
	848	30	2.0	22	3.2	8.0	-13	-	50	49	37	-	18.8	6	-
	891	33	2.2	25	3.3	8.0	-14	-	51	49	30	-	20.0	6	-
	964	32	2.1	24.5	3.0	7.0	-16	-	49	43	12	-	9.5	-12	-
04	96	40	2.5	28	4.0	8.0	-17	-	62	48	53	2.3	4.0	44	-
	156	84	7.8	22	8.0	8.5	-15	-	49	49	52	-		42	-
	226	82	7.4	21	8.0	9.0	-14	-	48	52	32	-	24.0	12	-
	258	82	7.4	21	8.0	9.5	-13	-	55	62	18	-	22.0	9	-
	325	77	6.6	21	8.0	11.0	-11	-	78	78	16	-	20.0	6	-
	383	69	5.8	20	7.5	12.0	- 9	-	86	85	20	-	19.0	9	-
	443	65	5.7	20	7.5	O/R	- 3	-	96	96	20	-	21.0	7	-
	480	68	5.6	20	7.0	O/R	- 5	-	76	82	20	-	20.0	7	-
	527	70	5.9	20	7.0	O/R	- 6	-	64	64	20	-	18.0	5	-
05	135	83	7.7	29	7.8	0.2	48	0	116	47	68	10.5	4.2	68	0
	280	71	6.1	27	7.0	0.5	47	0	158	47	69	14.0	3.5	68	0
	619	156	O/R		18.8	0.8	48	0	158	48	69	14.0	14.0	68	2.5
06		ALL DATA IN PARTIAL CAVITATION													
07	114	43	1.9	37	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	145	42	1.9	37	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	166	42	1.9	36	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	188	42	1.9	36	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	214	33	1.5	36	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	232	33	1.5	35	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	252	33	1.5	35	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	272	38	1.9	33	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	290	38	1.9	34	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	308	43	2.0	34	3.0	0	53	0	165	48	63	14.5	8.0	54	0
	324	44	2.1	34	3.0	0	53	0	165	48	63	14.5	8.0	54	0

TABLE IV. (Concluded)

DRIVE TURBINE RAW DATA AT TEST SPEED						DYNAMIC SEAL LEAKAGE DATA										
RUN NO.	TIME SLICE	TURBINE MANIFOLD PR. (PSIG)	TURBINE EXHAUST PR. (PSIG)	TURBINE INLET TOTAL TEMP (F)	TURBINE WHEEL TIP-TO-EXHAUST PR. ΔP (PSI)	PRIMARY SEAL			INTERMEDIATE SEAL				TURBINE SEAL			
						DRAIN PRESS (PSIG)	DRAIN TEMP. (°F)	LEAKAGE (FREON 12) (SCFM)	PURGE PRESS U/S ORIF (PSIG)	PURGE PRESS D/S ORIF (PSIG)	PURGE TEMP. U/S ORIF (°F)	PURGE TOTAL FLOW (GH_e) (SCFM)	DRAIN PRESS (PSIG)	DRAIN TEMP (°F)	LEAKAGE (GN_2) (SCFM)
08	53	95	6.5	35	8.0	0.2	58	0	162	35	65	14.4	5.0	52	0
	70	95	6.6	35	8.0	0.2	58	0	162	35	65	14.4	5.0	52	0
	85	95	6.8	35	8.0	0.2	58	0	162	35	65	14.4	5.0	52	0
	106	78	5.0	34	7.0	0.2	58	0	162	35	65	14.4	4.2	52	0
	129	75	5.0	34	7.0	0.2	58	0	162	35	65	14.4	4.3	52	0
	157	68	4.5	33	7.5	0.2	58	0	162	35	65	14.4	4.2	52	0
	178	67	4.7	32	7.5	0.2	58	0	162	35	65	14.4	4.3	52	0
09	70	152	11.5	44	13.5	0.4	60	0	162	48	58	14.4	10.0	61	0
	100	144	11.7	43	13.0	0.3	60	0	162	48	60	14.4	10.0	61	0
	125.5	135	10.0	42	12.5	0.3	60	0	162	48	60	14.4	8.5	61	0
	154	121	9.1	42	12.0	0.3	60	0	162	48	61	14.4	8.0	61	0
	182	111	8.1	41	11.0	0.3	60	0	162	48	61	14.4	8.0	61	0
	200	96	7.0	40	10.5	0.3	60	0	162	48	61	14.4	8.0	61	0
10	65	202	25	40	12.0	0.5	60	0	162	50	62	14.4	16.0	60	3.2
	73.5	202	25.8	40	12.5	0.5	60	0	156	48	62	13.9	13.0	60	2.3
11	34	45	2.0	43	2.0	0.4	63	0	175	46	68	15.3	5.0	6.5	0
	62	40	1.8	43	2.0	0.4	63	0	172	45	68	15.0	5.0	65	0
	82.5	40	1.8	44	2.0	0.5	63	0	175	44	68	15.3	5.0	65	0
	106.5	40	1.9	44	2.5	0.5	63	0	172	44	68	15.0	5.0	65	0
	130.5	28	1.2	44	2.0	0.5	63	0	172	44	68	15.0	5.0	65	0
	145.5	29	1.2	44	2.0	0.4	63	0	172	44	68	15.0	5.0	65	0

TABLE V. SUMMARY OF REDUCED DATA OBTAINED DURING FREON 12 H-Q TESTING IN ENGINEERING LABORATORY

	REDUCED PUMP DATA AT TEST SPEED								REDUCED PUMP DATA SCALED TO 75,000 RPM							
RUN NO	TIME SLICE	PUMP SPEED (RPM)	PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE TORQUE (IN-LB)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)	ΔT PUMP EFFICIENCY (%)	PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE TORQUE (IN-LB)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)	ΔT PUMP EFFICIENCY (%)	PUMP CONFIG. NUMBER (SEE TABLE II)	REMARKS
01	780	20,000	4.4	121					16.45	1695					I	
	837	20,000	3.2	133					11.97	1860						
02	56	18,000	4.9	97					20.45	1690					I	
	116	20,000	2.9	163					10.85	2280						
03	141	44,400	5.9	730*												
	231	43,100	6.4	686*												
	329	42,000	7.9	602*												
	380	42,000	8.6	501*												
	462	42,300	7.0	644*												
	555	44,000	5.7	732*												
	603	47,000	2.9	873*	5.67	4.22	22.9*	33.6*	4.62	2220*	14.4	17.1	22.9	33.6*	I	
	659	48,500	1.4	940*	5.63	4.34	11.5*	23.5*	2.16	2240*	13.45	16.0	11.5	23.5*		
	803	44,100	1.0	755*	4.27	2.98	9.9*	17.3*	1.70	2180*	12.35	14.6	9.9	17.3*		
	849	45,500	2.9	814*	5.07	3.66	25.0*	29.8*	4.78	2210*	13.75	16.4	25.0	29.8*		
	891	45,500	4.2	799*	5.21	3.76	34.6*	38.3*	6.93	2170*	14.2	16.8	34.6	38.3*		
	904	43,500	6.3	689*	5.49	3.79	44.6*	39.3*	10.85	2045*	16.3	19.4	44.6	39.3*		
04	90	44,100	7.2	682*	6.64	4.65	39.9*	40.9*	12.25	1970*	19.2	22.8	39.9*	40.9*		
	156	60,500	9.1	1144*	12.1	11.6	35.3*	50.1*	11.25	1750*	18.5	21.9	35.3*	50.1*		
	226	60,500	8.2	1265*	11.77	11.3	35.2*	51.4*	10.15	1937*	18.0	21.35	35.2*	51.4*		
	258	61,000	7.5	1374*	11.77	11.4	39.5*	60.0*	9.2	2075*	17.8	21.15	39.5*	60.0*		
	325	62,000	5.5	1458*	9.33	9.13	33.3*	44.6*	6.65	2125*	13.6	16.1	33.3*	44.6*		
	383	61,500	3.7	1460*	8.28	8.08	25.5*	34.7*	4.51	2170*	12.3	14.7	25.5*	34.7*	I	
	443	62,400	1.6	1533*	9.22	9.12	10.3*	I/D	1.92	2210*	13.2	15.8	10.3*	I/D		
	480	61,000	3.8	1464*	9.9	9.6	22.1*	32.3*	4.7	2210*	14.95	17.8	22.1*	32.3*		
	527	60,200	5.6	1289*				39.7*	6.97	2000*				39.7*		
05	135	61,500	8.9	1742	11.85	11.55	50.8	60.8	10.85	2585	17.60	20.85	50.8	60.8	II	ALL OTHER DATA SLICES IN PARTIAL CAVITATION DUE TO INSUFFICIENT INLET PRESSURE.
	280	60,000	7.3	1691	10.42	9.9	46.7	56.7	9.1	2640	16.25	19.5	46.7	56.7		
	619	77,500	9.8	2842	19.95	24.55	42.1	60.3	9.5	2665	18.6	22.25	42.1	60.3		
06															II	ALL DATA IN PARTIAL CAVITATION DUE TO INSUFFICIENT INLET PRESSURE
07	114	46,000	8.2	942	7.36	5.38	55.8	57.8	13.5	2505	19.6	23.4	55.8	57.8		BEARING CAVITY BLED LINE CLOSED.
	145	46,500	7.2	1024	7.17	5.29	54.2	62.3	11.67	2670	18.6	22.1	54.2	62.3		
	166	48,000	6.0	1111	7.12	5.43	47.4	58.7	9.4	2710	17.35	20.6	47.4	58.7	III	
	188	50,500	4.3	1265	6.95	5.58	37.5	46.7	6.38	2790	15.3	18.3	37.5	46.7		
	214	48,500	2.6	1234	5.94	4.58	26.75	32.5	4.02	2945	14.2	16.9	26.75	32.5		
	232	48,500	2.6	1193	5.54	4.26	28.0	43.5	4.02	2850	13.25	15.7	28.0	43.5		BEARING CAVITY BLED LINE OPEN.
	252	46,500	3.9	1092	5.65	4.17	39.6	55.8	6.3	2840	14.7	17.5	39.6	55.8		
	272	47,500	5.8	1098	6.40	4.83	51.0	69.2	9.15	2730	15.95	19.0	51.0	69.2	IV	
	290	46,000	7.2	985	6.49	4.74	58.0	77.8	11.75	2620	17.2	20.5	58.0	77.8		
	308	48,000	7.8	1064	7.25	5.52	58.4	85.2	12.2	2600	17.65	20.95	58.4	85.2		
	324	47,500	8.2	1036	7.40	5.57	59.0	85.9	12.95	2580	18.4	21.9	59.0	85.9		

I/D - INSUFFICIENT DATA TO PERFORM CALCULATION.
* - BASED ON APPARENTLY ERRONEOUS RAW DATA.

TABLE V. (Concluded)

BLN NO	REDUCED PUMP DATA AT TEST SPEED: TIME SLICE	PUMP SPEED (RPM)	PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE TORQUE (IN-LB)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)	ΔT PUMP EFFICIENCY (%)	REDUCED PUMP DATA SCALED TO 75,000 RPM: PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE TORQUE (IN-LB)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)	ΔT PUMP EFFICIENCY (%)	PUMP CONFIG. NUMBER (SEE TABLE II)	REMARKS
08	53	62,000	11.5	1580	14.0	13.8	50.9	63.6	13.9	2310	20.5	24.4	50.9	63.6		
	70	63,000	10.4	1723	13.9	13.9	49.7	66.1	12.4	2440	19.7	23.4	49.7	66.1		
	85	64,000	9.4	1843	13.8	14.0	47.7	63.6	11.05	2530	18.95	22.5	47.7	63.6		
	106	60,500	8.0	1690	11.67	11.2	46.4	61.5	9.9	2600	17.9	21.4	46.4	61.5	III	
	129	63,000	5.4	1899	11.5	11.5	33.9	47.5	6.55	2690	16.3	19.4	33.9	47.5		
	157	63,000	3.1	1979	10.5	10.5	21.9	I/D	3.7	2800	14.9	17.7	21.9	I/D		
	178	65,000	1.6	2090	9.61	9.91	12.4	I/D	1.84	2780	12.8	15.2	12.4	I/D		
09	70	73,000	14.1	1809	18.8	21.8	44.2	56.3	14.47	1905	19.8	23.65	44.2	56.3		PUMP IN PARTIAL CAVITATION DURING 70 SEC TIME SLICE.
	100	73,500	12.0	2443	19.05	22.2	49.8	70.7	12.25	2545	19.85	23.55	49.8	70.7		
	125.5	73,000	10.0	2574	17.95	20.8	46.8	66.7	10.25	2710	18.9	22.6	46.8	66.7	III	
	154	73,500	7.2	2652	16.0	18.65	38.5	52.6	7.35	2760	16.65	19.8	38.5	52.6		
	182	74,500	4.1	2798	14.57	17.35	24.2	I/D	4.13	2840	14.7	17.65	24.2	I/D		
	200	73,500	2.1	2770	12.83	15.0	13.94	I/D	2.14	2880	13.35	15.95	13.94	I/D		
10	65	79,500	14.2	2099	23.8	30.0	46.8	68.8	13.4	2400	21.15	25.2	46.8	68.8	III	TEST TERMINATED WHEN MAX DESIGN DISCHARGE PRESSURE ATTAINED. ALSO REACHED DISCHARGE PRESSURE MAX CHART CALIBRATION RANGE.
	73.5	80,500	12.3	3010	23.6	30.15	46.4	67.8	11.45	2610	20.5	24.6	46.4	67.8		
11	34	46,500	8.1	918	7.75	5.72	49.9	-	13.07	2395	20.10	24.05	49.9	-		
	62	45,000	7.2	914	6.86	4.90	51.2	-	12.0	2530	19.05	22.65	51.2	-		
	82.5	47,000	5.9	1051	6.82	5.09	46.5	-	9.41	2675	17.35	20.60	46.5	-	V	
	106.5	49,500	4.1	1247	6.64	5.22	37.1	-	6.2	2865	15.2	18.10	37.1	-		
	130.5	44,000	2.7	1015	4.92	3.43	30.1	-	4.6	2955	14.3	17.05	30.1	-		
	145.5	45,500	1.5	1099	4.85	3.50	17.7	-	2.47	2980	13.15	15.60	17.7	-		

I/D - INSUFFICIENT DATA AVAILABLE TO PERFORM CALCULATION was conducted at approximately 60 percent of design nominal speed, in the range 42,000 rpm to 48,500 rpm; test #004 was conducted at 80 percent of nominal speed, in the range 60,500 rpm to 62,400 rpm. On the latter two tests, an instrumentation problem was encountered which made the discharge pressure data invalid. On test #004, the primary seal leakage increased substantially, and as a result testing was interrupted and the pump was disassembled for inspection. A total time of 41 minutes was accumulated during the first four tests.

<u>Data Results</u>. The data from the first four tests revealed three potential problem areas:

1. Balance cavity pressure was higher (up to 500 psig, 3.45×10^6 N/m^2) than anticipated. The location of this pressure tap is illustrated in Fig. 19. The bearing cavity pressure shown in the same figure was not incorporated until test #007; therefore, on the initial tests the balance cavity pressure was used not only for thrust calculation, but also for an indication of pressure level to which the primary seal was exposed. Because the balance cavity pressure tap is situated directly in line with the rear wear-ring flow passage, the recorded value includes a substantial portion of the velocity head. As a result of this total pressure effect and because of the radial pressure gradient effect, the recorded value of the balance cavity pressure is substantially higher than the static pressure level in the bearing cavity. However, even after the above factors were discounted, it appeared the design operating limit of the primary seal of 200 psi (1.38×10^6 N/m^2) would be exceeded at full operating speed.
2. A shaft subsynchronous whirl of 0.008-inch (2×10^{-4} m) peak to peak was measured at the turbine disk.
3. High primary seal leakage was encountered.

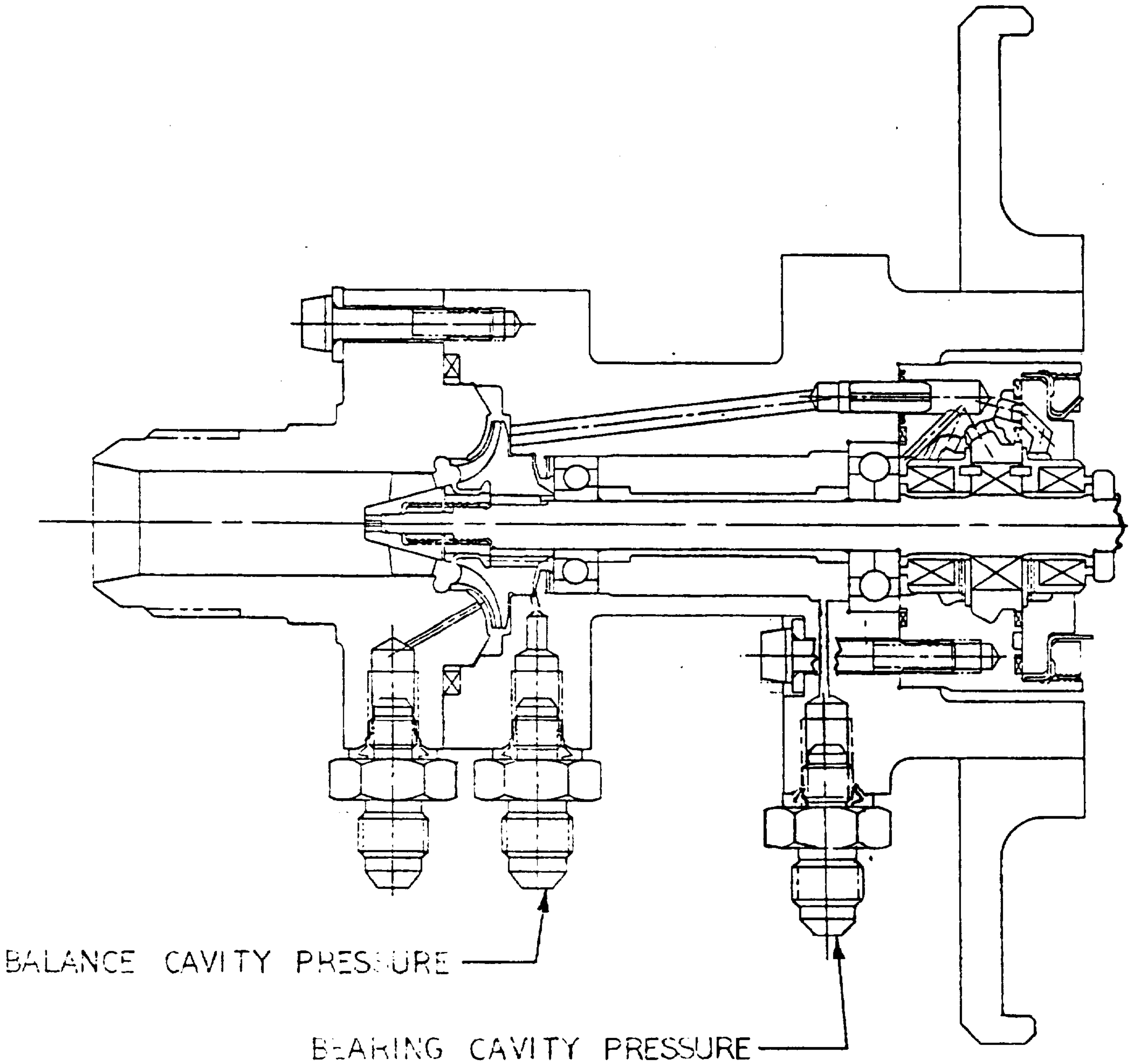

Figure 19. Pressure Tap Locations

Hardware Inspection. Inspection of the hardware after test #004 showed that no rubbing had taken place at the inducer tip or at the impeller wear rings. Both bearings as well as the intermediate seal were in excellent condition. However, the aluminum-oxide coating on the primary seal was worn and chipped and the turbine seal bellows had cracked at the first convolution adjacent to the housing.

Similar turbine seal failures had been encountered with the seal during a bearing and seal test series conducted on the AMPS program. Failure analysis indicated that the bellows failed in fatigue caused by a stick-slip vibration phenomenon. A stick-slip condition can be present when two materials such as carbon and chrome-plated steel are rubbed together without lubrication, and as a result the moving member does not slide smoothly against the static member but sticks and slips alternately. The resulting torsional vibration excites high-frequency axial vibrations in the bellows, leading to eventual fatigue. Although this condition is not anticipated to be present when the seal is operated with actual turbine gas in an engine, effort was initiated to incorporate a damper to ensure successful operation of the seal on this program.

Hardware Modifications. In response to the above observations, three modifications were made for the ensuing tests:

1. In analyzing the potential causes for the high balance cavity pressure, it was noted that the impeller nut restricted, at the impeller inlet face, the return flow path from the bearing cavity. The nut shoulder length was reduced to eliminate this restriction.

2. The second modification made was relative to the shaft whirl. One recognized potential source of excitation for subsynchronous whirl above the first critical is the presence of press fit parts along a rotor due to friction and hysteresis effects (Ref. 5 and 6). It was believed that in the present case the long intereference fit

between the bearing spacer and the shaft could be contributing to the whirl problem. Therefore, the fit was relieved by increasing the spacer inner diameter by 0.002 inch ($.5 \times 10^{-4}$ m), except at the two ends, where the interference was retained for piloting purposes.

3. Special primary seals incorporating carbon noses had been ordered early in the program for the contingency that Freon 12 may prove inadequate as a lubricant for the aluminum-oxide mating surfaces. One of these carbon nose seals was installed for the next test series.

The failed turbine seal was reused for the next test series to retain new seals for incorporating a nose damper. The amount of turbine gas leaking past the failed seal does not materially affect the power output of the turbine.

Tests #005 and #006. Test #005 was run at two speed levels, at an average of 60,000 rpm, and in the range 76,000 to 80,000 rpm. On this test series, the inlet pressure was maintained at the nominal 35-psia ($.241 \times 10^6$ N/m^2) level to minimize the pressure in the bearing cavity, and as a result, cavitation was experienced at the higher flowrates. However, at low Q/N, discharge pressures up to 1800 psig (11.7×10^6 N/m^2) were attained.

On test #005 an instrumentation problem was encountered with the turbine exhaust pressure; as a result, test #006 was planned to be a repeat of the high-speed portion of test #005. On test #006, a speed of 76,000 rpm was attained; however, the test was terminated due to a sudden increase in primary seal leakage, before valid H-Q data points could be obtained.

A total time of 956 seconds was accumulated on tests #005 and #006.

Data Results. A review of the data showed basically the same characteristics which were observed during the initial series. The balance cavity pressure was still high, and excessive primary seal leakage was evident on the last test. Shaft whirling was also present, although the amplitude did not increase above the 0.008-inch (2×10^{-4} m) peak to peak which was observed at the lower speeds.

Hardware Inpsection. The pump was disassembled after test #006, and it was found that the primary seal had incurred a bellows crack. All other hardware was in excellent condition; no rubbing occurred at the inducer tip or the impeller wear rings.

Hardware Modifications. Indications, from the nature of the primary seal failure incurred during test #006, were that the bellows cracked as a result of the high pressure level in the bearing cavity. To reduce the pressure on the primary seal, the backup volute with the original wear-ring diameter was used for subsequent testing, resulting in a reduction of rear wear-ring diametral clearance from 0.006 inch (1.52×10^{-4} m) to 0.0034 inch (0.865×10^{-4} m). A new carbon-nose primary seal and a new turbine seal were installed, and a pressure tap was added to monitor the bearing cavity pressure. This tap can also be used to bleed fluid from the bearing cavity externally, and thus effect a further reduction in the pressure level if required.

Tests #007 through #011. On this test series valid H-Q tests were obtained at 60, 80, 100, and 110 percent of nominal design speed.

On test #007, conducted in the speed range 46,000 to 50,500 rpm, an H-Q pass was made with the external bleed through the bearing cavity pressure line closed. Then, on this test only, the H-Q pass was repeated with the external bleed open and flowing approximately 3/4 gpm ($.47 \times 10^{-4}$ m^3/sec).

After test #007, the impeller was inspected and it was noted that the rear wear ring had just barely touched the housing. After test #008, conducted in the speed range 60,000 to 65,000 rpm, the impeller inspection was repeated, and the rear wear ring was found to have rubbed over 75 percent of its circumference. The nature of the rub was to polish the aluminum-oxide surface on the impeller wear ring, with no measurable increase in clearance. The impeller was substantially in the same condition at the conclusion of the test series.

On test #010, conducted in the speed range 79,500 to 80,500 rpm, a complete H-Q sweep to low Q/N could not be performed because the discharge pressure reached 1980 psig (13.65×10^6 N/m^2), which is already over the design structural limit of the pump.

For test #011, an impeller nut with the original wide shoulder configuration was reinstalled to determine if a high-velocity jet entering the main flow in the impeller from the bearing cavity return passage will have an effect on the developed head. The results showed no appreciable effect.

A total test time of 933 seconds was accumulated during the last series, bringing the test time accumulated during the H-Q tests to 1 hour and 12 minutes.

Data Results. Data from this test series showed an improvement in the pump operation in several respects as a result of the tighter rear wear-ring clearance. The balance cavity pressure was reduced by 120 psi ($.826 \times 10^6$ N/m^2) and the resulting bearing cavity pressure level came within the operating capability of the primary seal. As a result, the leakage was very low from the primary seal on all tests. Also, with the lower rear wear ring clearance, apparently sufficient damping was introduced to reduce the shaft whirl amplitude to an acceptable 0.002 inch (0.0508 mm) peak-to-peak maximum.

Hardware Inspection. After test #011, the pump was disassembled and the components were inspected. The impeller rear wear ring showed a slight amount of additional rubbing beyond that seen after test #008, but the only effect of rubbing was polishing the aluminum-oxide surface. No measurable increase in the clearance took place as a result of the rubbing. Figure 20 shows the appearance of the rear wear ring after test #011.

The turbine seal had incurred a bellows failure identical to that experienced after the initial test series. All other hardware, including the primary seal, was in excellent condition. Figure 21 shows the detail parts after test #011.

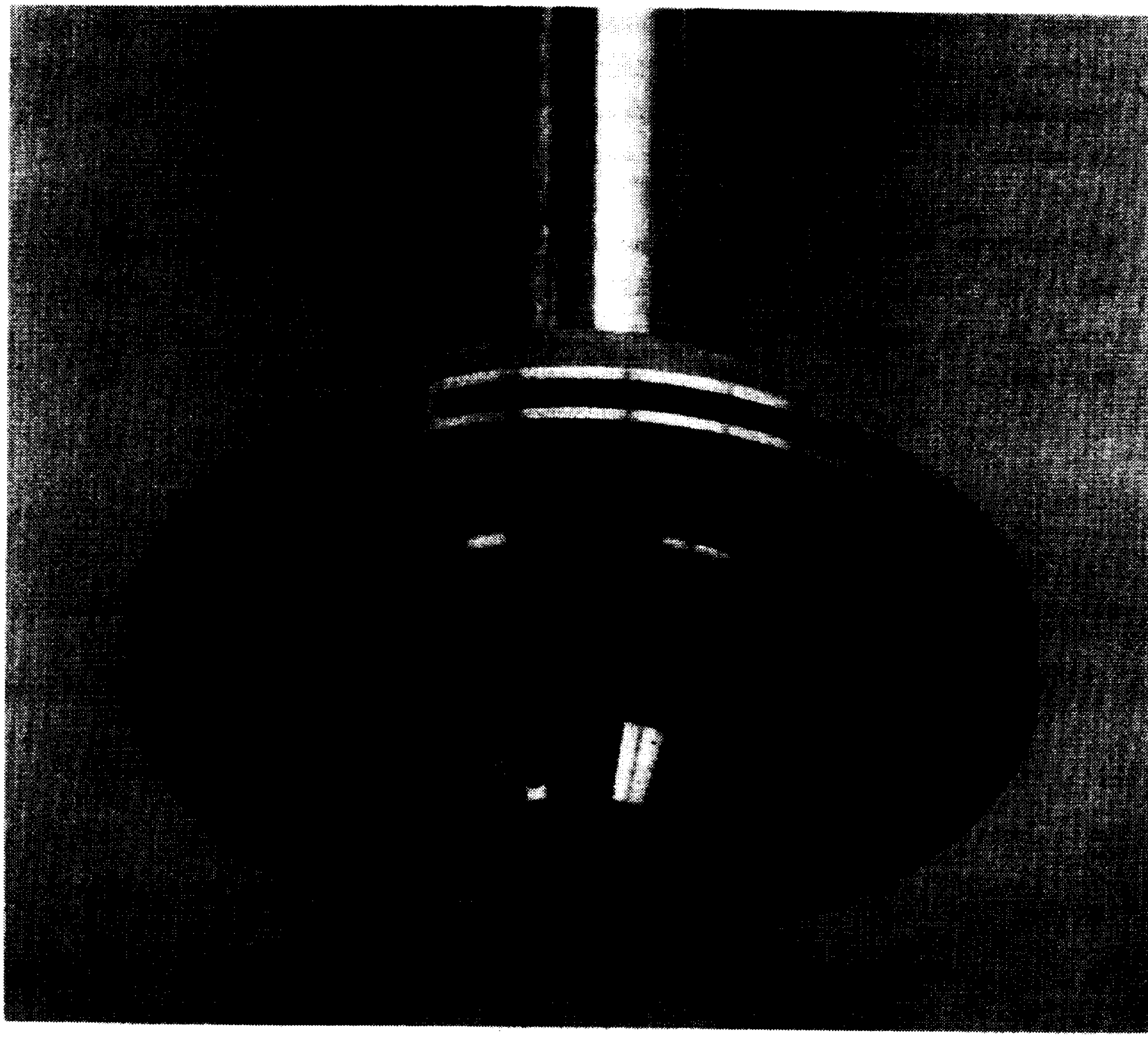

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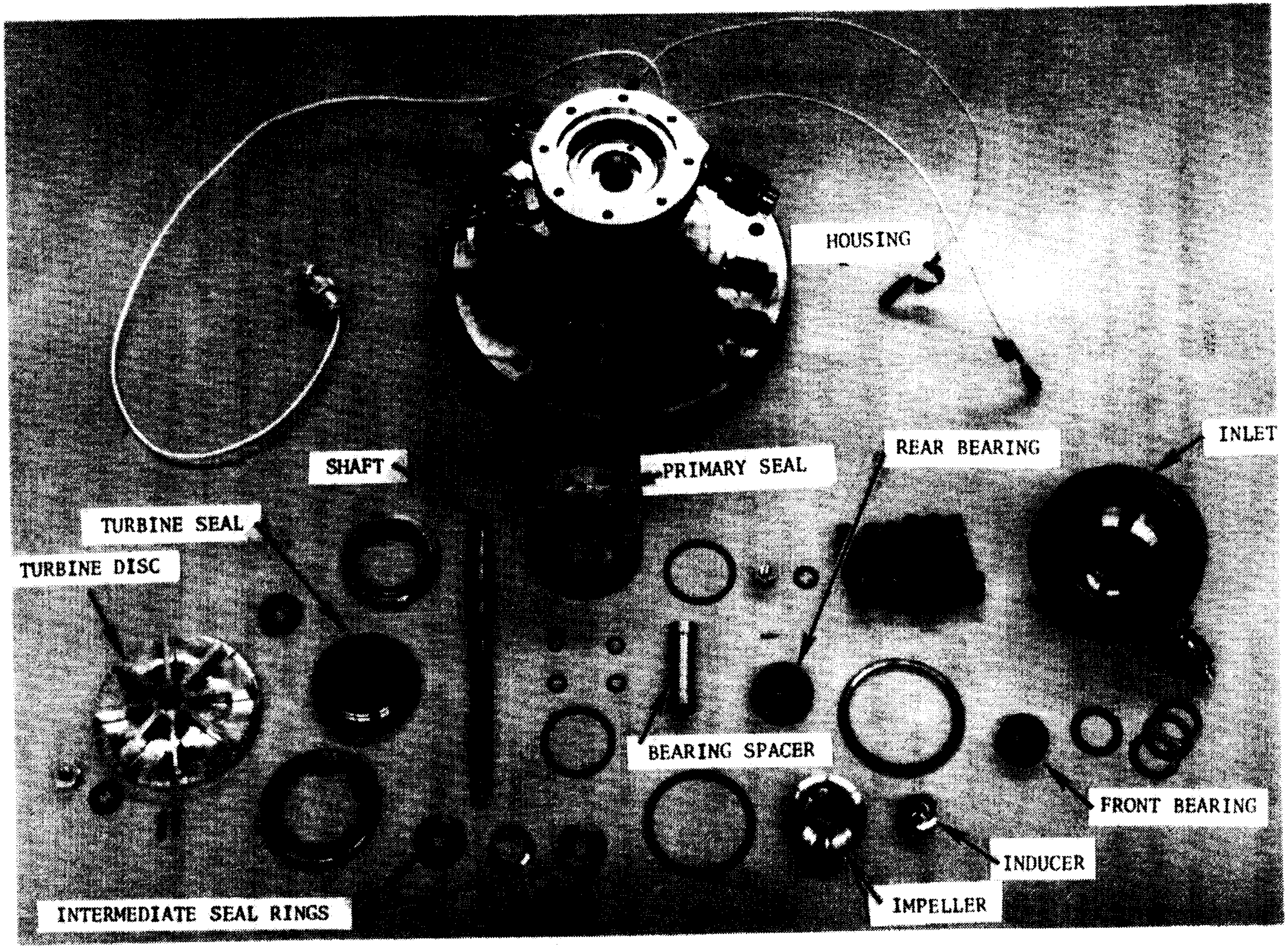
Figure 20. Mark 36 Centrifugal Pump Impeller Rear Wear Ring Hub After Test No. 11



1XY55-6/23/70-C1F

Figure 21. Mark 36 Centrifugal Pump Disassembled After Test No. 11

Freon Suction Performance Tests

The objective of the freon suction performance tests was to determine the minimum net positive head required at the inlet of the pump without substantial degradation in the head developed by the pump. Net positive suction head is defined as the total inlet head available in excess of the vapor head:

$$NPSH = \frac{P_s}{\gamma} + \frac{V^2}{2g} - \frac{P_v}{\gamma}$$

where NPSH = Expressed in feet (m)

P_s = Inlet pressure, psfa (N/m^2)

γ = Specific weight of the liquid, lb/ft^3 (N/m^3)

V = Fluid velocity at inlet, ft/sec (m/sec)

g = Gravitational constant ft/sec^2 (m/sec^2)

P_v = Vapor pressure, psfa (N/m^2)

As noted earlier, precautions were taken to minimize the amount of noncondensible gases in Freon 12 by using instrumentation grade freon, and by evacuating the run tank and pump fluid loop before freon was introduced from the shipping bottles. In addition, the tank ullage, where the noncondensible gases would tend to concentrate was evacuated for a minimum of half an hour whenever the tank was recharged with freon.

The normal test procedure was to start with the inlet pressure set sufficiently high to ensure noncavitating conditions, and after speed and flow have been adjusted to the desired levels, slowly reduce the inlet pressure until the head developed by the pump decreased by 10 percent. At this point the tank pressure was either increased for the next data point or the test was terminated.

Table VI presents the data points obtained during this test series and indicates the critical pump clearances. As during most of the H-Q tests in freon, a carbon nose primary shaft seal was used. The turbine seal was modified to include a corrugated spring wound circumferentially around the nose housing to dampen out vibrations and ensuing bellows cracks.

The inlet and volute housings were modified to incorporate an aluminum oxide coating on the wear ring surfaces. This was done to avoid a potential hazard later in fluorine as a result of rubbing on bare Inco 718. The 0.003 inch ($.76 \times 10^{-4}$ m) diametral clearance which was used on the final H-Q tests in Freon 12 was retained at the rear wear ring. The diametral clearance at the front wear ring was reduced from 0.011 inch (2.8×10^{-4} m) to 0.006 inch (1.52×10^{-4} m). The inducer diametral tip clearance was also reduced from .018 to .011 inch (4.6×10^{-4}m to 2.8×10^{-4}m).

Tests #12 and #13. Test No. 12 was conducted at speeds up to 20,000 rpm for a duration of 211 seconds. Test No. 13 was of 235 seconds duration, with a maximum speed of 25,000 rpm. Analysis of the Bently proximity detector data from these two tests indicated a synchronous shaft radial deflection in the plane of the turbine disc of 0.008 inch (2×10^{-4}m). Since this deflection was more than twice the level seen in this speed range on earlier tests, the pump was removed from the facility and disassembled for a rotor balance check.

Hardware Inspection. Disassembly revealed no rubbing at the inducer tip and at the front wear ring and only minor contact at the rear wear ring. All shaft seals were found in excellent condition. Of particular significance was that the turbine seal bellows was intact, indicating that the damper incorporated into the seal corrected the vibration and cracking problem which characterized the original design.

TABLE VI. MARK 36 (12 GPM) CENTRIFUGAL PUMP SUMMARY OF SUCTION PERFORMANCE TESTS IN FREON 12 REAR WEAR RING CLEARANCE = 0.0031 in. (0.79×10^{-4} m)

Test No.	Duration (sec)	Speed (rpm)	Delivered Flow (gpm)	Delivered Flow (m^3/sec $x10^4$)	Inducer Clear. (in.)	Inducer Clear. (m $x10^4$)	Impeller Frt. Wear Ring Cl. (in.)	Impeller Frt. Wear Ring Cl. (m $x10^4$)	Brg. Cavity Bleed	Remarks
12	211	20,000	3.1	1.96	.011	2.8	.006	1.5	Open	Facility & pump mechanical checkout. Turbine disc defl. = .008 in. P/P. ($2 x10^{-4}$ m)
13	235	25,000	-		.011	2.8	.006	1.5	Open	As above.
14	5	33,000	-		.011	2.8	.006	1.5	Open	Rotordynamic check. Turbine disc defl. = .032 in P/P. ($8.1 x10^{-4}$ m)
15	295	15,600	-		.018	4.6	.011	2.8	Open	Erroneous speed ind.
16	408	28,000	-		.018	4.6	.011	2.8	Open	Rotordynamic check. Turbine disc defl. = .002 in P/P. ($.5 x10^{-4}$ m)
17	2500	60,000	9.6	6.1	.018	4.6	.011	2.8	Open	Satisfactory cav. data
		58,500	7.3	4.6					Open	Satisfactory cav. data
		58,500	5.6	3.5					Open	Satisfactory cav. data
		71,000	7.2	4.5					Open	Satisfactory cav. data
		73,800	9.2	5.8					Open	No non-cav. reference
		74,500	11.6	7.3					Open	No non-cav. reference
18	600	74,000	12.0	7.6	.018	4.6	.011	2.8	Open	No non-cav. reference
		74,000	12.0	7.6	.018	4.6	.011	2.8	Closed	No non-cav. reference
19	476	62,000	9.6	6.1	.011	2.8	.006	1.5	Open	Satisfactory cav. data
		63,000	10.6	6.7					Open	Satisfactory cav. data

TABLE VI. (Concluded)

Test No.	Duration (sec)	Speed (rpm)	Delivered Flow		Inducer Clear.		Impeller Frt. Wear Ring Cl.		Brg. Cavity Bleed	Remarks
			(gpm)	(m^3/sec x10^4)	(in.)	(m x10^4)	(in.)	(m x10^4)		
20	148	61,000	7.3	4.6	.011	2.8	.006	1.5	Open	Satisfactory cav. data
21	197	62,000	5.5	3.5	.011	2.8	.006	1.5	Open	Satisfactory cav. data
22	103	62,000	11.2	7.1	.011	2.8	.006	1.5	Open	Satisfactory cav. data
23	98	74,000	11.7	7.4	.011	2.8	.006	1.5	Open	Satisfactory cav. data
24	82	72,000	14.4	9.1	.011	2.8	.006	1.5	Open	No non-cav. reference
25	190	74,000	13.4	8.5	.011	2.8	.006	1.5	Open	No non-cav. reference
26	121	73,300	9.6	6.1	.011	2.8	.006	1.5	Open	Satisfactory cav. data
27	185	71,000	7.2	4.5	.011	2.8	.006	1.5	Open	Did not reach cav.
28	110	60,000	9.0	5.7	.011	2.8	.006	1.5	Closed	Satisfactory cav. data
29	161	61,700	9.3	5.9	.011	2.8	.006	1.5	Closed	Satisfactory cav. data
30	125	73,000	11.8	7.4	.011	2.8	.006	1.5	Closed	Satisfactory cav. data

Hardware Modification. A balance check of the rotor assembly disclosed 0.037 gram-inch (9.2×10^{-8} Nm) unbalance in the plane of the impeller and 0.051 gram-inch (12.7×10^{-8} Nm) unbalance in the plane of the turbine disc. Grind corrections were made until the unbalance was reduced to 0.011 and 0.019 gram-inch (2.7×10^{-8} and 4.7×10^{-8} Nm) in the planes of the impeller and turbine disc, respectively. The pump was reassembled, with runouts being taken at each significant step to guard against changes in the unbalance. Two springs were used in parallel to preload the bearings to 140 lb (622 N).

Test #014. On Test No. 14 a speed of 33,000 rpm was attained, but the test had to be aborted five seconds after start because the turbine disc deflections reached an amplitude of .032 inch peak-to-peak (8.1×10^{-4} m).

Hardware Inspection. The radial runout of the turbine disc after the test was 0.018 inch (4.6×10^{-4} m) total indicated reading compared to 0.0007 inch ($.18 \times 10^{-4}$ m) before the test. Since this indicated a bend or distortion in the rotor, the pump was disassembled. Again, no rubbing was evident in the pump, although the turbine disc rubbed radially at the tip as well as axially on the upstream surface. The primary and turbine seals were in good condition, as were the intermediate seal rings, despite the fact that the intermediate seal anti-rotation pins were dislodged from the housings. A runout of .0012 inch ($.3 \times 10^{-4}$ m) was measured at the turbine end of the shaft after disassembly indicating that the shaft was bent slightly. Detail inspection of the other hardware disclosed that the bearing spacer ends were 0.0037 inch ($.94 \times 10^{-4}$ m) out of parallel compared to a print requirement of 0.0005 inch ($.13 \times 10^{-4}$ m) maximum.

Hardware Modification. Analysis of the hardware condition and data indicated that the probable cause for the rotor instability was the discrepant bearing spacer. Three other potential excitation sources were also present: A small amount of unbalance could be introduced into the rotor when the shaft is removed from the balance bearings and mounted in the run bearing. Due to repeated rebuilds the turbine disc to shaft pilot fit became looser than during earlier successful testing and this could result in a relocation error during final assembly. And finally a hydrodynamic excitation could be generated at the front wear ring, where the diametral clearance was reduced to 0.006 inch (1.5×10^{-4}m) from 0.011 inch (2.8×10^{-4}m) used during successful testing.

To ensure successful mechanical operation of the pump, the following changes were made:

1. A new bearing spacer was used which had the ends parallel within 0.0005 inch ($.13 \times 10^{-4}$ m).
2. The balancing fixture was modified to permit balancing with the actual test bearings. The two bearings and the bearing spacer remained on the shaft after balancing.
3. The backup turbine disc was used with a fit of 0.0008 inch ($.20 \times 10^{-4}$ m) interference on the shaft pilot versus 0.0004 inch ($.10 \times 10^{-4}$ m) with the used disc.
4. An inlet housing with a larger front wear ring diameter and inducer tunnel diameter was used, resulting in .011 inch (2.8×10^{-4} m) front wear ring diametral clearance.

The rotor was rebalanced and the pump was reassembled with the above modifications.

Tests #15 through #18. The objective of tests No. 15 and No. 16 was to ramp the speed slowly to approximately 28,000 rpm, just under the first critical speed, and observe the rotor deflections for signs of unbalance. Test No. 15 was terminated 295 seconds after start with the speed at 15,600 rpm, because an erroneous speed printout indicated 31,000 rpm. On test No. 16 a speed of 28,000 rpm was attained. The radial deflections in the plane of the turbine disc remained at a very low level on both tests, indicating that the source of rotor instability had been removed.

Test No. 17 was conducted for a duration of 2500 seconds (41.7 minutes). Valid suction performance data was obtained at 59,000 rpm (80 percent of nominal speed) at 60 percent, 80 percent and 100 percent of nominal Q/N. Satisfactory data was also obtained at 71,000 rpm and 60 percent of nominal Q/N. Test points at higher Q/N values were attempted, however, due to facility limitations the pump inlet pressure could not be sufficiently increased to obtain a non-cavitating reference point.

On test No. 18 another attempt was made to obtain the high Q/N data at 75,000 rpm, but the inlet pressure still proved to be insufficient. The duration of test No. 18 was 600 seconds.

The rotordynamic operation of the pump was excellent on both test No. 17 and No. 18; the maximum turbine disc radial motion noted was approximately .001 to .002 inch ($.25 \times 10^{-4}$ m to $.5 \times 10^{-4}$ m).

<u>Hardware Modifications</u>. In view of the satisfactory rotordynamic performance of the pump on the foregoing tests, it was decided to install the alternate inlet housing which resulted in an inducer diametral tip clearance of .011 inch (2.8×10^{-4} m) and an impeller front wear ring diametral clearance of .006 inch (1.5×10^{-4} m). At the same time modifications were made to the facility to reduce the pressure drop from the run tank to the pump inlet to facilitate obtaining non-cavitating reference points at the higher flowrates.

<u>Tests #19 through #30</u>. With the foregoing modifications test No. 19 through No. 30 were conducted, during which satisfactory data was accumulated to define the suction performance of the pump over the required Q/N range. On test No. 19 through No. 27 the bearing cavity bleed was maintained open and approximately 3/4 gpm ($.47 \times 10^{-4}$ m^3/sec) was returned externally to the run tank. On the last three tests the bleed was closed and two data points were repeated to determine the effect on suction performance.

Rotordynamically the pump operated very smoothly, the deflections in the plane of the turbine disc being of the order of .001 to .002 inch ($.25 \times 10^{-4}$ m to $.5 \times 10^{-4}$ m). Seal performance was also excellent, with no measurable leakage from either the primary or the turbine shaft seals.

Hardware Inspection. Disassembly of the pump after the test series disclosed some pitting on the front bearing inner race indicating the beginning of a fatigue condition. The balls and outer race of the front bearing as well as all components of the rear bearing showed no deleterious effects. The total accumulated time on these bearings was 1.9 hours in Freon-12. An inspection of the bearing ball diameters disclosed the following values:

Rear Bearing		Front Bearing	
Inch	m $\times 10^3$	Inch	m $\times 10^3$
.187520	4.76301	.156448	3.97378
.187520	4.76301	.156488	3.97480
.187510	4.76275	.156468	3.97429
.187530	4.76326	.156448	3.97378
.187510	4.76275	.156488	3.97480
.187520	4.76301	.156448	3.97378
.187510	4.76275	.156448	3.97378
.187520	4.76301		

These compare with the nominal drawing requirement of .1875 inch (4.7625×10^{-3} m) for the rear bearing and .15625 inch (3.96875×10^{-3} m) for the front bearing balls. Thus it appears that no significant ball wear had taken place during the tests.

All other components of the pump were in excellent condition.

Hydrodynamic Performance in Freon 12

The H-Q data obtained at the various test speed levels is presented in Fig. 22 . The design point is shown at 12 gpm (7.57×10^{-4} m^3/sec) delivered flowrate and 2265 ft (691 m) developed head. The actual head developed by the pump at 12 gpm and the nominal design speed of 75,000 rpm exceeded the design requirement by approximately 300 feet (92 m). As a result, design flow and head can be achieved by operating the pump at a lower speed of 71,200 rpm instead of 75,000 rpm.

Figure 23 shows the pump performance curves scaled from the various speeds at which the tests were run, to the design nominal speed of 75,000 rpm. The predicted overall efficiency with the wear-ring clearances used during the last test series was 47 percent; the actual average efficiency obtained was 5 percent-point higher at 52 percent.

In Fig. 24 , the various types of efficiencies calculated from the data are presented. The curve denoted "hydraulic efficiency" indicates the actual head developed by the pump as a percent of the ideal head which could be generated. This bears no relation to the power requirement of the pump; it is strictly an indication of its head-producing capability. The predicted value for this efficiency at nominal flow was 91 percent; the actual average obtained was 92 percent.

The isentropic efficiency is obtained by dividing the fluid horsepower $\frac{\dot{w}H}{550}$ by $C_p \dot{w} \Delta T \frac{778}{550}$, where ΔT is the fluid temperature rise from the pump inlet to discharge. This efficiency accounts for all internal losses in the pump due to recirculation, disk friction, incidence, diffusion, etc. It also includes the bearing friction losses, which are considered very small.

The overall efficiency curve is obtained by dividing the delivered fluid power of the pump by the power output of the drive turbine. The difference between the isentropic efficiency and the overall efficiency represents the amount of power loss due to the seal package. Based on AMPS bearing and

SYM.	TEST NO.	AVERAGE TEST SPEED
▽	02	20,000 RPM
○	07	47,000 RPM
△	08	62,000 RPM
◇	09	74,000 RPM
0	10	80,000 RPM

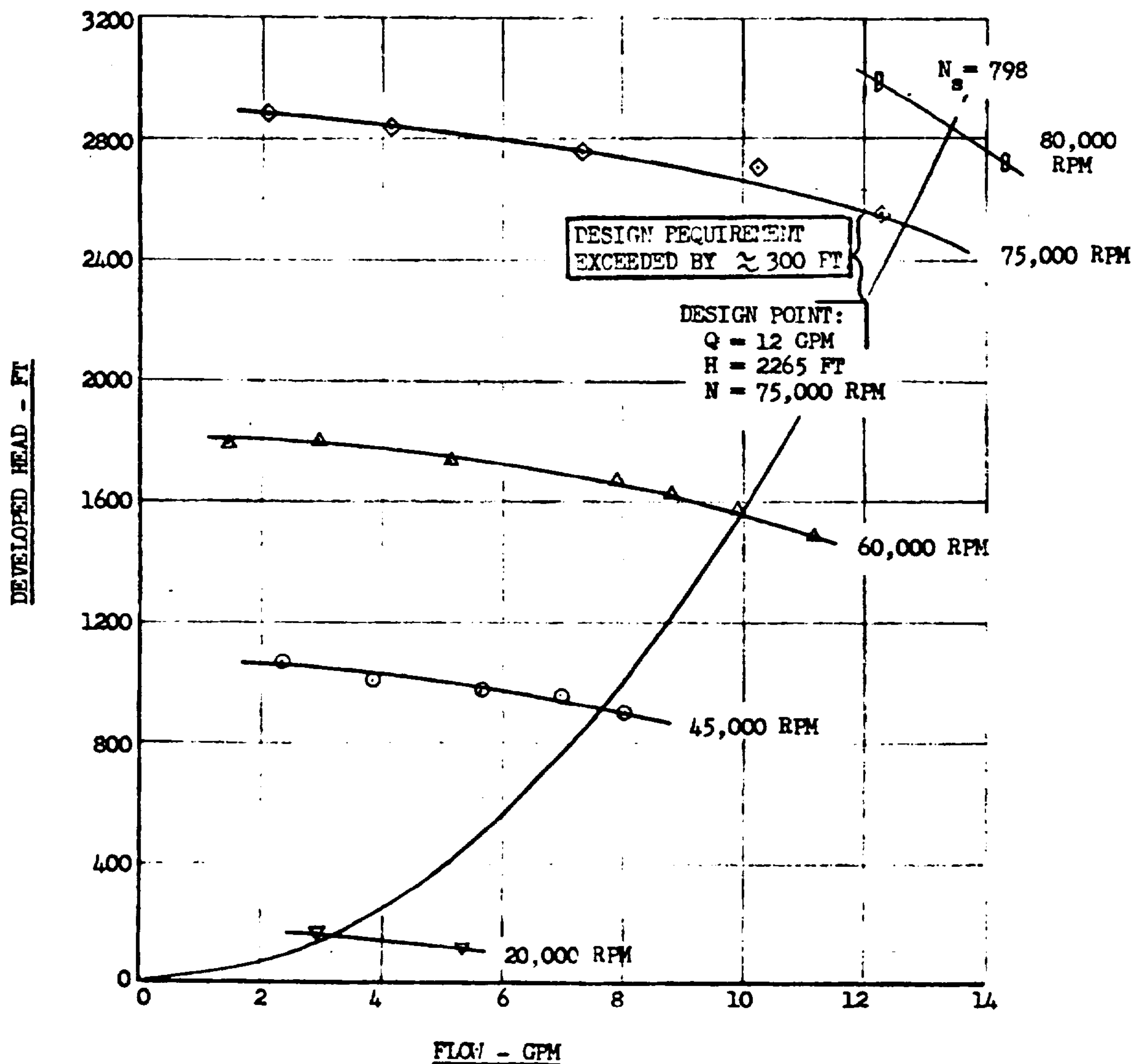


Figure 22. Mark 36 (12 gpm) Centrifugal Pump in Freon 12

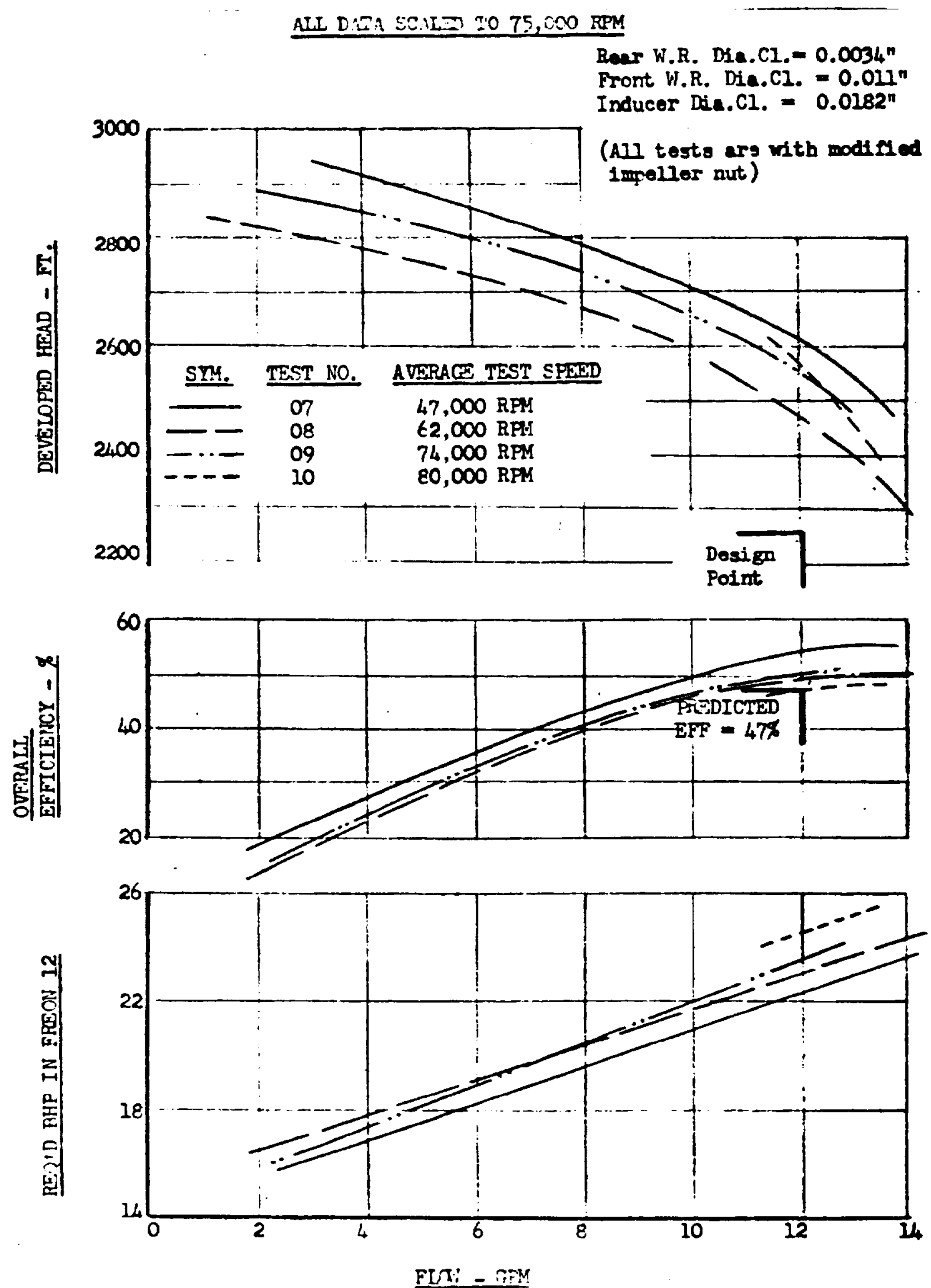


Figure 23. Mark 36 (12 gpm) Centrifugal Pump in Freon 12

CURVE SPEED = 75,000 RPM

SYM	TEST NO.	AVERAGE SPEED
○	07	47,000 RPM
△	08	62,000 RPM
◇	09	74,000 RPM
0	10	80,000 RPM

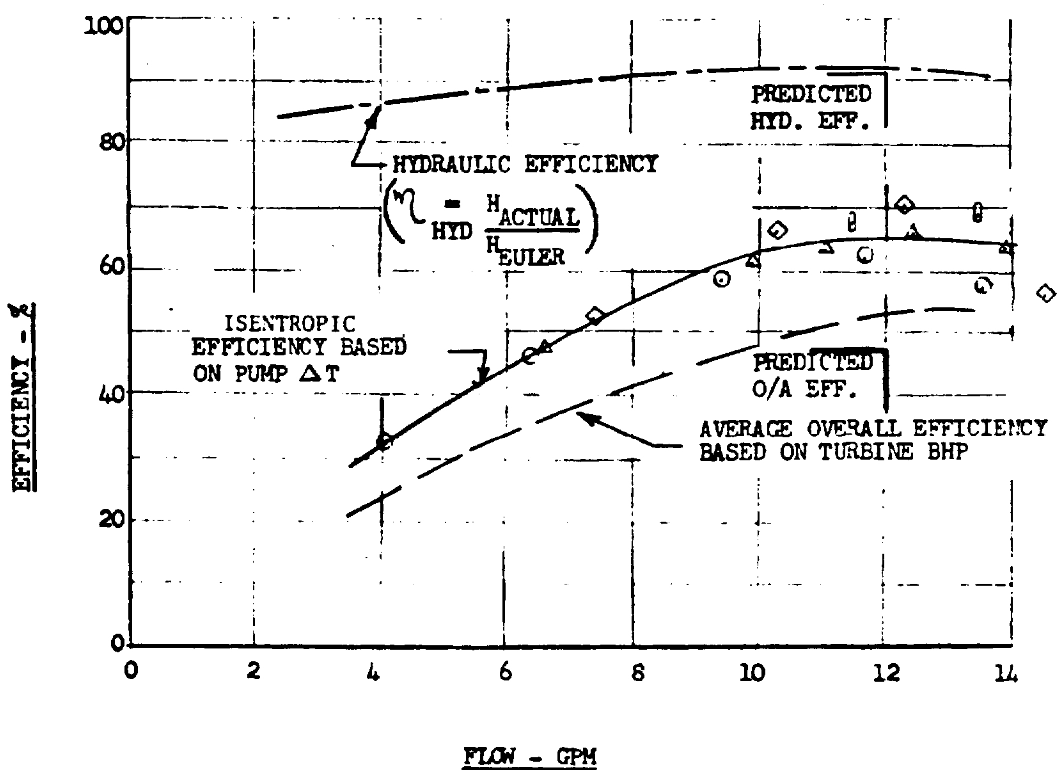


Figure 24. Mark 36 (12 gpm) Centrifugal Pump in Freon 12; Comparison of Hydraulic and Overall Efficiencies

seal test data, this power is calculated at 4.8 HP (3580 watts) with the primary seal pressurized to 200 psig (1.38×10^6 N/m^2). With the seal package absorbing 4.8 HP, the difference between the isentropic efficiency and the overall efficiency should be 13 percentage-points. The actual data correlates with that very well, in that the difference between the average curves is exactly 13 percentage points at nominal Q/N.

Figure 25 shows the effect of rear wear-ring clearance and external bleeding on the balance cavity and bearing cavity pressure levels. Balance cavity pressure minus inlet pressure and bearing cavity pressure minus inlet pressure are plotted as a function of delivered flowrate, with all data scaled to 75,000 rpm. At the nominal 12 gpm (7.57×10^{-4} m^3/sec) flowrate, the balance cavity pressure level was reduced by 120 psi ($.8 \times 10^6$ N/m^2) with the tighter rear wear-ring clearance. An additional reduction of 40 psi ($.275 \times 10^6$ N/m^2) was realized by opening the overboard bleed.

The bearing cavity pressure was not measured during the initial tests, thus data with the large wear-ring clearance is not included in Fig. 25 . With the small wear-ring clearance, the bearing cavity pressure was 160 to 200 psi (1.1×10^6 to 1.38×10^6 N/m^2) above inlet pressure (at nominal flow). With the external bleed open, the pressure level is reduced to 120 psi ($.8 \times 10^6$ N/m^2) over the inlet, which is in the intended operating range for the primary seal.

The performance effects associated with external bleeding of the bearing cavity are illustrated in Fig. 26 . The developed head is not affected appreciably by bleeding; however, the pump efficiency is 4 percent-points higher with the external bleed open and flowing 3/4 gpm ($.47 \times 10^{-4} m^5/sec$). This is attributed to the fact that, with the bleed open, the return flow through the impeller holes is less, and as a result less disturbance is introduced to the main flow between the inducer and impeller. A slight improvement in performance is also realized from the fact that, with the lower bearing cavity pressure, the friction torque of the primary seal is reduced.

ALL DATA SCALED TO 75,000 RPM

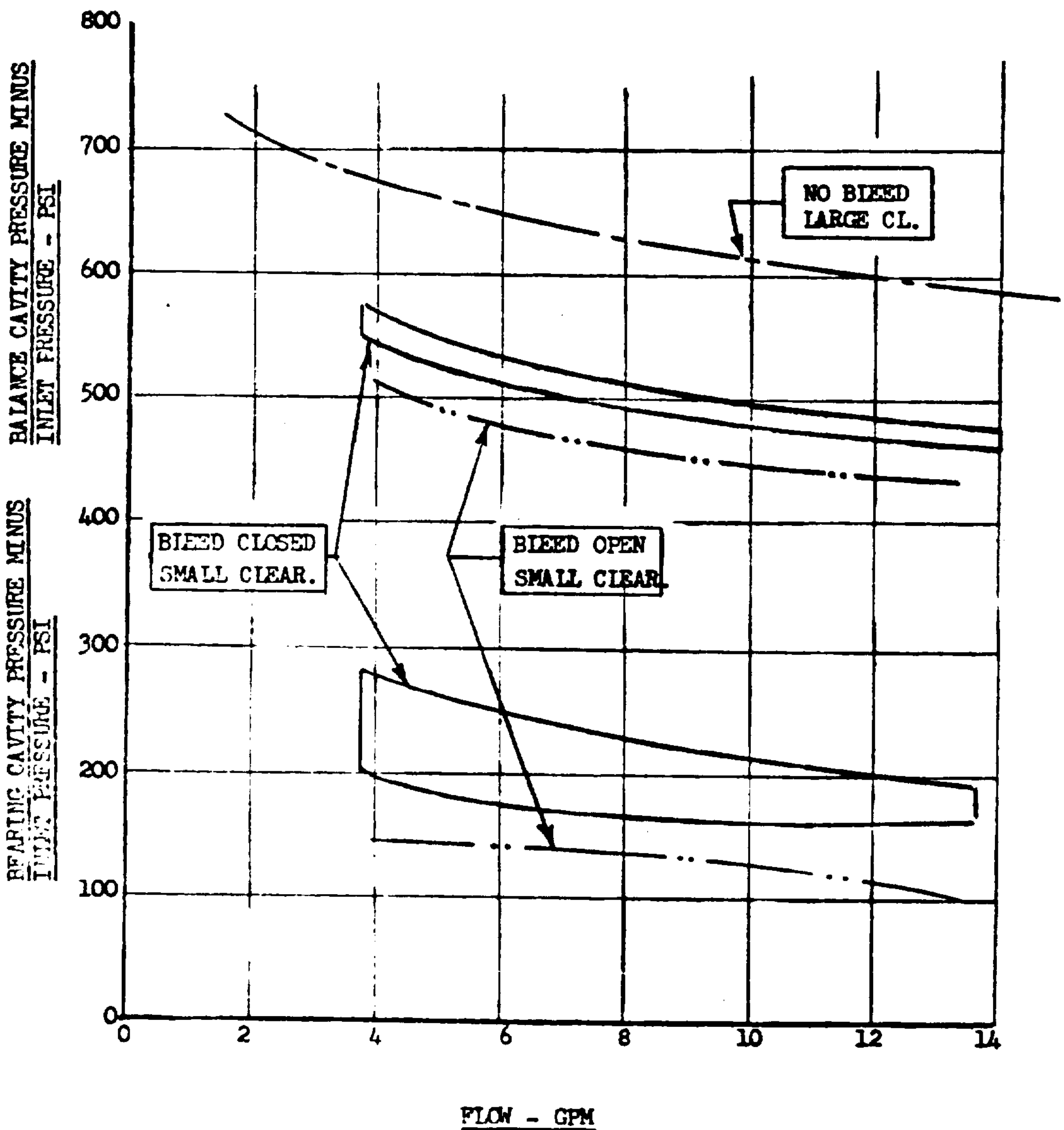


Figure 25. Mark 36 (12 gpm) Centrifugal Pump in Freon 12; Effect of Rear Wear Clearance and Bearing Cavity Bleed Flow on Balance Cavity and Bearing Cavity Pressures

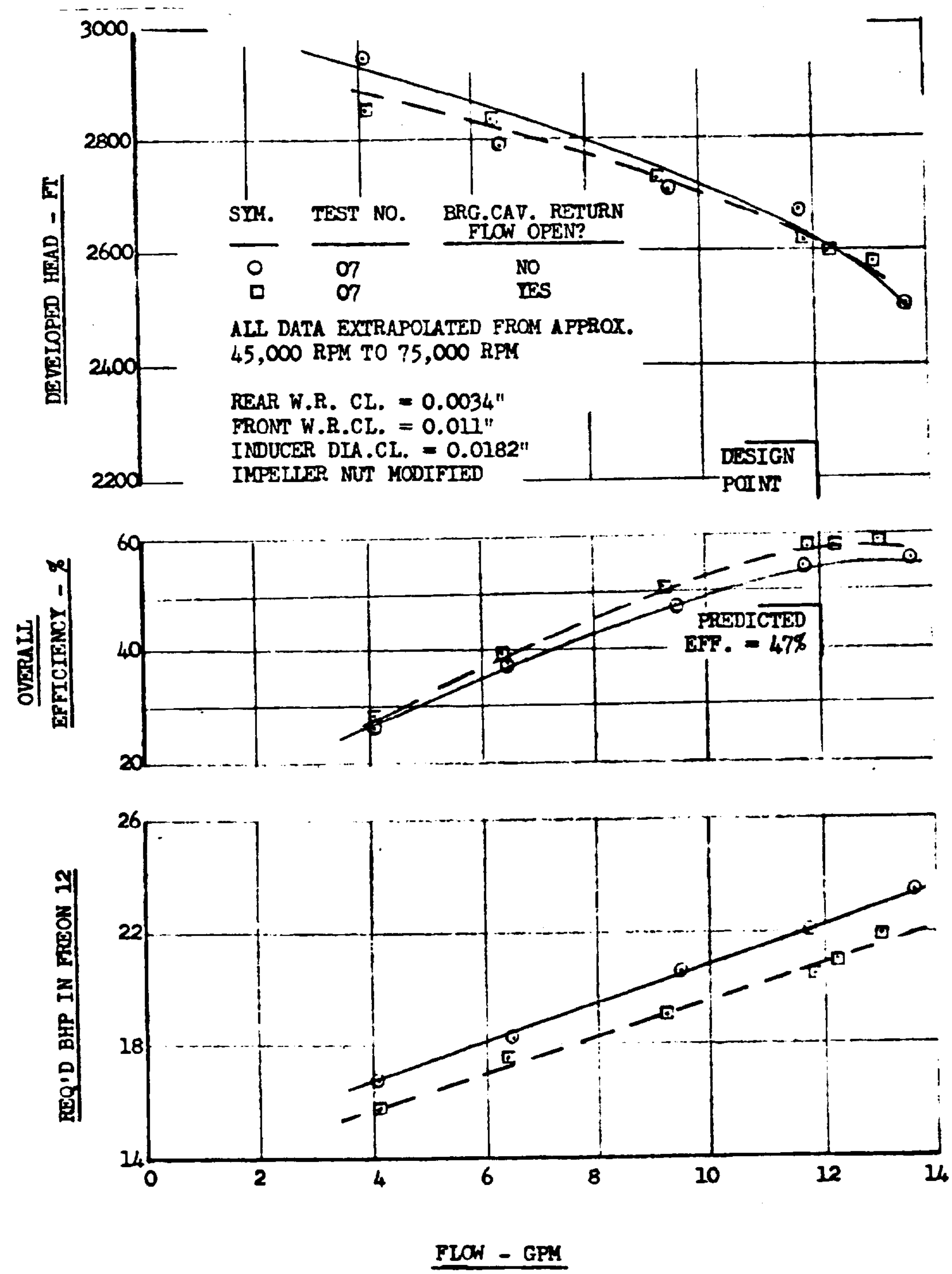


Figure 26. Mark 36 (12 gpm) Centrifugal Pump in Freon 12; Performance Comparison With and Without Bearing Cavity Return Flow

The suction performance data obtained with the inducer tip diametral clearance at .018 inch (2.8×10^{-4} m) and .011 inch (2.8×10^{-4} m) is presented in Fig. 27 and 28 , respectively. In Fig. 29 the NPSH level at 2 percent and 4 percent head drop is plotted as a function of inducer flowrate for .011 inch (2.8×10^{-4} m) inducer clearance. The same figure also shows the NPSH requirement at 2 percent head drop with an inducer clearance of 0.018 inch (4.6×10^{-4} m) diametral clearance. It is noteworthy that little difference exists between the 2 percent and 4 percent criteria, and indication of a steep $\Delta H/NPSH$ characteristic in the critical zone. In examaning Fig. 28 this is seen particularly true at low flowrates; as the flowrate increases there is a growing tendency to droop.

In Fig. 30 the suction specific speed as a function of flowrate is presented for the final configuration (.011 inch (2.8×10^{-4} m) clearance) using the 2 percent ΔH as the criteria. Suction specific speed is defined by the relationship:

$$N_{SS} = \frac{NQ^{1/2}}{(NPSH_{CRIT})^{3/4}}$$

where

N = speed, rpm

Q = inducer flow, gpm (m^3/sec)

$NPSH_{CRIT}$ = net positive suction head at 2% head drop, ft (m)

The above expression can be converted into dimensionless form by multiplying by the factor 3.66×10^{-4} (Ref. Eq. 1a, Volume I). The maximum suction specific speed attained was 36,500 (13.3 dimensionless) at 9 gpm (5.68×10^{-4} m^3/sec). At the design flow of 12 gpm ($7.56 \times 10^{-4} m^3$/sec) the suction specific speed was 21,000 (7.7 dimensionless). The peak suction performance could be shifted to the design flow by increasing the inducer diameter.

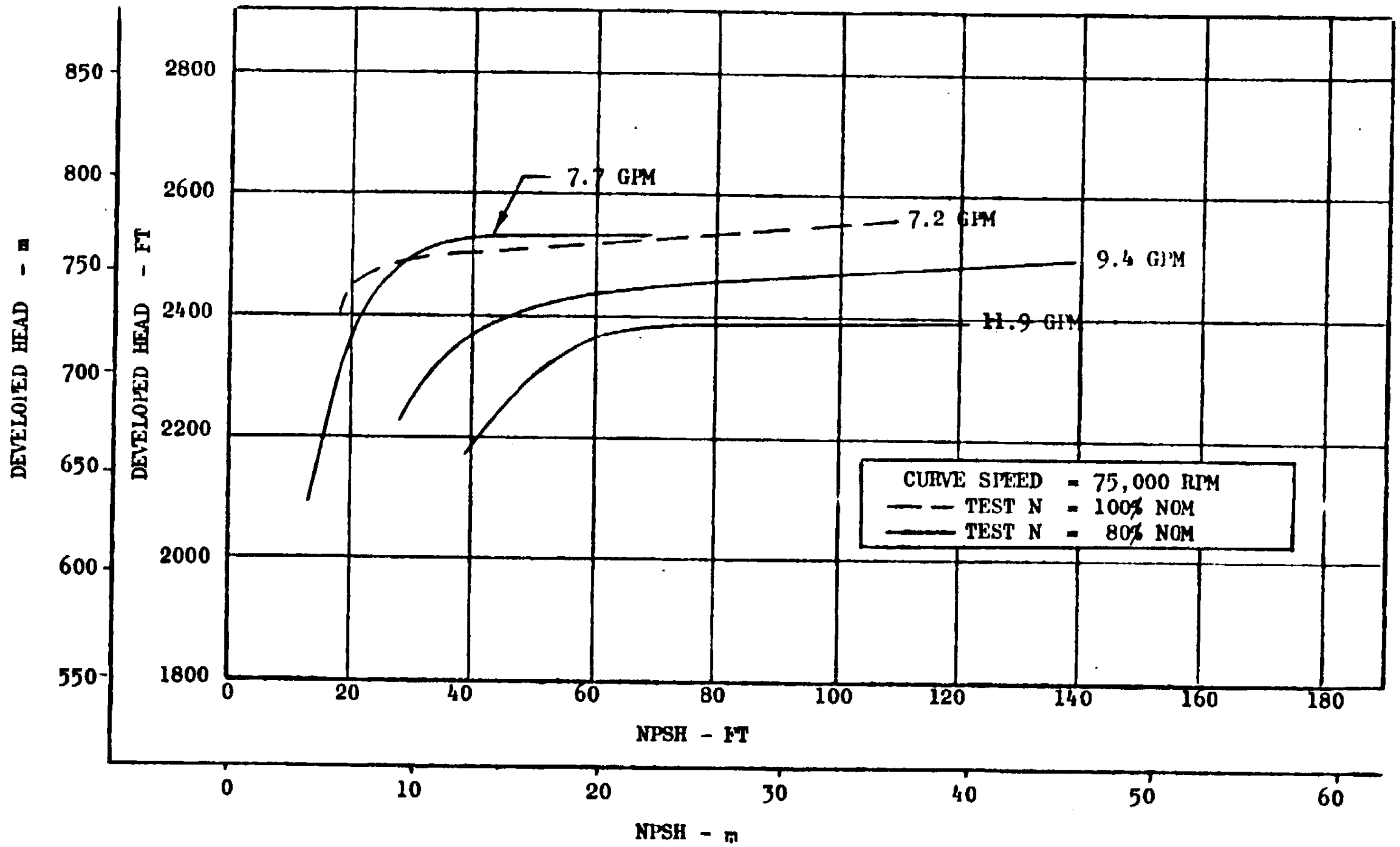


Figure 27. 12 gpm Rotating Pump Suction Performance in Freon 12 (Inducer Clear = 0.018 in. (4.4×10^{-4}m) Diameter)

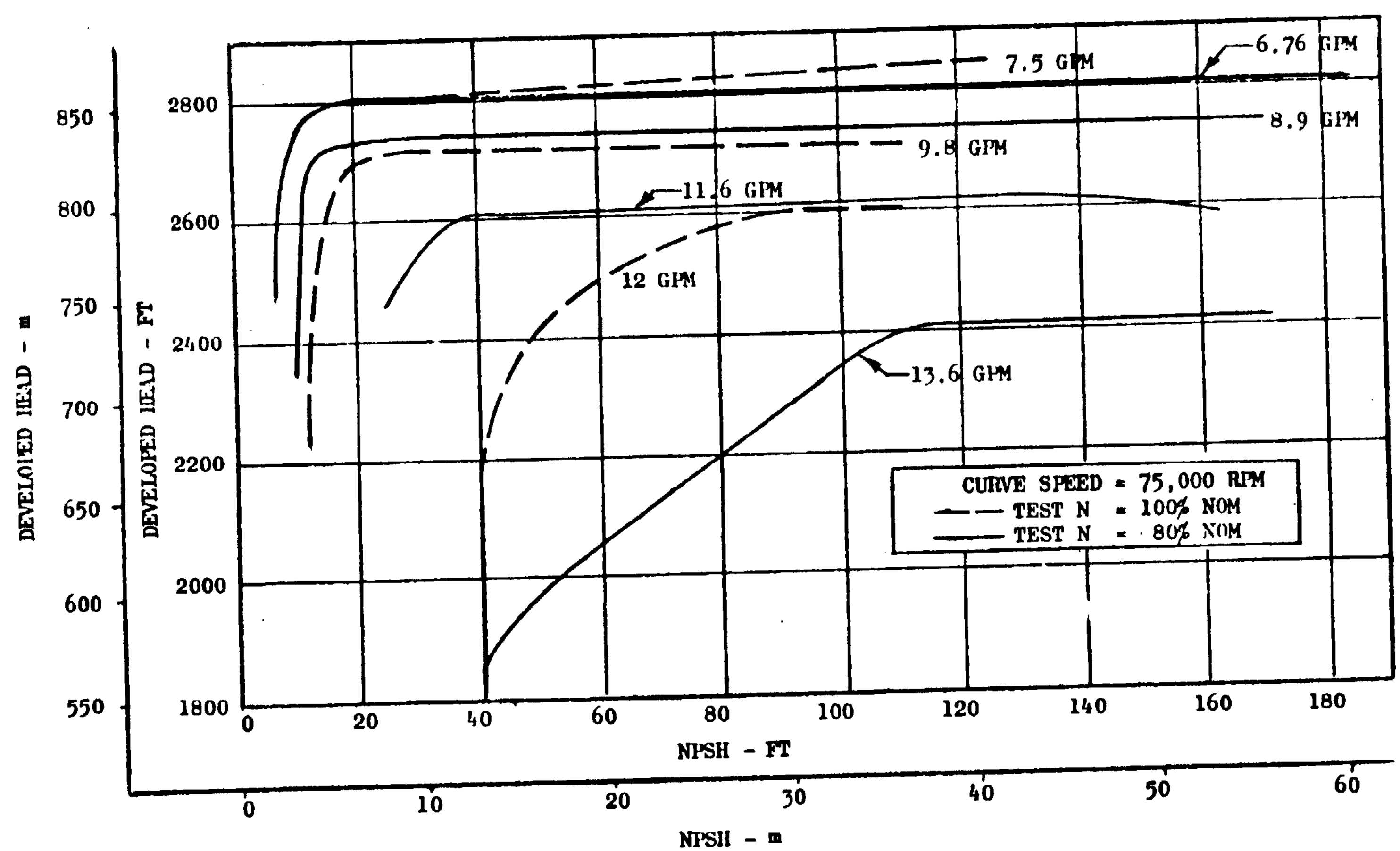


Figure 28. 12 gpm Rotating Pump Suction Performance in Freon 12 (Inducer Tip Clear = 0.011 in. (2.8×10^{-4}m) Diameter)

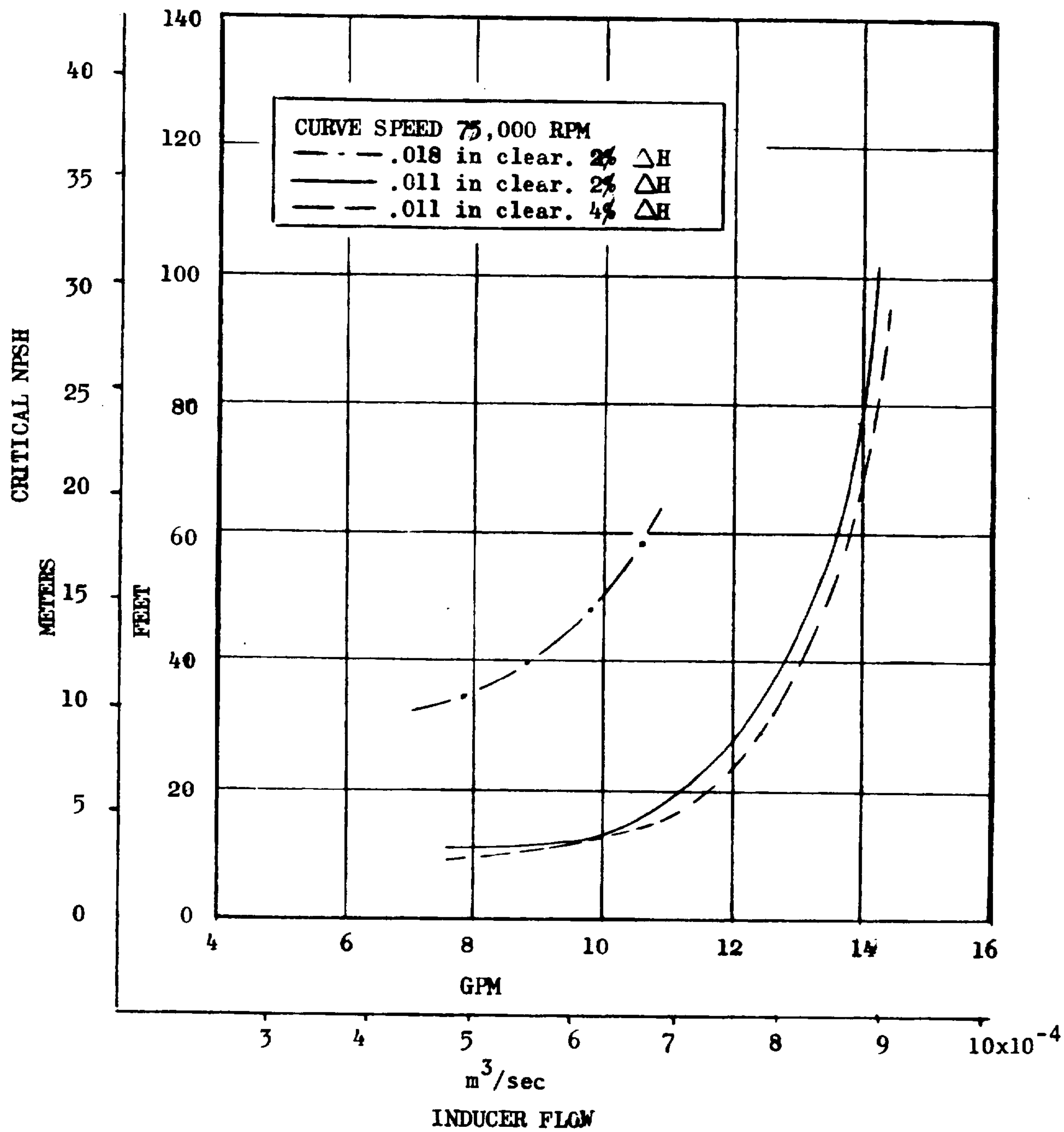


Figure 29. 12 gpm Rotating Pump Suction Performance in Freon 12

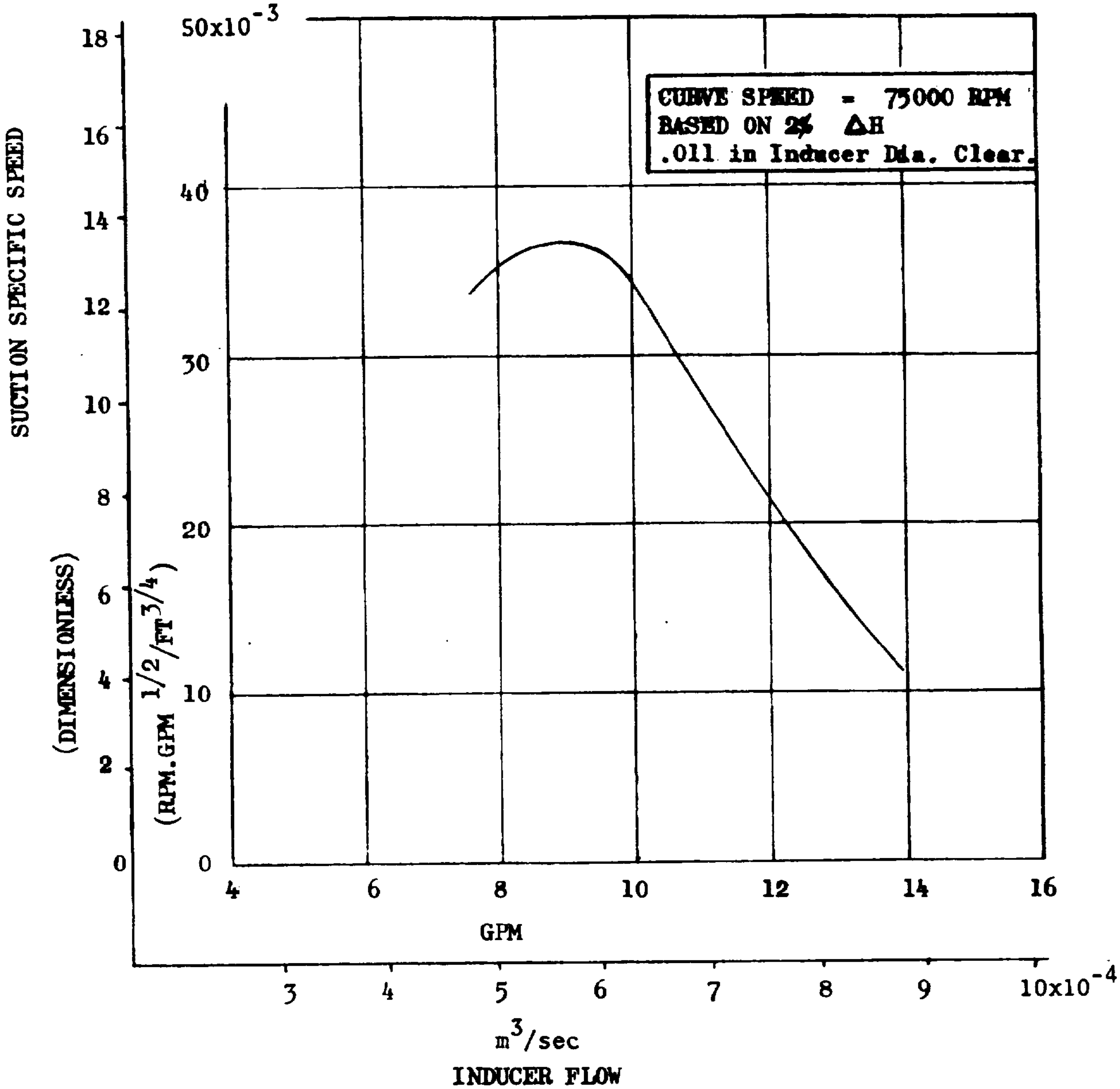


Figure 30. 12 gpm Rotating Pump Suction Specific Speed in Freon 12

ASSEMBLY AND INSTALLATION FOR FLUORINE TESTING

In reassembling the pump for liquid fluorine testing a majority of the components involved in the freon tests were reused. The carbon nose primary seal was replaced with one incorporating an aluminum oxide nose. New intermediate seal rings and new front and rear bearings were also installed. The inlet housing and volute with the Al_2O_3 wear ring coatings from the last freon test series were used resulting in the following diametral clearances

Inducer tip clearance = .011 in (2.8×10^{-4} m)
Impeller front wear ring clearance = .006 in (1.5×10^{-4} m)
Impeller rear wear ring clearance = .0031 in ($.79 \times 10^{-4}$ m)

A balance check of the rotor with the new bearings revealed no unbalance in the plane of the impeller; a minor correction was made in the plane of the turbine disc to reduce the unbalance to the machine accuracy limit of 0.01 gr. in. ($.0249 \times 10^{-6}$ Nm).

Before assembly all parts were subjected to ultrasonic cleaning followed by flushing with freon. Standard cleanliness standards established for liquid oxygen service were observed during assembly. In addition, stringent procedures were observed to ensure that the internal surfaces of the pump were maintained free of moisture. As a final precaution the pump was dried after assembly for 8 hours at 250°F (395°K) and the pump was shipped in sealed polyethylene bags filled with dry gaseous helium.

At the test site installation was initiated by applying a 10 psig (6,900 N/m^2) GHe pressure to the pump cavity and to the seals through the intermediate seal purge port. This positive pressure was maintained throughout installation and between tests when the pump was not pressurized with another medium.

When metal parts are exposed to fluorine a thin film of fluoride is formed on the exposed surfaces which provides a certain amount of protection against burning and explosion. The pump components were not individually passivated before assembly because the fluoride film is hygroscopic and the parts would have to be maintained in a moisture-free atmosphere, making assembly procedures cumbersome. Passivation was accomplished in the assembled state, in the test facility by the following procedure:

1. Gaseous fluorine was introduced into the pump and maintained at 30 psig (2.07×10^5 N/m^2) for one hour.
2. The pump shaft was rotated to expose the bearing surfaces which were in contact above.
3. The GF_2 pressure was increased to 60 psig (4.14×10^5 N/m^2) and maintained for one hour.
4. The pump shaft was rotated.
5. The GF_2 pressure was increased to 100 psig and maintained for one hour.

Further passivation before dynamic operation was realized during pre test childown when the pump was filled with two phase and liquid fluorine for approximately a half an hour.

FLUORINE TEST FACILITY

Facility Description

Testing of the centrifugal pump with liquid fluorine was conducted in the Alfa stand of Rocketdyne's Propulsion Research Facility at Santa Susana. A schematic of the facility configuration is presented in Fig. 31 . The pump installation is shown in Fig. 32, 33 and 34.

The liquid fluorine pumped was recirculated in a closed loop system consisting of a 43-gallon run tank, inlet line. the pump and return line. A bleed line was provided from the bearing cavity to downstream of the discharge throttle valve. All lines and valves carrying liquid fluorine were LN_2 jacketed.

FOLDOUT FRAME 1

FOLDOUT FRAME 2

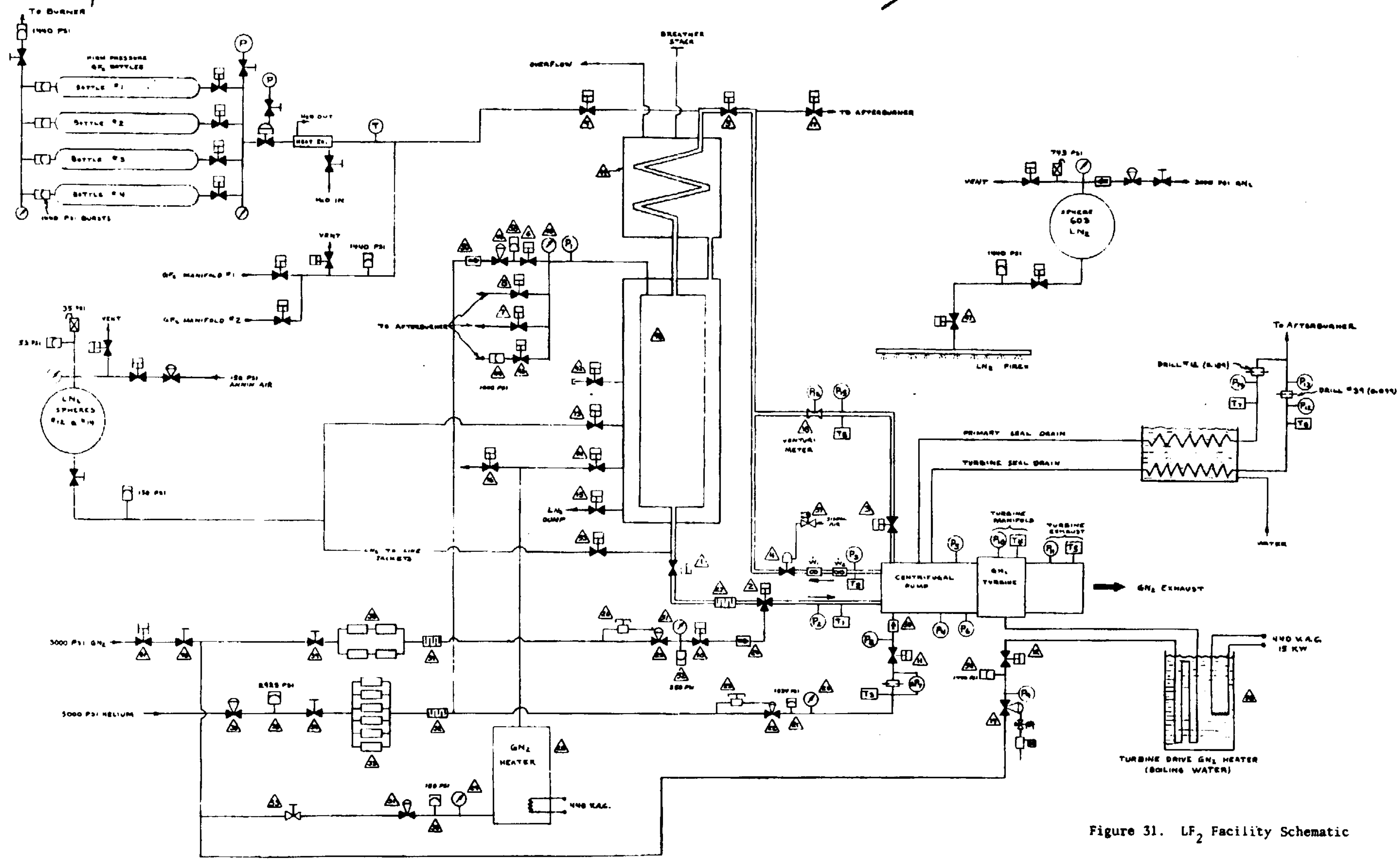
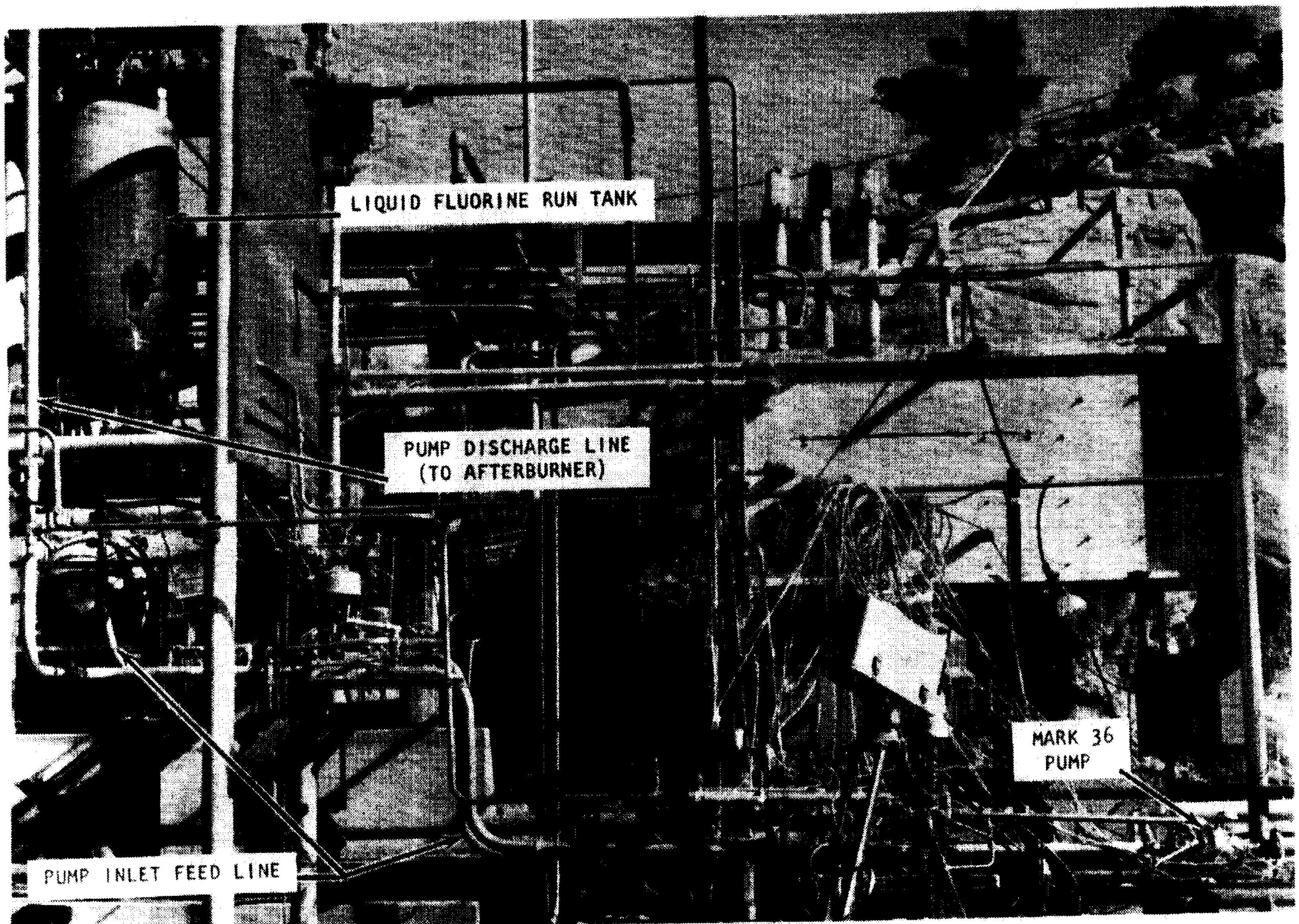


Figure 31. LF_2 Facility Schematic

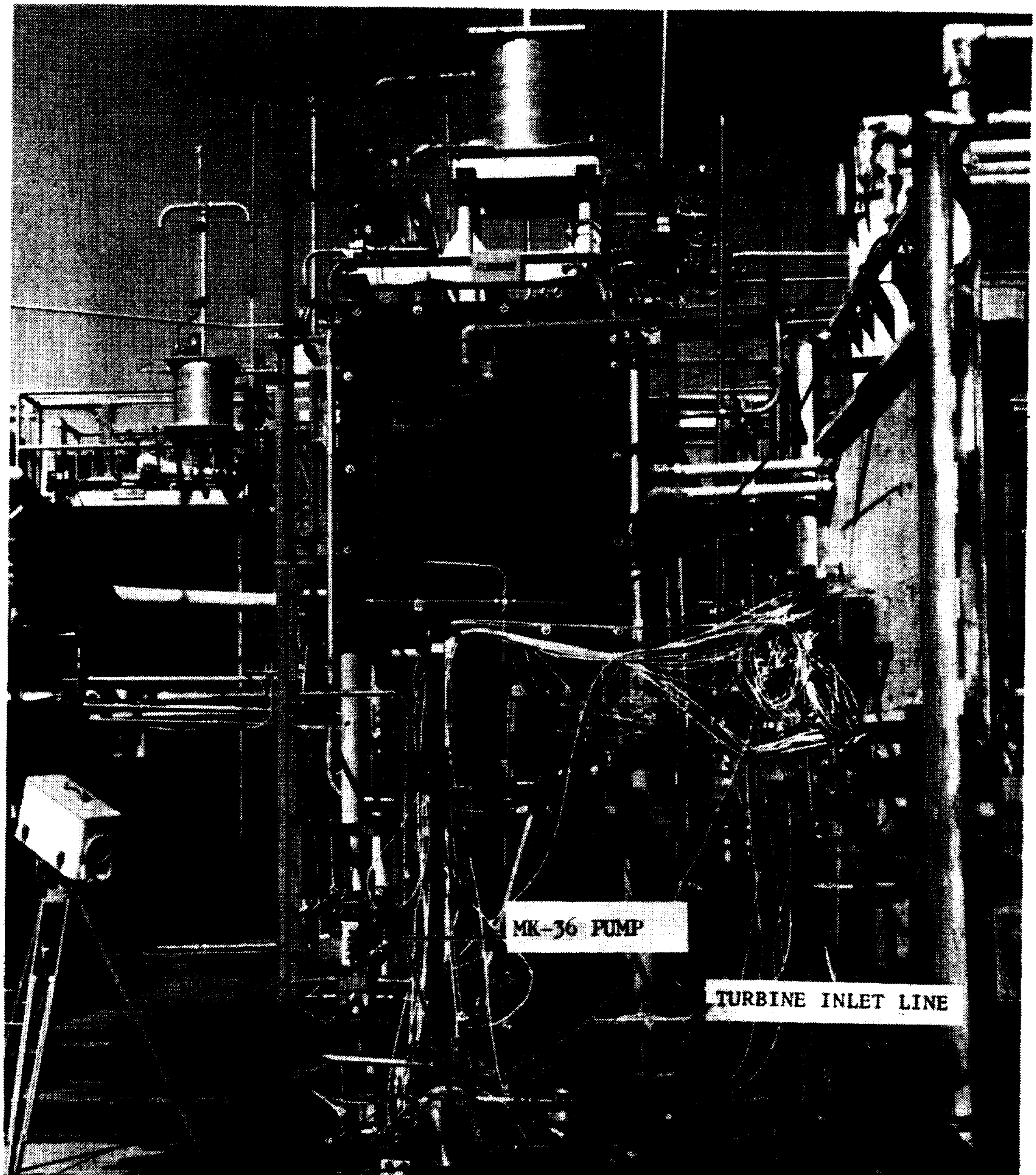
R-8494-2

71/72



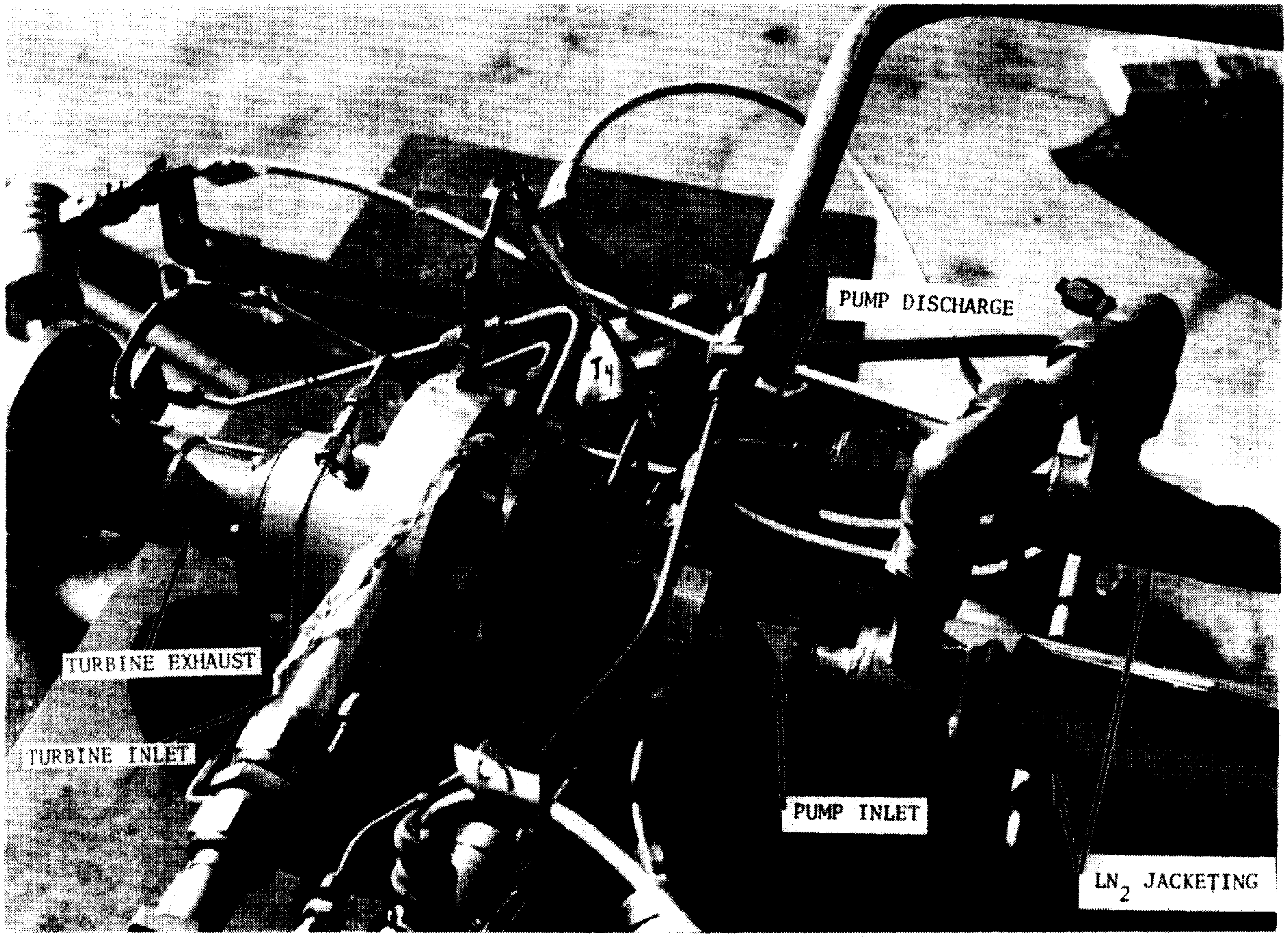
5AJ24-3/4/71-S1C

Figure 32. LF_2 Test Facility



5AL24-3/4/71-S1F

Figure 33. LF_2 Test Facility



5AL24-3/4/71-S1D

Figure 34. Mark 36 Pump Installation in LF_2 Test Facility

Fluorine was stored in the test area in the gaseous state in bottles and condensed into the run tank for each test day. At the conclusion of the day's testing the fluorine was gasified by heating and returned to the bottles. Emergency provisions included a liquid nitrogen firex system and a high pressure GN_2 source into the pump inlet which would be used to purge out liquid fluorine from the pump in case of a chemical reaction.

Instrumentation

A list of instrumentation used during fluorine testing with ranges, types of transducers and method of recording is included in Table VII . The code letters noted can be used to locate the instrumentation on Fig. 31.

PUMP TESTS WITH LIQUID FLUORINE

The objective of the liquid fluorine tests was to demonstrate operating capability and determine the pump hydrodynamic performance in liquid fluorine. Steady state operation to 42,000 rpm was achieved with transient speed up to 104,000 rpm. Testing was curtailed due to funding limitations when high rotor vibration and bearing ball wear were encountered.

Performance Tests

The pump and turbine data and seal leakage data obtained during the liquid fluorine tests are presented in Tables VIII and IX. The reduced data at actual test speed and scaled to the design speed of 75,000 rpm are given in Table X.

TABLE VII. LIQUID FLUORINE TEST INSTRUMENTATION LIST

Parameter	Code	Range	Transducer	Recording
LF_2 Tank Pressure	P_1	0-1000	Taber	D
Pump Inlet Pressure	P_2	0-100	Taber	D,B,T,0*
Pump Discharge Pressure	P_3	0-2000	Taber	D,B,T,0
Impeller Front Shroud Pressure	P_4	0-2000	Taber	B
Impeller Rear Shroud Pressure	P_5	0-2000	Taber	B
Balance Cavity Pressure	P_6	0-500	Taber	B
Intermed. Seal Purge Orifice ΔP	P_7	0-20	CEC ΔP	D,B
Intermed. Seal Purge Pressure	P_8	0-200	Taber	D,B*
Turbine Line Pressure	P_9	0-200	Taber	B
Turbine Manifold Pressure	P_{10}	0-200	Taber	D,B
Turbine Exhaust Pressure	P_{11}	0-20	Taber	B
Turbine Seal Orifice U/S Press.	P_{12}	0-50	Taber	B
Turbine Seal Orifice D/S Press.	P_{13}	0-50	Taber	B
Primary Seal Orifice U/S Press.	P_{14}	0-50	Taber	B
Bearing Cavity Pressure	P_{15}	0-500	Taber	B,0*
Bleed Flow Venturi Pressure	P_{16}	0-500	Taber	B
Pump Inlet Temperature	T_1	(-325 to -275 F)	Rosemount	D,B*
Pump Discharge Temperature	T_2	(-325 to -275 F)	Rosemount	D,B*
Seal Purge Temperature	T_3	(-320 to+100 F)	I/C T.C.	B
Turbine Inlet Temperature	T_4	0 to 100 F	I/C T.C.	B
Turbine Exhaust Temperature	T_5	(-320 to +100 F)	I/C T.C.	B
Turbine Seal Drain Temperature	T_6	(-320 to +100 F)	I/C T.C.	B
Primary Seal Drain Temperature	T_7	(-320 to +100 F)	I/C T.C.	B*
Bearing Cavity Bleed Temp.	T_8	(-320 to +100 F)	I/C T.C.	D,B
Pump LF_2 Flowrate	W_1	0 to 20 gpm	(1-45 Duplex F/M)	D,B,T,0
Pump LF_2 Flowrate	W_2	0 to 20 gpm	(1-45 Duplex F/M)	D,B,T,0
Bearing Cavity Bleed Flow	W_3	0 to 1 gpm	.098 Venturi	D,B
Bently (Speed & Defl.)	BV_1	0 to 15V	Bently	S,D,B,T,0*

D = DIGR, B = Beckman, T = Tape, 0 = Oscillograph, S = Oscilloscope

*Comparator Cut Circuits

TABLE VIII. MARK 36 PUMP LF_2 TEST DATA

RUN NO	DATE	TIME SLICE (BECKMAN) (SEC)	PUMP SPEED (RPM)	PUMP FLOW (GPM)	PUMP INLET PR. (PSIG)	PUMP DISCH. PR. (PSIG)	PUMP INLET TEMP (°F)	PUMP DISCHARGE TEMP. (°F)	IMPELLER FRONT SHROUD PR. (PSIG)	IMPELLER REAR SHROUD PR. (PSIG)	BRG. CAVITY BLEED PR. (PSIG)
01	3-5-71	-	-	-	-	-	-	-	-	-	-
02	3-6-71	170	16,500	6.29	61.5	71.0	-311.8	-307.9	65.4	63.3	48.5
		209	20,400	2.66	66.0	203.8	-311.5	-309.1	146.3	134.2	75.8
		219	17,700	5.19	63.4	119.8	-311.6	-308.4	87.8	84.1	61.8
		232	19,600	3.79	65.0	173.8	-311.4	-309.0	123.3	116.6	71.1
		258	21,200	2.65	66.3	211.4	-311.1	-308.7	152.9	134.5	77.0
		275	25,700	0	68.5	294.4	-310.2	-308.0	242.7	208.8	92.3
		298	23,200	1.38	68.0	252.2	-309.1	-305.8	193.0	170.6	85.6
		318	23,900	.83	68.5	265.8	-309.5	-305.6	210.3	180.9	88.3
03	3-6-71	7	33,600	4.68	61.6	428.4	-308.0	-303.8	270.9	258.1	94.7
		13	42,000	6.44	58.7	634.4	-310.9	-304.5	382.0	364.8	109.9
		17	41,100	6.35	58.9	605.6	-311.6	-306.4	364.4	344.9	109.7
		26	39,200	6.13	58.5	554.9	-312.2	-307.8	333.5	315.5	95.3
		37	37,800	5.95	58.5	522.0	-312.4	-308.5	315.6	296.5	90.3
		44	36,600	5.64	58.3	474.0	-312.4	-308.8	288.5	271.6	87.7
04	3-20-71	2.3	-	4.65	51.0	1209.9	-306.0	-303.4	807.0	767.0	165.8
05	3-20-71	2.1	-	7.4	46.9	630.1	-306.5	-305.3	370.4	382.6	105.2

TABLE IX. MARK 36 PUMP LF_2 TEST TURBINE AND SEAL DATA

DRIVE TURBINE RAW DATA AT TEST SPEED					DYNAMIC SEAL LEAKAGE DATA									
					PRIMARY SEAL			INTERMEDIATE SEAL				TURBINE SEAL		
RUN NO.	TIME SINCE [illegible] (SEC)	TURBINE MANIFOLD PR. (PS[illegible])	TURBINE EXHAUST PR. (PSIG)	TURBINE INLET TOTAL TEMP. (°F)	DRAIN PRESS (PSIG)	DRAIN TEMP. (°F)	LEAKAGE (FREON 12) (SCFM)	PURGE PRESS U/S ORIF (PSIG)	PURGE PRESS D/S ORIF (PSIG)	PURGE TEMP. U/S ORIF (°F)	PURGE TOTAL FLOW (GH) ([illegible])	DRAIN PRESS ([illegible])	DRAIN TEMP. ([illegible])	LEAKAGE (GN_2) ([illegible])
0[illegible]	-	-	-	-	-	-	-	-	-	-	-	-	-	-
0[illegible]	170	9.08	.31	10.0	0.32	53	0	56.2	49.0	52	3.53	.88	54	.28
	209	9.37	.32	8.0	0.98	53	.99	56.4	49.3	52	3.52	1.00	53	.28
	219	8.79	.31	7.5	1.00	52	.90	56.3	48.9	52	3.59	1.08	53	.30
	232	8.84	.31	7.0	0.86	51	.78	56.3	48.9	52	3.59	1.07	52	.32
	258	9.50	.33	5.9	0.90	50	.84	56.3	48.9	52	3.59	1.07	51	.32
	275	9.60	.38	5.4	1.06	50	1.07	56.4	49.1	52	3.57	1.10	51	.31
	298	9.20	.35	5.0	0.74	50	.53	56.4	49.0	52	3.59	1.02	51	.30
	318	9.50	.37	4.7	0.79	50	.66	56.4	49.1	53	3.56	1.01	52	.30
03	7	27.0	.9	22.7	.95	48	.90	57.9	50.7	41	3.69	1.38	48	.42
	13	46.8	1.9	14.1	1.52	48	1.67	57.9	50.6	41	3.71	1.50	48	.46
	17	44.7	1.8	9.9	1.57	48	1.50	57.8	49.3	41	4.00	1.64	49	.49
	26	41.8	1.6	5.3	1.30	48	1.42	57.8	48.2	42	4.25	1.68	49	.51
	37	46.2	1.6	3.0	1.16	48	1.26	57.3	47.3	42	4.31	1.64	49	.51
	44	-	-	2.2	1.32	47	1.44	57.6	46.9	43	4.48	1.73	49	.53
04	2.3	77.3	4.0	95.5	2.9	66	2.80	56.2	55.7	72	1.0	2.7	63	.7
05	2.1	51.9	2.3	94.7	9.2	69	6.6	77.8	63.5	75	6.4	20.3	70	4.6

TABLE X. MARK 36 PUMP LF_2 TEST REDUCED DATA

	Reduced Pump Data at Test Speed							Reduced Pump Data Scaled to 75,000 RPM			
RUN NO	TIME SLICE (BECKMAN) (SEC)	PUMP SPEED (RPM)	PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE TORQUE (IN-LB)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)	PUMP FLOW (GPM)	DEV. PUMP HEAD (FT)	TURBINE BHP (HP)	OVERALL PUMP EFFICIENCY (%)
01	-	-	-	-	-	-	-	-	-	-	-
02	170	16,500	6.29	24.9	1.90	.50	12	28.6	515	46.7	12
	209	20,400	2.66	207.1	1.92	.62	35	9.8	2799	30.9	35
	219	17,700	5.19	91.4	1.83	.51	36	22.0	1640	39.1	36
	232	19,600	3.79	165.9	1.81	.56	44	14.5	2428	31.6	44
	258	21,200	2.65	218.2	1.93	.65	35	9.4	2730	28.8	35
	275	25,700	0	337.5	1.90	.78	0	0	2875	19.3	0
	298	23,200	1.38	277.2	1.86	.68	22	4.5	2897	23.1	22
	318	23,900	.83	296.9	1.89	.72	13	2.6	2924	22.6	13
03	7	33,600	4.68	1006	5.14	6.59	37	10.5	2794	30.5	37
	13	42,000	6.44	1012	8.30	6.81	40	11.5	2811	31.5	40
	17	41,100	6.35	999	7.90	6.78	40	11.6	2776	31.4	40
	26	39,200	6.13	994	8.10	7.59	36	11.7	2762	35.1	36
	37	37,800	5.95	997	8.40	8.50	32	11.8	2769	39.4	32
	44	36,600	5.64	953	-	-	-	11.6	2647	-	-
04	2.3	-	4.65	1760	-	-	-	-	-	-	-
05	2.1	-	7.4	894	-	-	-	-	-	-	-

Tests #01 & 02. The purpose of the initial tests was threefold: Check instrumentation and general test procedures; demonstrate the capability of operating the pump in LF_2 environment; and obtain H-Q data at approximately 25 percent of design speed. Since the target speed was below the shaft first critical, the start procedure was to open the turbine inlet valve in small increments until the desired speed level is achieved, after which the throttle valve in the pump discharge was exercised to obtain data between 10 percent and 120 percent of nominal Q/N. On Test #01 successful operation with the pump was achieved; however, the test was prematurely terminated after 16 seconds because of a lack of speed display. The instrumentation discrepancy was corrected, and Test #02 was conducted for a planned 314 seconds duration, during which H-Q data over a very wide range of flowrates was obtained at approximately 25 percent of the design speed. Review of the data showed that the pump was operating in a satisfactory manner. Shaft deflections as measured by the Bently transducer at the tip of the turbine disc were approximately .001 in. (0.254×10^{-4} m), indicating that the rotor was balanced and assembled properly. Seal leakage rates were low. No temperature or pressure spikes or fires were evident, indicating that there was no adverse chemical reaction in the pump.

Test # 03. The objective on Test #03 was to obtain H-Q data at 60 percent of the design speed (45,000 rpm). Since this speed is over the first critical of the shaft, the start procedure was altered to effect a rapid acceleration to the target speed by pre-setting the turbine power source pressure and opening an inlet valve. Satisfactory operation up to a speed level of 42,000 rpm was achieved with the pump discharge throttle valve set at nominal Q/N. Before the throttle valve could be adjusted to other Q/N values, a decay in speed occurred which brought the operating level near the first shaft critical. As a result the shaft vibration increased to .030 inch (7.6×10^{-4} m) peak-to-peak and the test was terminated on the basis of the Bently signal displayed on the oscilloscope. The test duration was 43 seconds.

Examination of the data revealed that concurrently with the speed decay the turbine inlet pressure was increasing. However, other parameters in the turbine propellant system, such as valve pressure drops indicated that the GN_2 flow to the turbine was decreasing. This pointed to a reduction in the turbine nozzle area. When visual inspection of the turbine disclosed no foreign matter in the nozzle, it was concluded that the nozzle area was reduced by an ice film formed from the moisture in the drive gas. (Turbine inlet temperature was substantially below the freezing point.) Corroborating evidence was found in the form of water traces in the bottom of the manifold. To prevent a recurrence on subsequent tests, the heat exchanger shown in Fig. 31 immediately upstream of the turbine was added to the facility, which raised the turbine inlet temperature substantially above the freezing point.

Analysis of the rotordynamic data yielded the shaft deflection versus speed relationship illustrated in Fig. 35. Thus deflections were normal below the first critical and appeared to subside as the speed was increased above the immediate vicinity of the critical speed. This is normal rotordynamic behaviour.

Approximately 15 seconds after Test #03 was terminated, during the procedure for securing the test stand, the turbine inlet valve was reopened by an operator error and the pump speed reached 92,000 rpm for approximately 3 seconds. Although the shaft vibration characteristics during the overspeed could not be evaluated because the high frequency tape was already turned off, shaft torque checks, seal leakage measurements, and dimensional checks after the test showed that the pump did not sustain any gross damage.

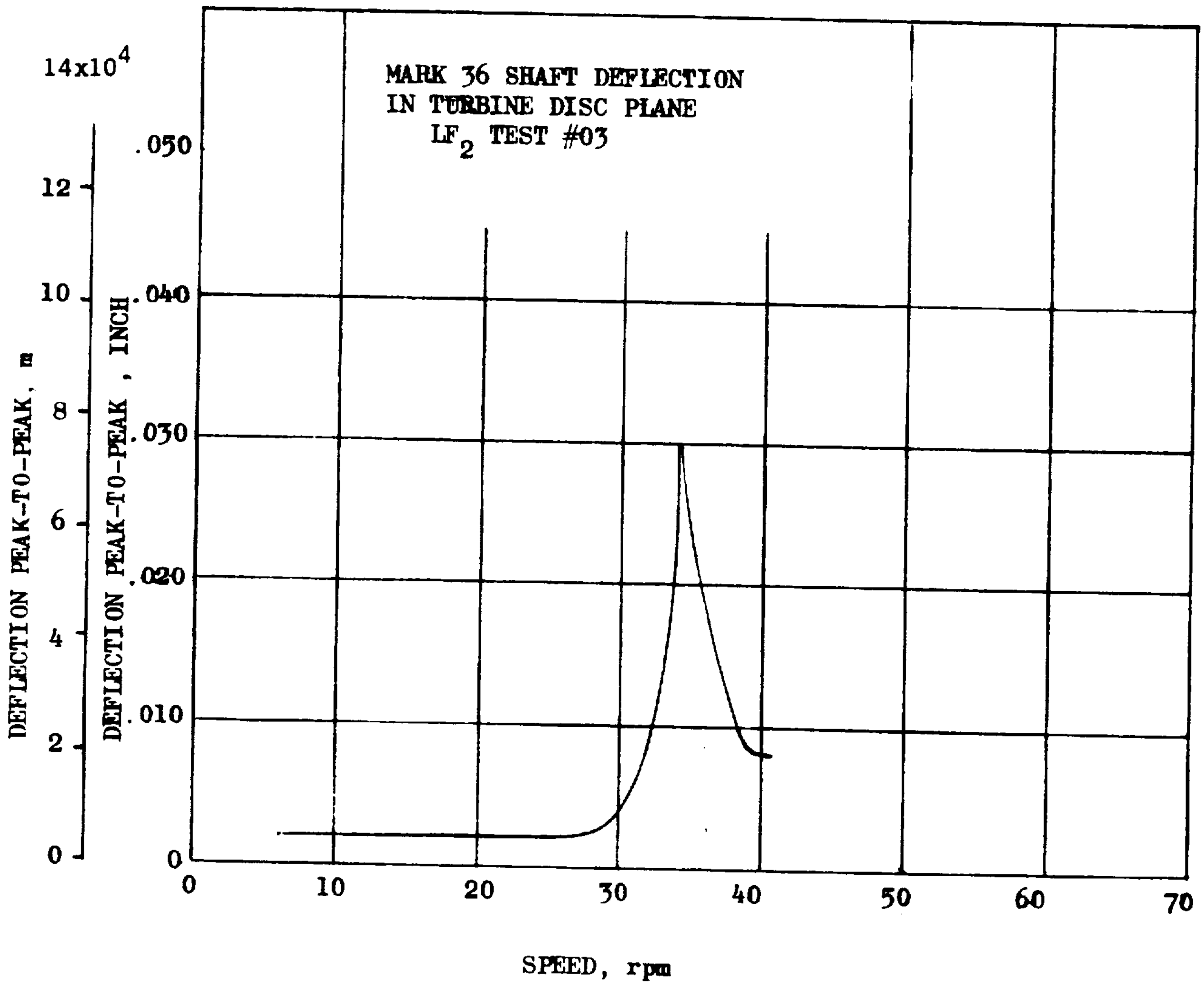


Figure 35. Mark 36 Shaft Deflection, LF_2 Test No. 03

Tests #04 & 05. Since the H-Q characteristics obtained on Tests #02 & #03 appeared to correlate with data obtained in freon, and due to funding limits, further H-Q testing was deleted from the program and the objective of Tests #04 and #05 was to obtain suction performance data at the design speed of 75,000 rpm. Thus plans for Test #04 were to reach a speed of 75,000 rpm at nominal Q/N and with the pump inlet pressure set sufficiently high to ensure a non-cavitating condition, then reduce the inlet pressure until the developed head decreased by 10 percent.

Normal start characteristics were evident on Test #04 for approximately 0.8 seconds, during which the speed stabilized at 60,000 rpm, and the discharge pressure and delivered pump flowrate reached 1280 psig and 5 gpm, respectively. At this point a sudden increase in speed to 102,000 rpm was experienced, attendant with a drop in flowrate and discharge pressure. The test was terminated 3.5 seconds after start when the turbine disc rubbed on the Bently pickup and the speed count was lost. The speed indication recorded on the basis of the Bently frequency was erroneous and as a result the overspeed trip set at 80,000 rpm did not actuate.

Since the temperature and pressure measurement in the inlet and discharge indicated a good quality liquid with more than adequate NPSH based on the freon tests, the behaviour of the pump on Test #04 cannot be attributed to normal cavitation. It is believed that a vapor bubble was introduced into the pump from the inlet line, which unloaded the pump and resulted in the overspeed. An analysis of the facility inlet system was made in an attempt to locate the source of the bubble. The flowrate was integrated from start to the point where the sudden change in parameters occurred. The results indicated that for the bubble to arrive at the inducer at 0.8 seconds after start, it had to be located 1.5 ft (0.46 m) upstream. Examination of the facility revealed two elements located in that area: the pump inlet temperature and pressure ports, 14 inches and 17 inches upstream of the pump, respectively. Other potential sources were more than 10 feet (3 m) further upstream. The temperature probe protrudes into the inlet line and therefore does not represent a natural gas trap. The inlet pressure port included approximately 3 inches (.076 m) of 1/4 inch (.0063 m) uninsulated tubing to the Taber transducer. Although this is considered "close-coupled" by normal standards, a vapor bubble could have been belched into the inlet from this line on start, when the inlet pressure decreased due to line pressure drop. Since normal precautions such as chilling the lines with LN_2 and overboard bleeding of liquid fluorine were taken to condition the propellant in the facility lines, the experience on Test #04 indicates that this pump is substantially more sensitive to a vapor bubble in the system than larger size cryogenic pumps.

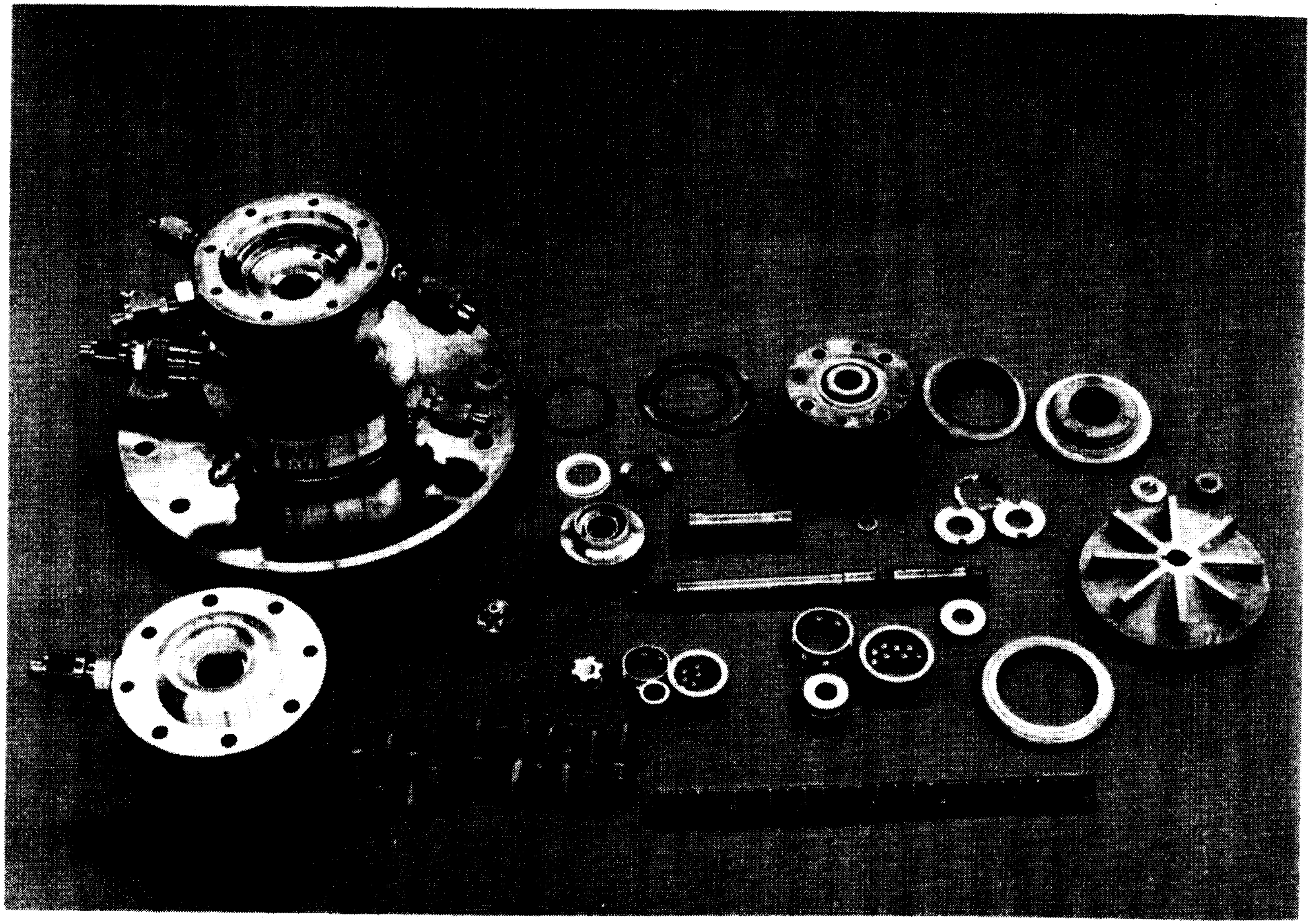
Analysis of the Bently data revealed that shaft deflections remained under .010 inch (2.54×10^{-4} m) up to 100,000 rpm, but suddenly increased at 104,000 rpm to a level sufficiently high to result in rubbing at the Bently tip which was set at a gap of .035 inch (8.9×10^{-4} m).

Another attempt was made to obtain suction performance data on Test #05, but the test was terminated after 10 seconds when problems were encountered in obtaining a reliable speed count.

Analysis of the Bently data of Test #05 revealed a significant increase in the vibration level of the rotor ($>$ .020 inch (5.1×10^{-4} m) peak-to-peak). These high deflections represented a potential hazard to the hardware and made the electronic analysis of the Bently output for an accurate speed count extremely difficult. Because of this and because of funding limitations, testing was discontinued and the pump was disassembled.

Hardware Condition After LF_2 Testing

Disassembly of the pump disclosed that the hardware was in general in good condition. No trace of fluorine burning was evident on components enclosed in the pump and exposed to liquid fluorine. The pump components after disassembly are shown in Figure 36.



1XY55-4/5/71-C1H

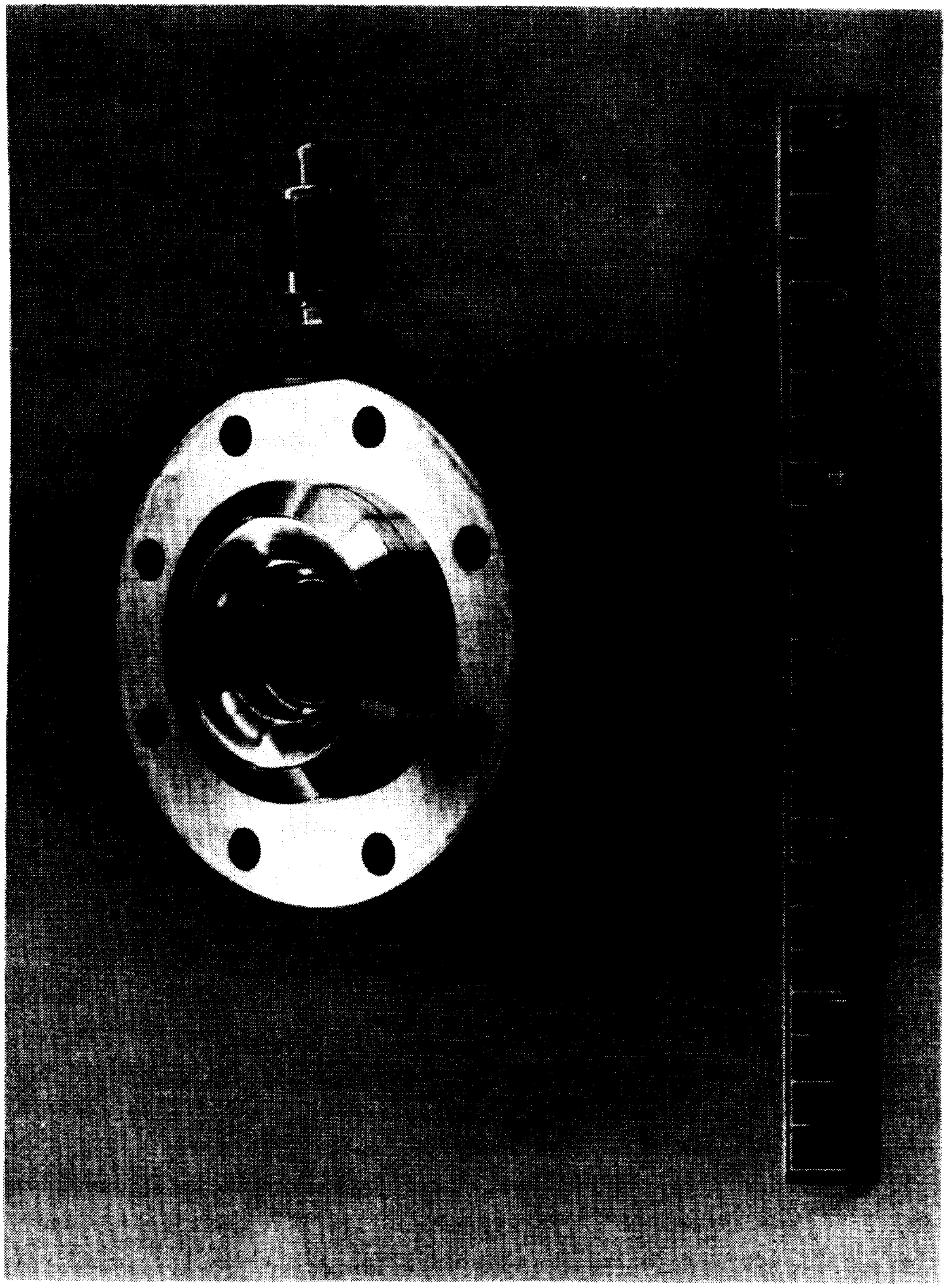
Figure 36. Mark 36 Pump Components After LF_2 Testing

Inlet and Volute Housings. Both parts were in excellent condition, without any sign of fluorine reaction in the enclosed areas. There was evidence of fluorine leakage past the flange seals between the volute and the primary seal. As a result corrosion was evident on portions of the volute which were exposed to moisture from the turbine drive gas or from the atmosphere. The aluminum oxide coatings which were added to the wear ring surfaces showed evidence of minor contact, but were otherwise in excellent condition (Ref. Fig. 37 and 38).

Inducer and Impeller. To obtain good hydrodynamic performance the inducer and impeller required reasonably sharp vane leading edges. One question to be answered by the fluorine test program was whether these salient edges would be attacked by liquid fluorine. Neither the inducer nor the impeller evidenced any sign of reaction or erosion which testifies to the excellent fluorine compatibility of the Inco 718 material chosen for these parts (Fig. 39).

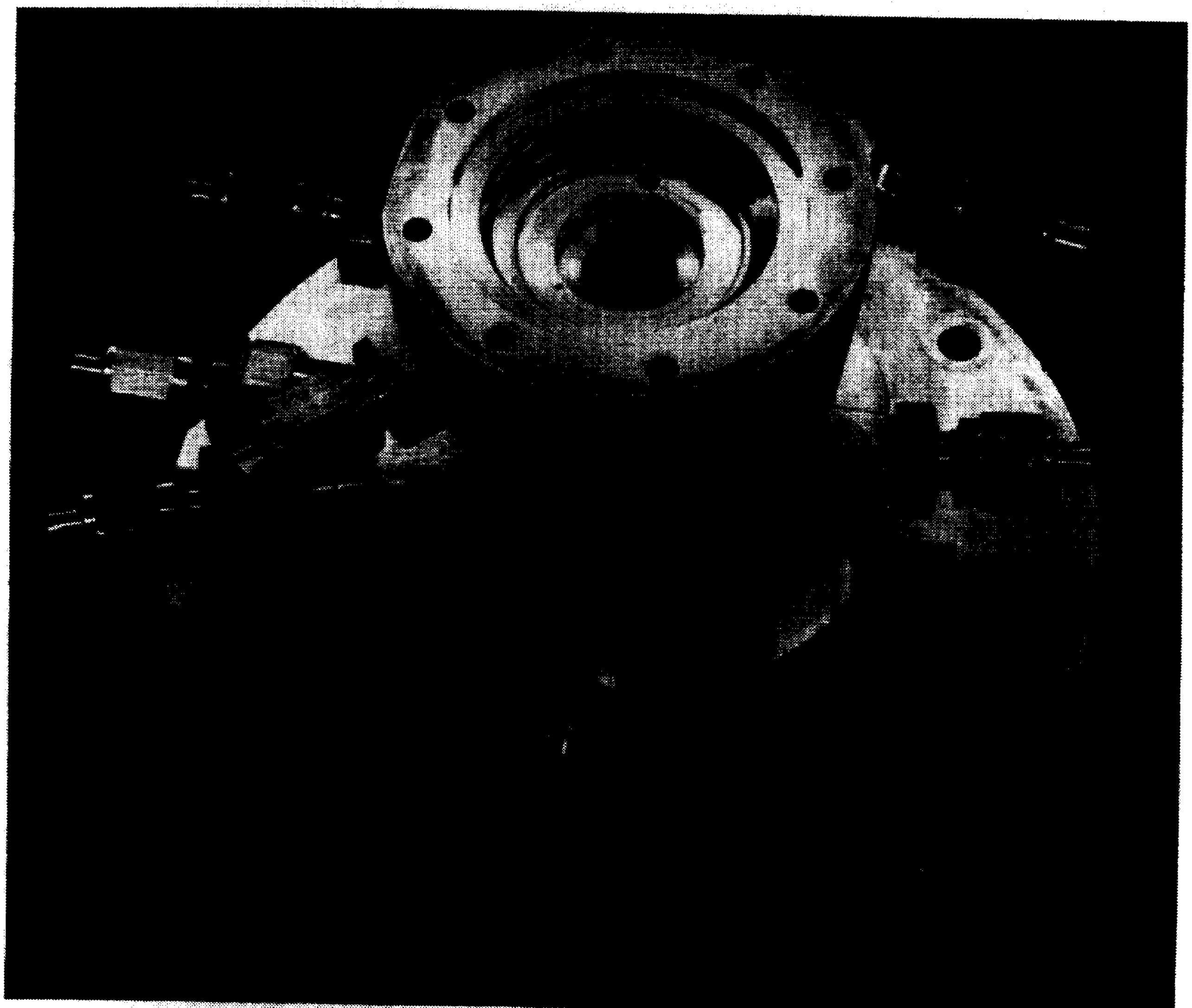
The aluminum oxide coating came off the impeller front wear ring over approximately 30 percent of the circumference and on the rear wear ring over about 1/8 inch (0.003 m) of the circumference. Since contact at the wear rings was very light the failure of the coating is attributed to two potential sources: Excessive centrifugal stresses when the pump was oversped or a differential thermal contraction problem at LF_2 temperature. Since earlier this same impeller successfully sustained chilling to LN_2 temperature without flaking of the Al_2O_3 coating, the excessive speed is a likelier suspect, but additional testing would have to be done to substantiate this.

Seals. Figure 40 shows the shaft dynamic seals after turbopump disassembly. The aluminum oxide rubbing surfaces on the primary seal nose and mating ring were in good condition. The wear on the mating ring was .0002 inch (5.1×10^{-6} m). The nose on the seal housing wore unevenly with approximately .004 inch (1.02×10^{-4} m) wear on one side and practically no wear at a point diametrically opposed. There was evidence of fluorine leaking past the flange seals and combining with moisture from the turbine drive gas to corrode the primary seal housing. Pressure check of the primary seal bellows showed no cracks.



1XY55-4/5/71-C1F

Figure 37. Mark 36 Inlet Housing After LF_2 Tests



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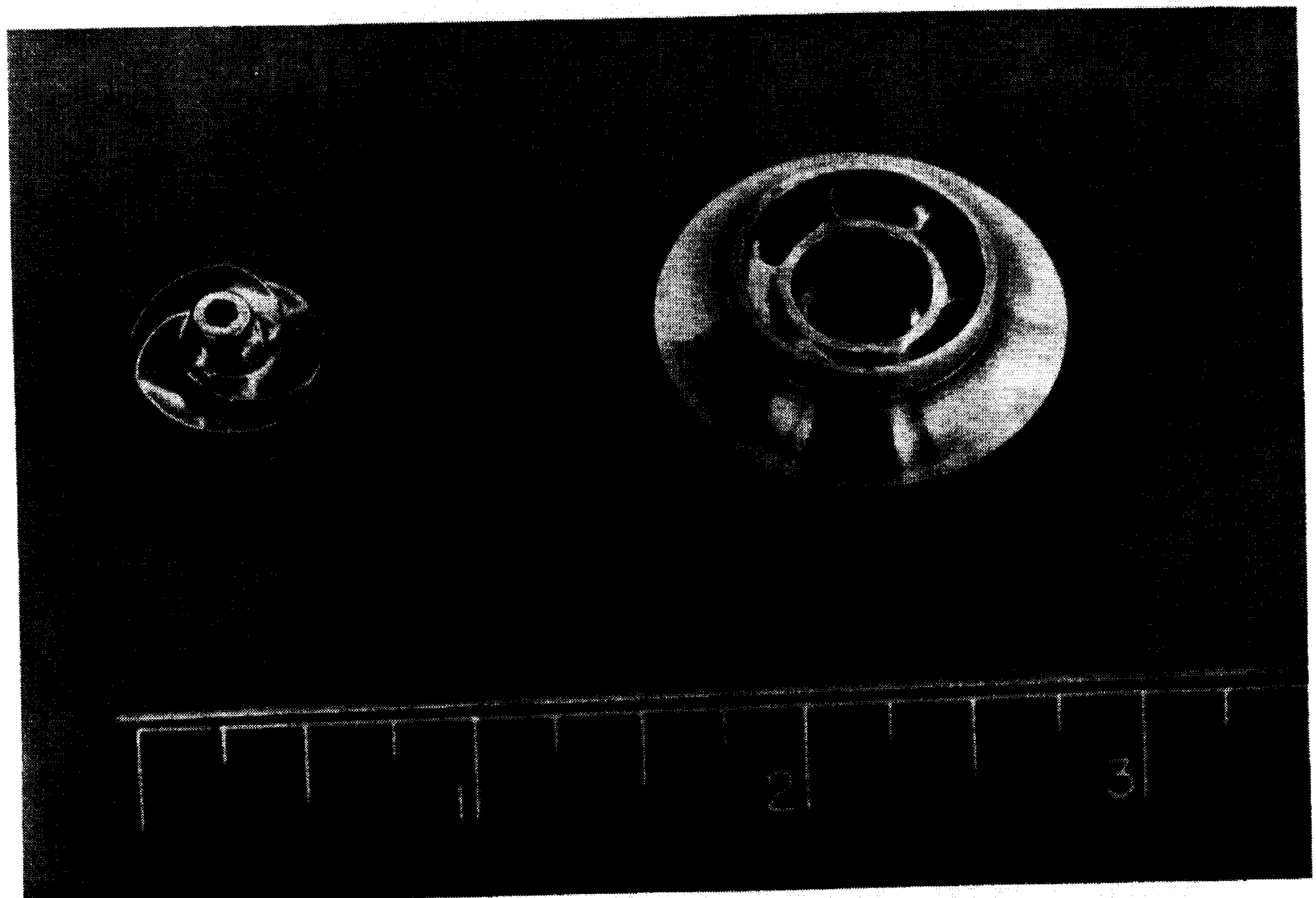
1XY55-4/5/71-C1G

Figure 38. Mark 36 Volute After LF_2 Testing



1XY55-4/5/71-C1E

Figure 39. Mark 36 Impeller and Inducer After LF_2 Testing

1XY55-4/5/71-C1C

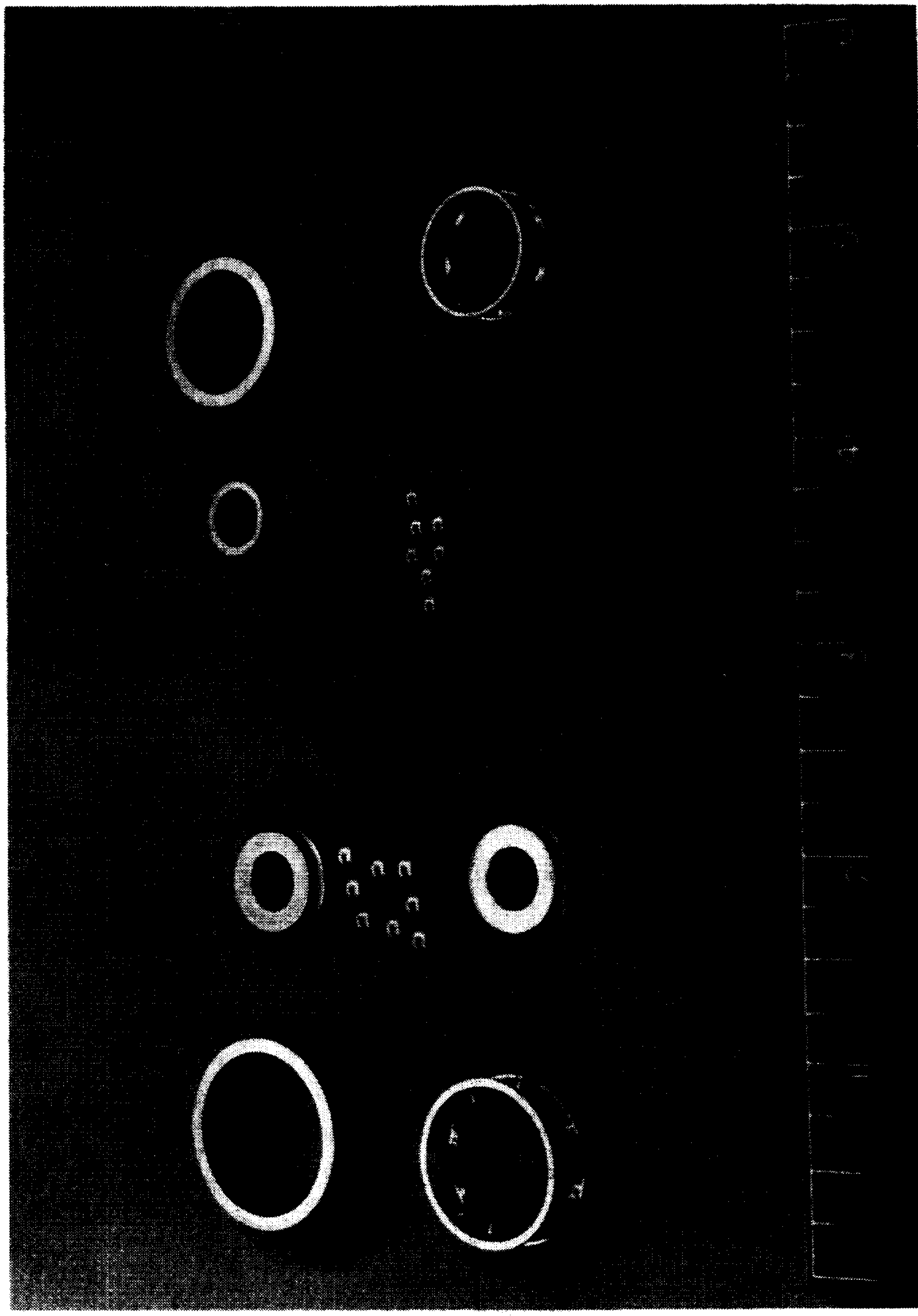
Figure 40. Mark 36 Seals After LF_2 Testing

The intermediate seal ring located on the turbine side of the seal cracked in half through the anti-rotation pin slots. This is believed to have resulted from the high shaft deflections which were encountered at the end of Test #04 and throughout Test #05. The other ring was intact although it too showed deep indentations at the pin slots.

The turbine seal was in excellent condition. Leak checking the bellows revealed no cracks, corroborating previous data obtained during freon cavitation tests which showed that the corrugated damper added to the seal eliminated stick-slip mode vibrations and the ensuing bellows failures. The carbon nose wear was negligible.

Bearings. Visual and dimensional inspection of the bearings revealed a significant cage pocket and ball wear problem (Fig. 40 and 42). The ball pockets on the cages of both bearings were worn noticeably; part of the cage material was actually pushed out of the pockets and was attached to the sides of the pockets in a thin sliver form. No pitting or other surface failure indications were present on the balls or races, although the rear (turbine end) bearing balls in particular took on a dull appearance. The bearing ball diameters were measured and the results are presented in Table XI. The front bearing balls wore approximately .0004 inch (1.1×10^{-5} m) whereas the rear bearing balls wore .0013 inch (3.3×10^{-5} m).

Turbine. Evidence of minor fluoride corrosion was visible on the turbine disc and to a more significant degree at the bottom of the manifold and turbine exhaust tunnel, where moisture from the turbine gas or from the atmosphere would tend to collect. A small amount of material was removed from the tips of blades where they rubbed into the nozzle, evidently when the large deflections on Test #04 occurred. No other deleterious effects from the high speed excursions were present.



1XY55-4/5/71-C1D

Figure 41. Mark 36 Bearings After LF_2 Testing

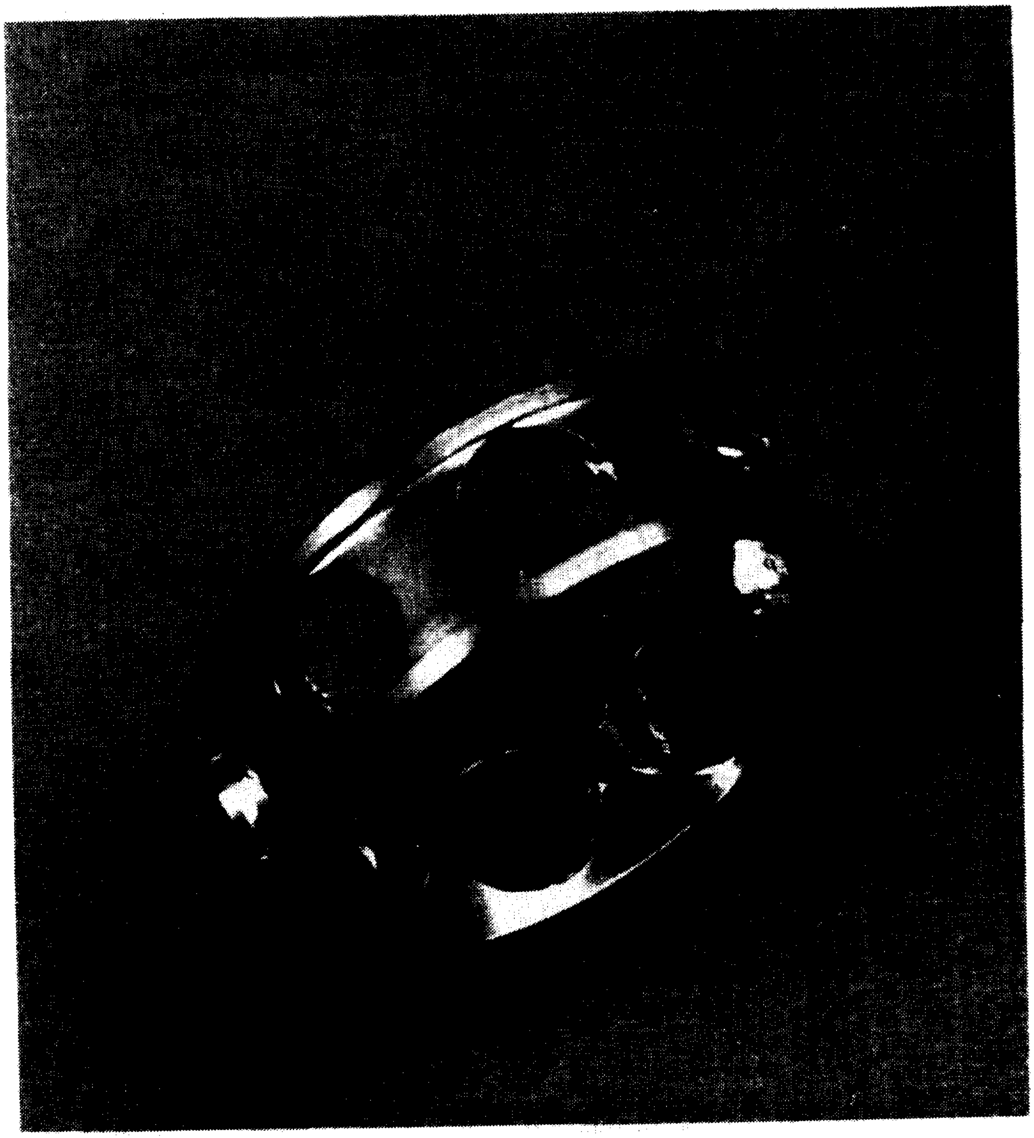


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Figure 42. Mark 36 Rear Bearing Cage After LF_2 Testing

TABLE XI. MARK 36 BEARING BALL DIAMETERS

FRONT BEARING BALL DIAMETERS (in.)	
BEFORE LF_2 TESTING	AFTER LF_2 TESTING
.156458	.155970
.156448	.156000
.156448	.155970
.156448	.155990
.156438	.155990
.156448	.155990
.156438	.155980
AVERAGE .156447	AVERAGE .155984

REAR BEARING BALL DIAMETERS (in.)	
BEFORE LF_2 TESTING	AFTER LF_2 TESTING
.187540	.186260
.187520	.186240
.187520	.186260
.187520	.186260
.187520	.186250
.187510	.186250
.187510	.186260
.187530	.186240
AVERAGE .187521	AVERAGE .186253

Shaft. The remaining components of the pump including the shaft and bearing spacer were in good condition (Ref. Fig. 43). Indications were present on the shaft that radial rubbing between it and the turbine seal housing had taken place.

Hydrodynamic Performance in LF_2

Figures 44 and 45 show the H-Q data points obtained with liquid fluorine in Test #02 and #03, respectively. The results from both tests are also presented in Fig. 46, scaled to the design speed of 75,000 rpm. The data shows a close correspondence with the characteristics obtained earlier on freon tests. The developed head is an average 130 ft. (40 m) higher in liquid fluorine at the design point that the head obtained in freon, which is to be expected considering that the freon H-Q tests were conducted with larger front wear ring clearance and larger inducer clearance than the fluorine tests.

Because of larger line capacities a much wider flow range was explored in liquid fluorine. The characteristics displayed in Fig. 44 and Fig. 46 show a good stable negative slope over a wide range of flows; only at very low levels, below 2.5 gpm (1.58×10^{-4} m^3/sec) at design speed, is a slight negative slope present. This is a desirable feature in applications where constant speed throttling is used because it does not present a surging problem.

The operating level on both Tests #02 and 03 was too low to obtain a reliable internal efficiency indication on the basis of pump fluid temperature rise. The pump overall efficiency calculated on the basis of turbine delivered horsepower (Ref. Table X) shows a 12 percentage point drop below the 52 percent average established for the design flow during freon tests. This is attributable to two potential sources: The aluminum oxide coating from the impeller wear rings could have come off on the initial test, increasing the internal recirculation and thereby degrading pump performance. A more probable explanation is that some icing of the turbine nozzle took place on Test #02, and on the early part of Test #03 before the gross nozzle area change at the

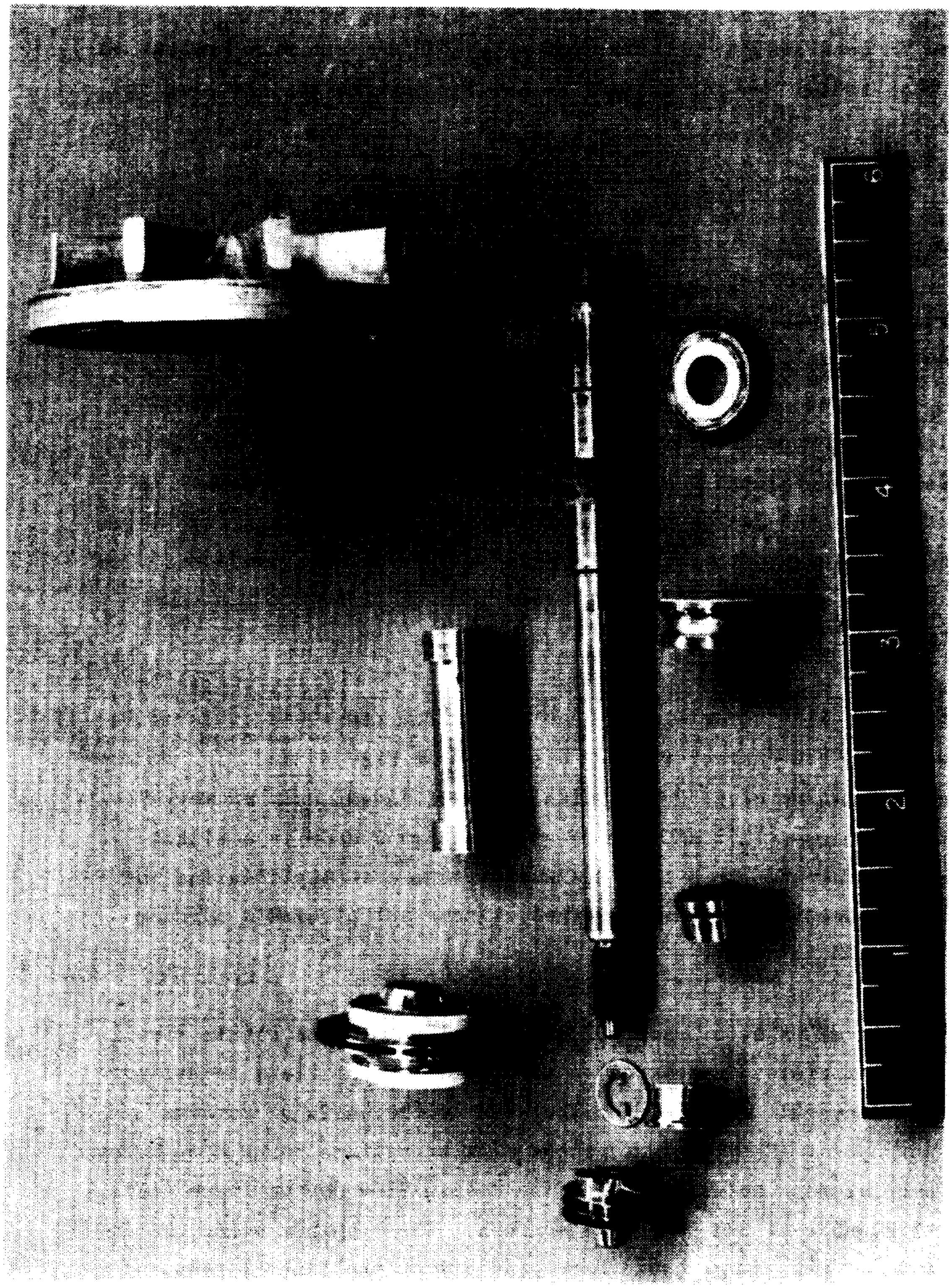

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Figure 43. Mark 36 Rotating Parts After LF_2 Testing

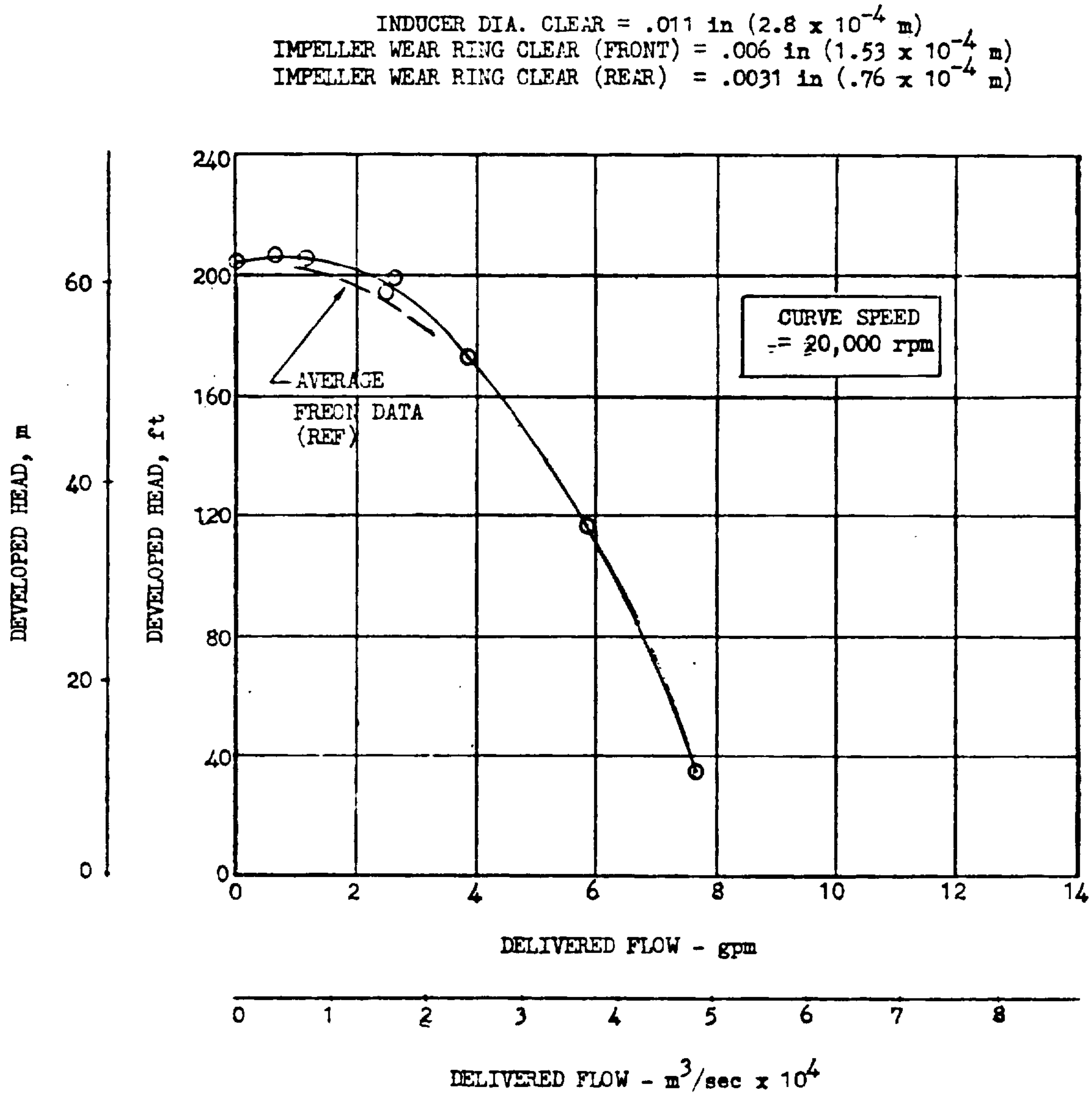


Figure 44. Mark 36 (12 gpm) Centrifugal Pump Test No. 02
H-Q Characteristics in Liquid Fluorine

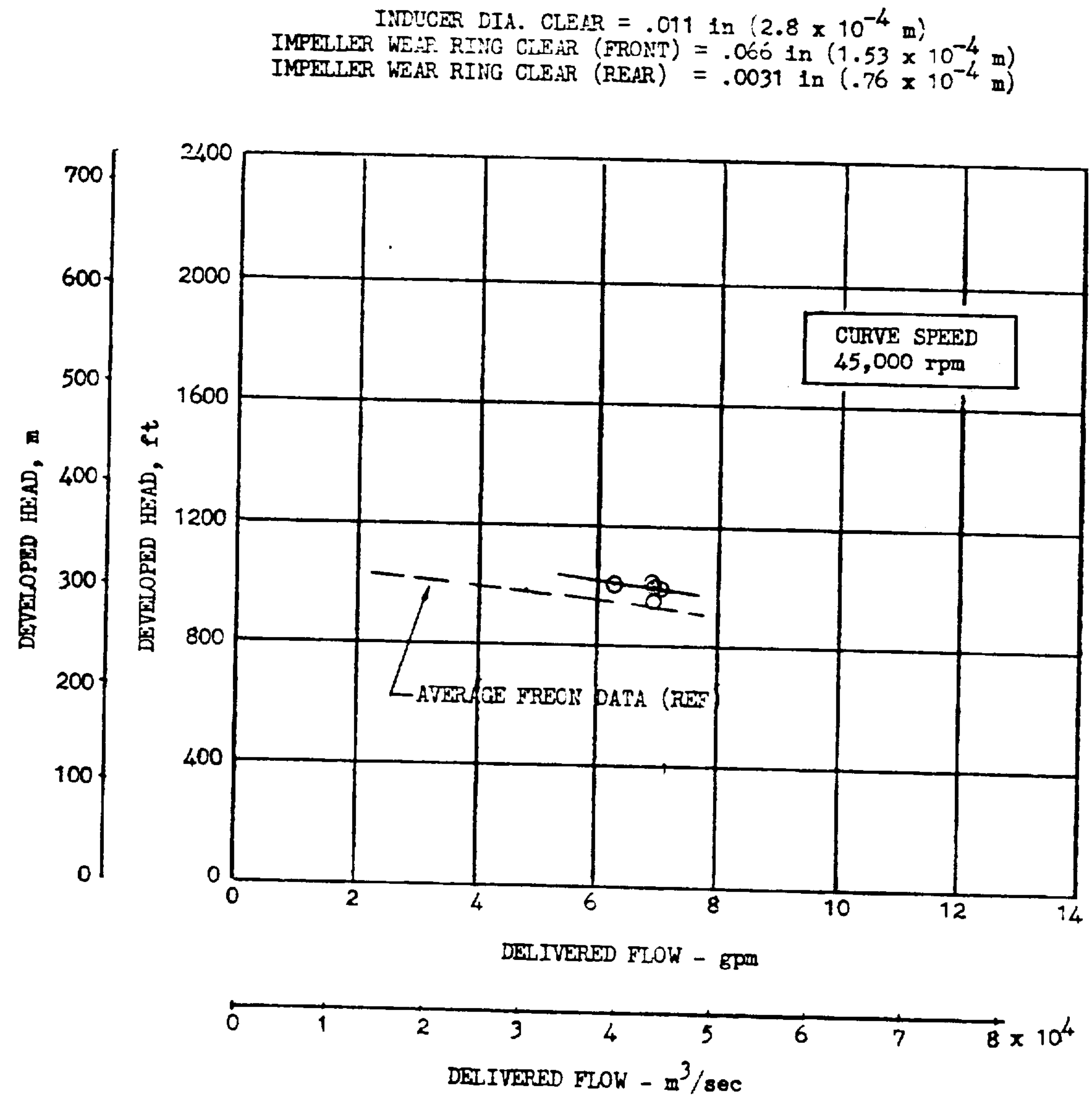


Figure 45. Mark 36 (12 gpm) Centrifugal Pump Test No. 03
H-Q Characteristics in Liquid Fluorine

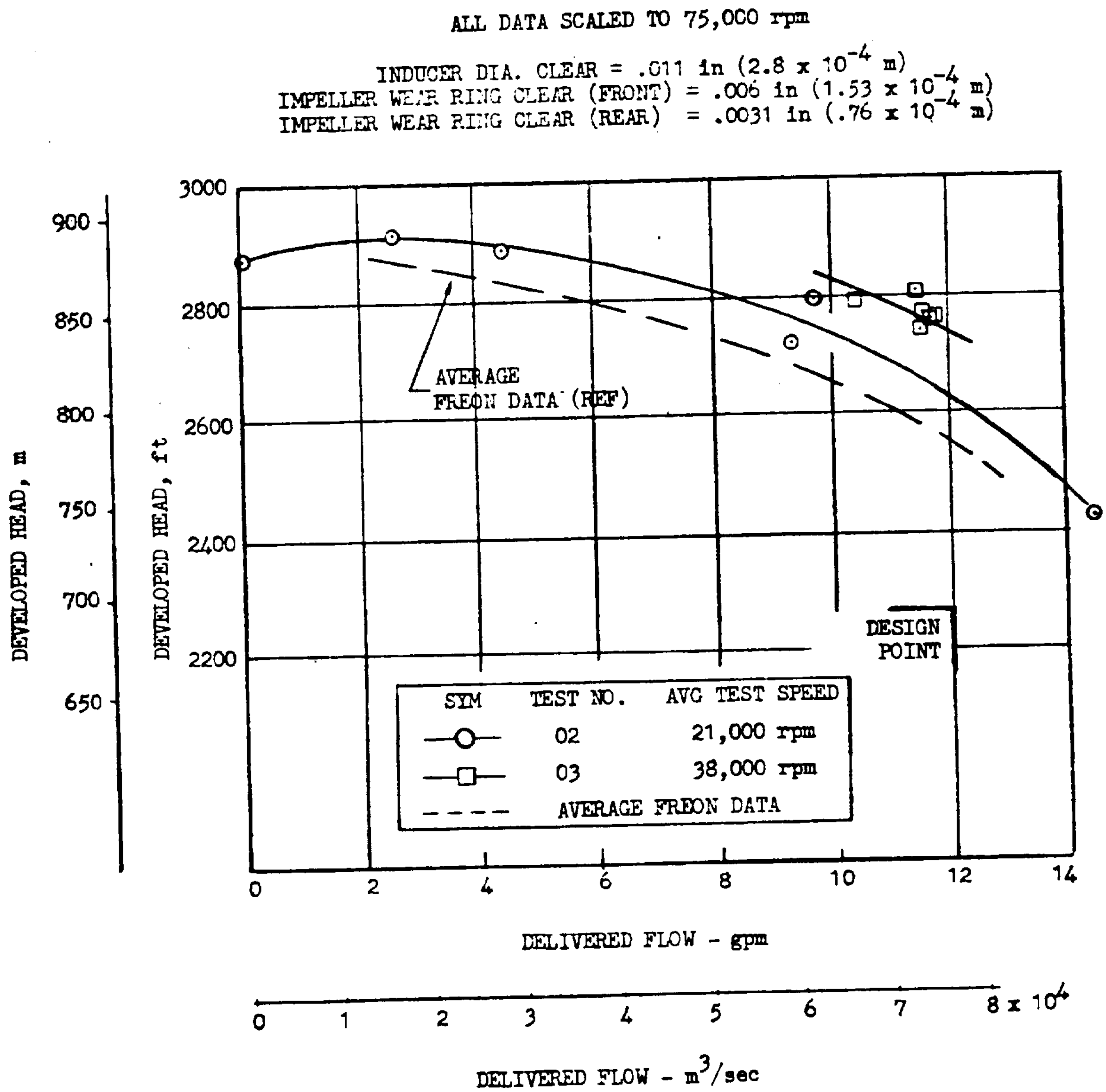


Figure 46. Mark 36 (12 gpm) Centrifugal Pump H-Q Characteristics in Liquid Fluorine

end of the test became evident. Since turbine power calculations are made using the performance map based on a constant nozzle area, any reduction in this area gives an erroneously high calculated power, which in turn makes the pump efficiency appear lower. Regardless of which of the above hypotheses is valid, the efficiency calculated from the LF_2 test data (Ref. Table X) would not represent the true performance of the pump. The measurements obtained during freon tests are accepted as a true indication of the pump performance because the two factors discussed above were absent and because the data was generated at higher speeds and more data points were available.

MECHANICAL PERFORMANCE SUMMARY

Rotordynamics

Analysis of shaft deflection data on starts disclosed that the first critical speed was in the range 28,000 - 34,000 rpm, depending on rate of acceleration, compared with the predicted steady state value of 28,000 rpm.

During turbine calibration and on the initial freon 12 tests with a large (.006 in. (1.5×10^{-4} m)) impeller rear wear-ring clearance, the shaft experienced subsynchronous whirl above 45,000 rpm, with a maximum amplitude of 0.008-inch (0.02 mm) peak-to-peak. With the close clearance rear wear ring (.003 in. ($.76 \times 10^{-4}$ m)), whirl amplitude was reduced to a negligible level of 0.002-inch ($.51 \times 10^{-4}$ m) peak-to-peak.

When testing was resumed to obtain suction performance data in freon 12, high amplitude (up to .032 in. (8.1×10^{-4} m) peak-to-peak) synchronous vibration was encountered. This was traced to a faulty bearing spacer whose ends were out of parallel by 0.0037 in. ($.94 \times 10^{-4}$ m). When the spacer was replaced the synchronous motion decreased to a negligible level. On Tests #14 through # 30, the bearing axial preload was increased from 60 pounds (267 N) to 140 pounds (623 N). This had the effect of eliminating all subsynchronous vibration and brought about a very smooth rotor operation.

Although rotor operation was very satisfactory at the higher bearing preloads, disassembly revealed that the front bearing had been distressed. This fact and reports (Ref. 2) of high ball wear rates encountered in bearing tests in fluorine indicated that a high axial load may lead to bearing problems in liquid fluorine. As a result, the bearing preload for fluorine testing was reduced to 60 pounds (267 N) from the 140 pounds (623 N) used on the last freon test series. Successful operation had been demonstrated earlier at this preload in freon in conjunction with a small rear wear ring clearance.

Initial operation below the first critical speed was normal in liquid fluorine, with shaft deflections within the scatterband of the disc surface runout. Shaft deflections at the turbine end increased on Test #03 but were still at a tolerable level (< 0.010 in. (2.54×10^{-4} m)) when operating away from the first critical speed (Ref. Fig. 35). This was also the case on Test #04 where the vibration levels were acceptable up to 100,000 rpm, and only at 104,000 rpm were excessive deflections evident. The increase in vibration level on Test #05 is attributable principally to the unbalance which was introduced in the rotor by rubbing on the turbine vanes at the end of Test #04 and possibly by losing part of the Al_2O_3 coating from the impeller. A significant contributing factor to the higher vibration levels on the last three starts was the wear in the bearing ball diameters which has the effect of reducing the bearing axial preload.

Test experience in general showed the following significant points relative to rotordynamics:

1. The pump operated satisfactorily at steady state speeds up to 80,000 rpm with the rotor balanced at 2950 rpm, i.e. without the benefit of high speed balance.
2. Subsynchronous whirl was evident at speeds above 45,000 rpm.
3. Tighter impeller wear ring clearance tended to reduce shaft deflections.
4. Higher bearing axial preloads had the effect of reducing shaft vibrations.

Bearings

Bearing operation was satisfactory in Freon 12 when the axial preload was at approximately 60 pounds (266 N). No wear or surface distress was noted with bearings operating for 41, 16 and 15.5 minutes. When the preload was increased during the freon suction performance tests to 140 pounds (622 N) and operating time was increased to 1.9 hours, pitting of the front bearing inner race occurred, although the condition of the remaining bearing components was still satisfactory and no ball wear was noted.

In liquid fluorine, with the preload set at 60 pounds (266 N), the front and rear bearing balls wore an average of .0004 inch (1.1×10^{-5} m) and .0013 (3.3×10^{-5} m) during a total operating time of 387 seconds. There was also a substantial degree of cage pocket wear.

Potential means of eliminating bearing ball and cage wear could not be explored within the scope of this program; approaches which may hold the solution include the use of alternate cage materials or "chuting" the inner races, i.e. machine relief grooves in the inner races which would allow the balls to reposition themselves in the cage pockets.

Seals

Primary Seal. In the initial test series in freon, excessive wear of the aluminum-oxide surfaces of the primary seal was encountered. This was attributed partly to the high-pressure levels in the bearing cavity before the rear wear-ring clearance was reduced, and partly to the relatively poor lubricity of Freon 12. It was decided to defer the evaluation of the aluminum-oxide surfaces until the fluorine tests, and to use primary seals with P5N carbon noses for freon testing, which had been procured for this eventuality.

One instance of bellows cracking was encountered during freon testing with the primary seal. This was caused by the high differential pressure to which the seal was subjected in the initial test series. After the bearing cavity pressure was lowered by tightening the rear wear-ring clearance, no further bellows problems were encountered, even though the pump was operated at higher discharge pressures.

The Al_2O_3 material combination on the seal nose and mating ring operated satisfactorily in liquid fluorine during the limited operating span of the pump. Leakage rates were acceptable and the seal was in good condition after the test series. The seal nose wore unevenly, approximately .004 inch (1.02×10^{-4} m) on one side and practically nothing at a point diametrically opposite.

Intermediate Seal. The controlled gap floating ring seal intermediate seal design proved to be an effective and durable concept. A total purge flowrate of less than 4 scfm ($1.9 \times 10^{-3} m^3/s$) was required to maintain a pressure of approximately 50 psig (3.44×10^6 N/m) in the seal in the absence of shaft vibrations. Sustained operation with the shaft vibrating resulted in an increase in the required purge flow to 15 scfm (7×10^{-3} m^3/s). Large shaft deflections led to dislodged antirotation pins in one instance and to a cracked ring in another.

Turbine Seal. Bellows cracking in two instances was encountered with the turbine seal. The mode of failure indicated that the bellows cracked from a stick-slip mode vibration as a result of operating the seal in dry gaseous nitrogen. Although the failure mode may not be present when the seal operates in turbine propellants which are actually used in engines, such as methane or hydrogen, corrective action was taken to eliminate the vibration by incorporating a damping device. A damper was added to the original seal on a reworkable basis, by machining away part of the housing and attaching a newly fabricated piece which contains a retaining groove for the damper (Fig. 47). The damper used was a corrugated Inconel-X spring wound circumferentially around the nose housing. It provides a friction drag of approximately 0.5 to 1.0 pound (0.23-0.46 kg).

Testing with the damped seal showed that the bellows cracking problem was eliminated. Over 1.8 hours of operations was logged in with a seal incorporating the damper, during which leakage rates were low, nose wear rate was essentially zero and the bellows remained intact.

Structural Performance

The general structural performance of the Mark 36 pump was excellent. The only structural problem encountered was flaking of the aluminum oxide coating from the impeller wear rings. Despite the fact that the pump was operated up to 138 percent of the nominal design speed no cracks or other discrepancies were evident on rotating components and stationary parts successfully sustained pressure levels ranging up to 2000 psig (13.8×10^6 N/m^2).

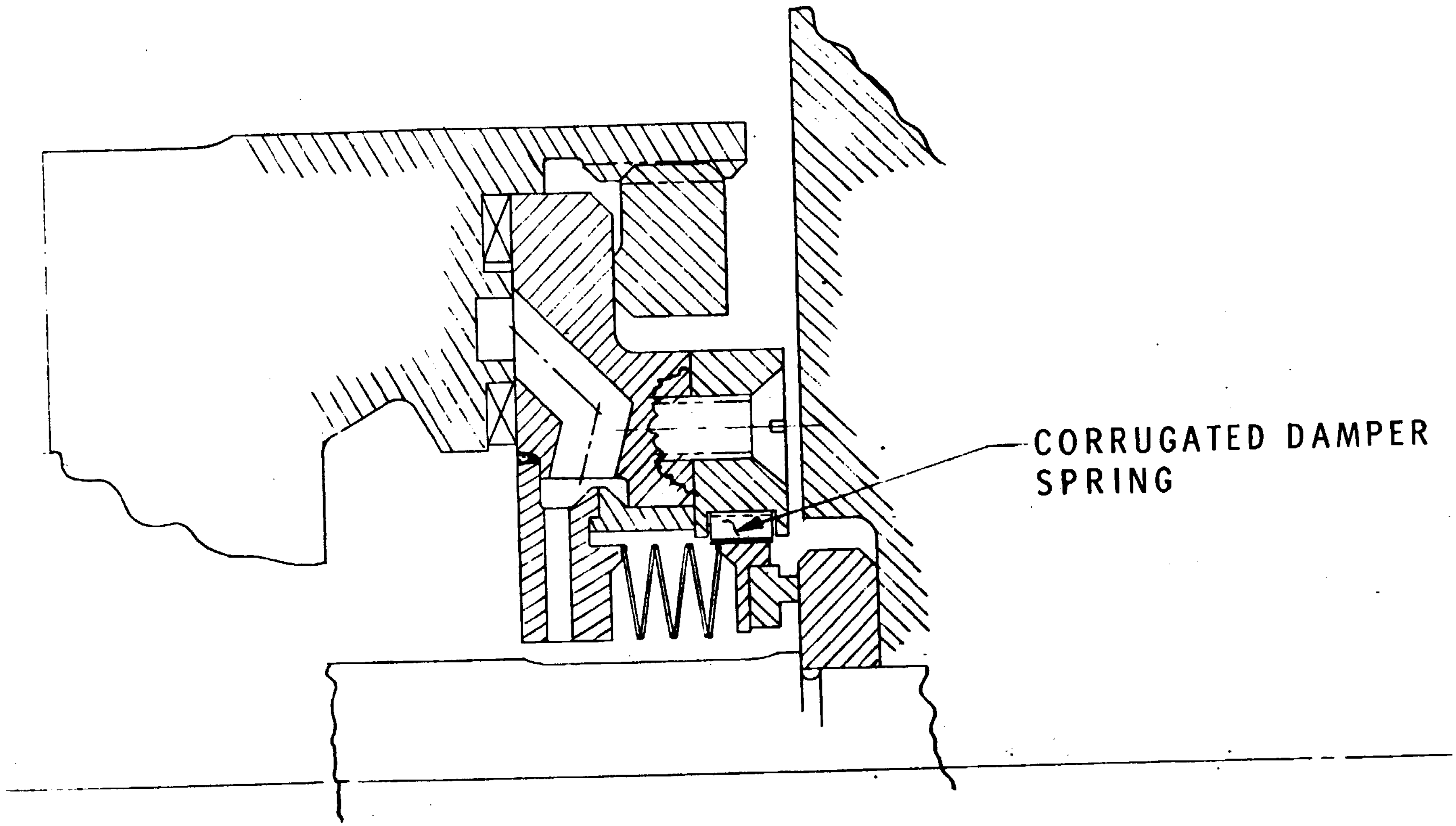


Figure 47. Mark 36 Pump Turbine Seal With Damper

PROCUREMENT AND ASSEMBLY

The configuration of the gear pump which was procured for testing is illustrated in Fig. 48. Two complete pump assemblies were procured, plus spare bearings and seals. The two main external housings, i.e., the front housing which included the inlet and discharge ports as well as the seal and bearing drains, and the rear housing which retained the two main shaft seals, were cast from Inco 718, and finish machined in the critical areas to the required tolerances. The other principal parts of the pump including the gears, both bearing retaining plates, and the center plates were machined from wrought Inco 718 alloy. Surface coating was applied to the above parts in two areas: Chromium plating was used on the bearing journals of the gear shafts, and plasma-sprayed aluminum-oxide coating was applied to the side plates in the area where gears were in contact with the side plates.

The mating rings for the shaft seals were machined from solid Kentanium K-162B material. The rubbing noses of all three shaft seals were of plasma-sprayed aluminum oxide. To obtain better adherence between the aluminum-oxide rubbing nose and the Inco 718 seal housing, a base coat of 0.002-inch (0.051 mm) 100-percent nichrome was applied to the housing, followed by a 0.003-inch- (0.075 mm) thick layer of 50-percent nichrome and 50-percent Al_2O_3, and another 0.003-inch (0.076 mm) layer of 25-percent nichrome and 75-percent Al_2O_3.

The bearing rollers and races were machined from wrought 440C steel alloy, and the cagesfrom K-monel bar. To minimize the tendency of roller skewing in the bearings, the internal fit was established at 0.0001-0.0003-inch (0.0025-0.0076mm) interference. The fit tolerance was held within the foregoing range by custom-machining the inner race of each bearing: Inner races with slightly oversized outer diameters were installed on the journals and the outer diameters were subsequently finish ground to the dimensions which resulted in the above fit for each bearing.

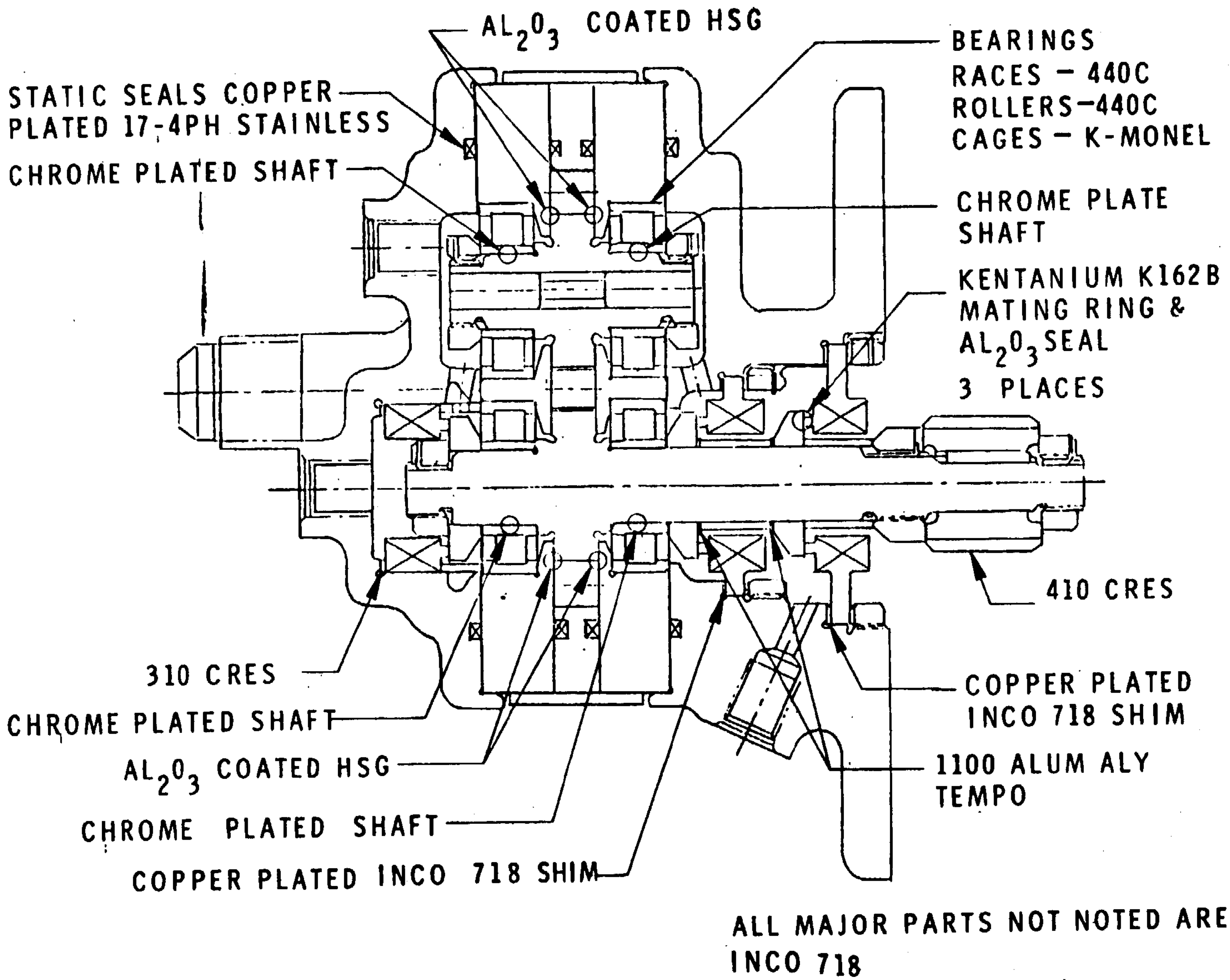


Figure 48. Mark 37 Pump

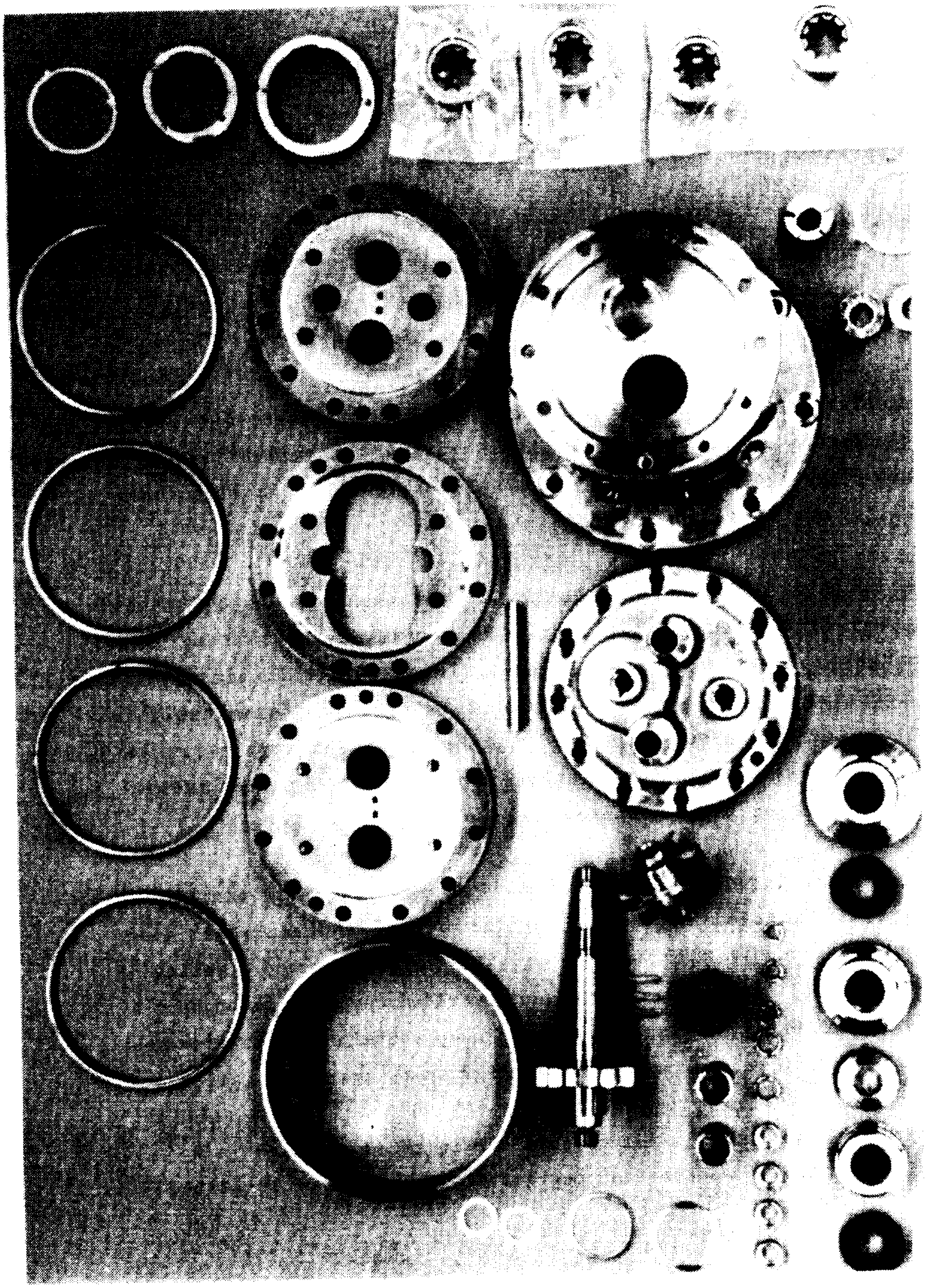
Because of the low speed and small diameters involved, balancing of the rotating components of the pump was not required.

The component parts of the gear pump, prior to assembly, are shown in Figure 49.

Buildup of the pump was started by assembling loosely the center plate, the two gears (with the bearing races already on the journals), and the two side plates with the appropriate flange seals. The bearing rollers, cages, and outer races were next installed by a light press. The three plates were then aligned by installing a press fit pin and the shrink fit external retaining ring. The four axial bolts which retain the plates and the idler gear bearing inner race retaining nuts were installed and locked.

The axial thrust balancing seal was installed into the front housing with a shim to adjust its operating length, and its mating ring was secured to the driver shaft by a retaining nut. Subsequently the front and rear housings were attached with appropriate flange seals and secured with axial retaining bolts. The primary seal and its mating ring and the secondary seal and its mating ring were installed. The operating length of each seal was adjusted to the proper value by shimming at the mounting flange. The assembly was concluded by installing the spline and its retaining nut and lock on the driver shaft.

The breakaway and rotating torque of the driver shaft was measured and ambient leak checks were taken of the shaft seals with gaseous helium at 30 psig.



1XY52-2/25/70-C1

Figure 49. Mark 37 Gear Pump

The pertinent clearance and leak check values obtained on the initial pump build are presented in Table XII.

Figure 51 shows the gear pump completely assembled. The weight of the pump without external fittings was 18.5 pounds (82.3 n).

FREON 12 TESTS

Facility Description

Testing of the Mark 37 gear pump was accomplished in Rotary Test Cell #2 of Rocketdyne's Engineering Development Laboratory at Canoga Park, California. The pump fluid loop used for the gear pump was identical to that shown in Fig. 21 for the centrifugal pump.

Figures 51 through 54 show the mounting of the gear pump and the general arrangement of the facility. The drive unit utilized was a 40-hp U.S. Motors Varidrive Aero Test Stand. The output speed of this unit, variable from 2,500 rpm to 15,000 rpm, is derived from a constant speed motor (2,200 rpm) driving a self-contained speed increased through two belts with variable ratio pulleys. The control motor for the pulley ratio change mechanism is powered and controlled remotely.

The torque metering device used for gear pump testing was a noncontacting, foil-type strain gaged torquemeter manufactured by S. Himmelstein and Company, model number MCRT 9-02T(1-3). The torquemeter had a rating of 1000 in.-lb (113 nm) with a 100-percent overlaod capacity throughout a speed range of 0 to 7500 rpm. A speed pickup contained in the torquemeter provided 60 pulses per revolution for shaft speed indication.

This type of torquemeter utilizes foil strain gages arranged in a Wheatstone bridge, positioned 45 degrees from the axis of rotation,

TABLE XII. MARK 37 GEAR PUMP INITIAL ASSEMBLY VALUES

Total axial clearance	idler gear	0.00014 - 0.0003 in. (0.0035 - 0.0076 mm)	
	driver gear	0.00009 - 0.0002 in. (0.0023 - 0.0051 mm)	
Diametral clearance	idler gear	0.0004 in. (0.0102 mm)	
	driver gear	0.001 in. (0.0254 mm)	
Initial shaft torque		210 in. lb (23.8 n.m)	break
		190 in. lb (21.4 n.m)	run
Shaft torque after 150 "break in" hand rotations		150 in. lb (17.0 n.m)	break
		135 in. lb (15.3 n.m)	run
Primary seal leakage @ 30 psig; GH_e		28 scim ($7.65 \times 10^{-6} \frac{m^3}{sec}$)	
Secondary seal leakage @ 30 psig; GH_e		135 scim ($36.8 \times 10^{-6} \frac{m^3}{sec}$)	
Pressure balance seal leakage @ 30 psig; GH_e		3 scim ($0.82 \times 10^{-6} \frac{m^3}{sec}$)	

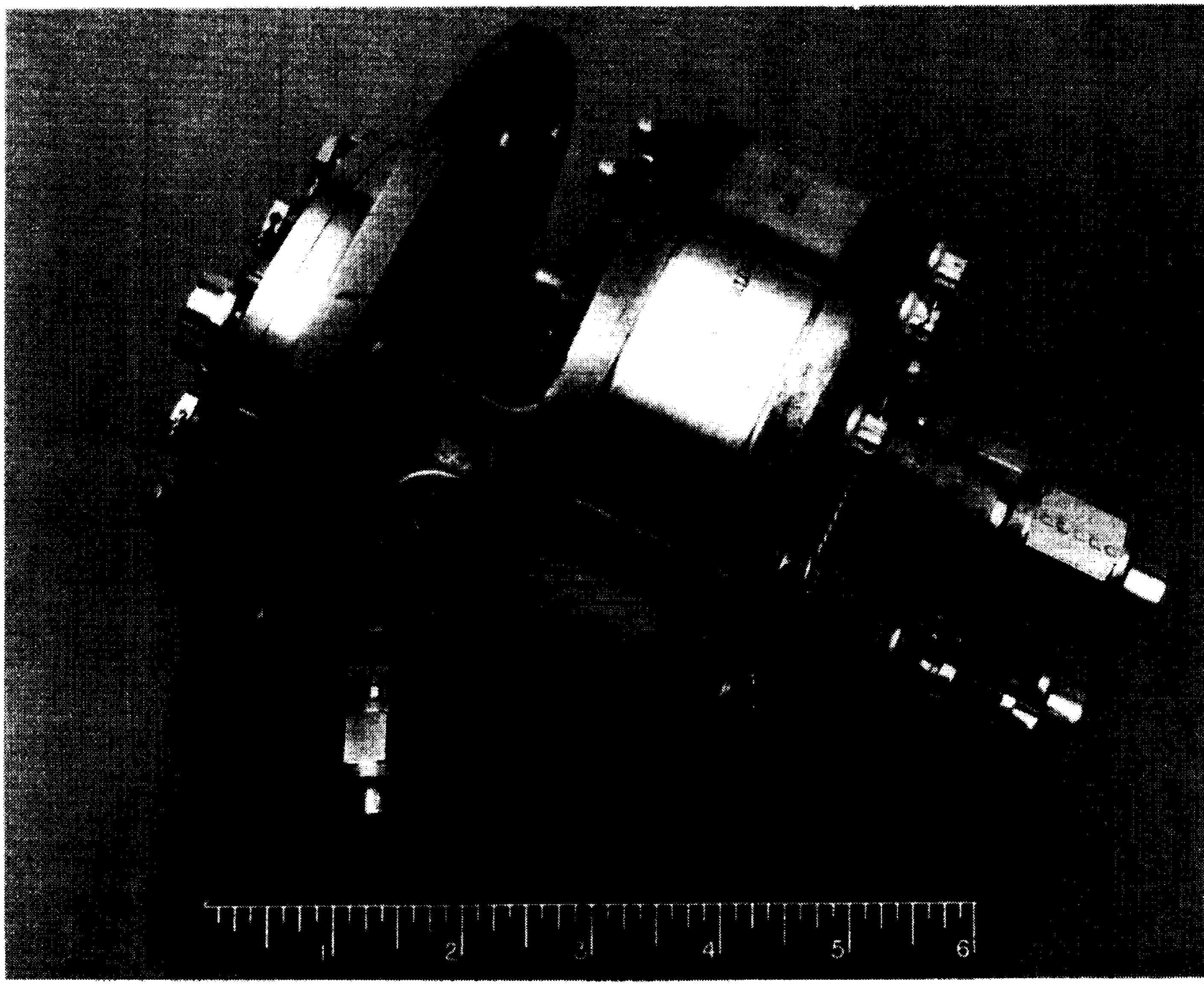


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Figure 50. Mark 37 Gear Pump

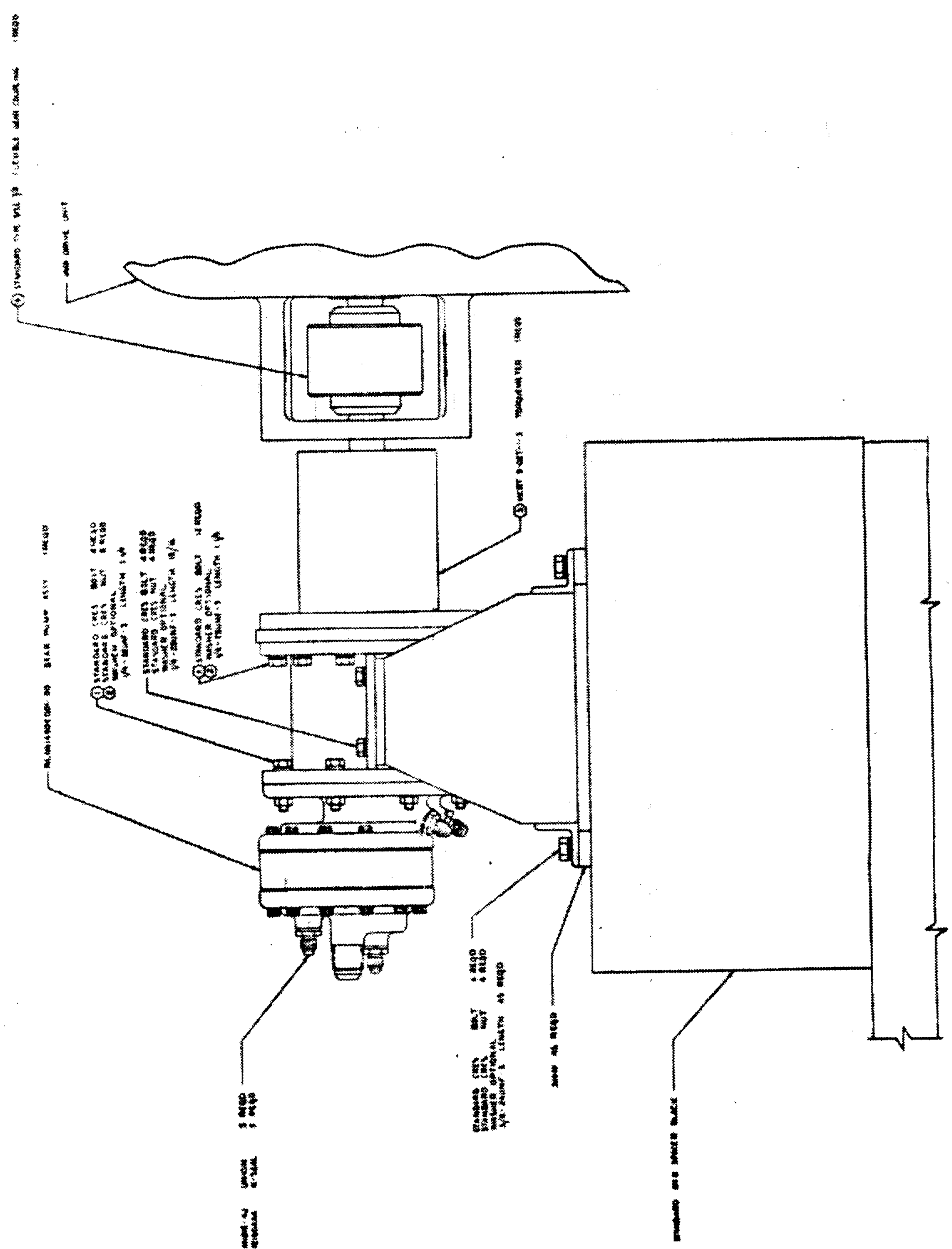

Figure 51. Mark 37 Pump Test Bed

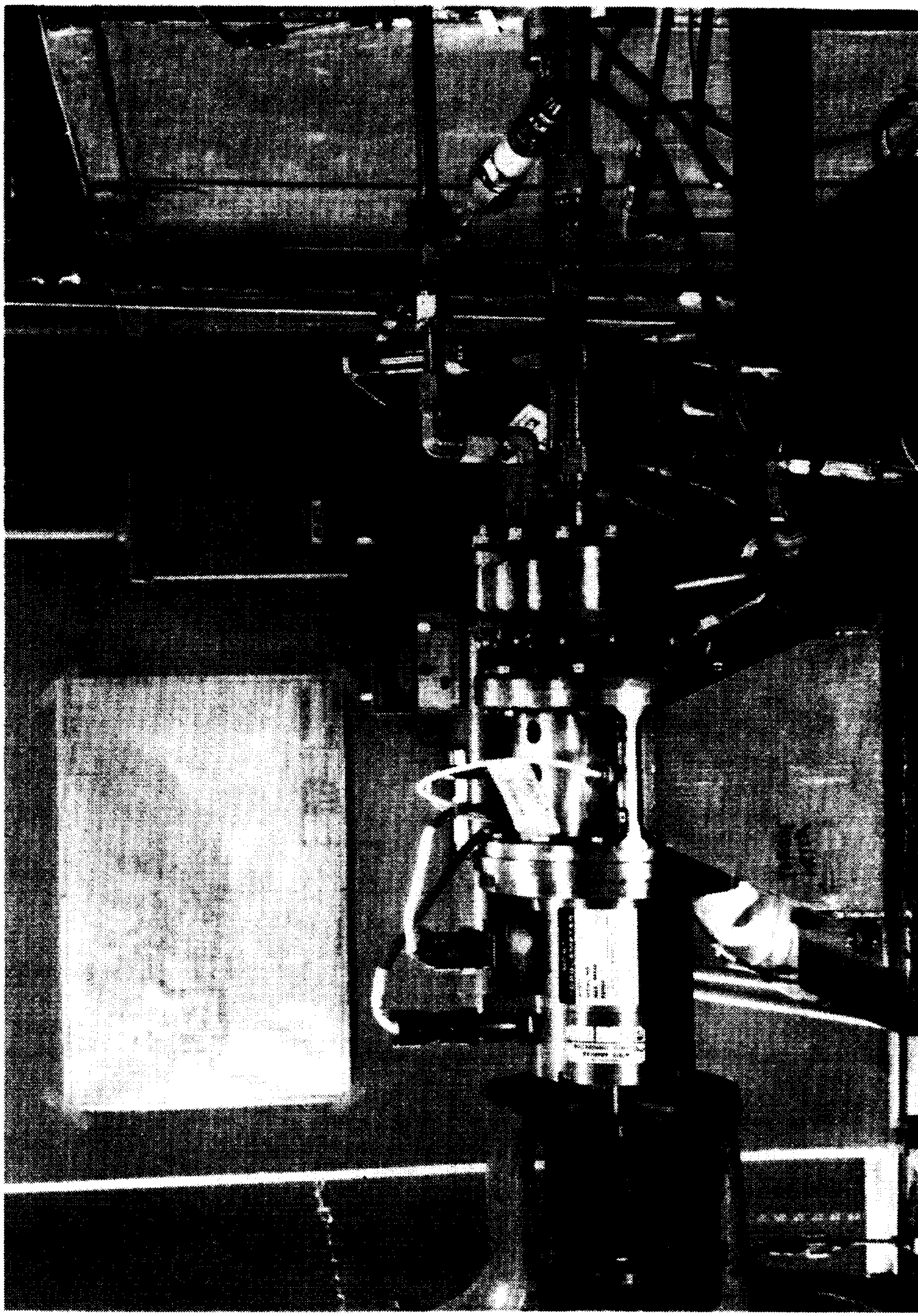

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Figure 52. Mark 37 Gear Pump Freon 12 Test Facility

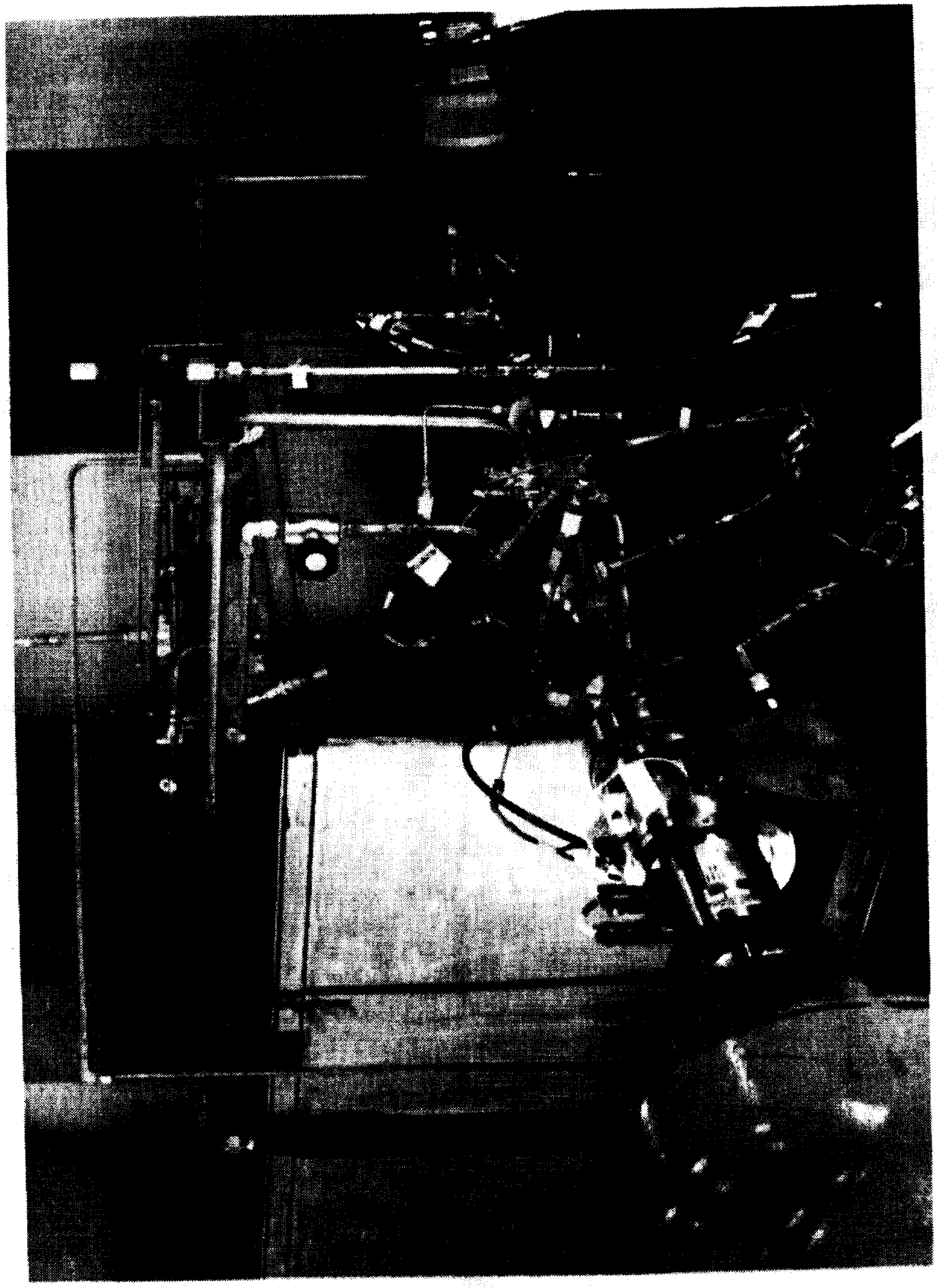

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Figure 53. Mark 37 Gear Pump Freon 12 Test Facility

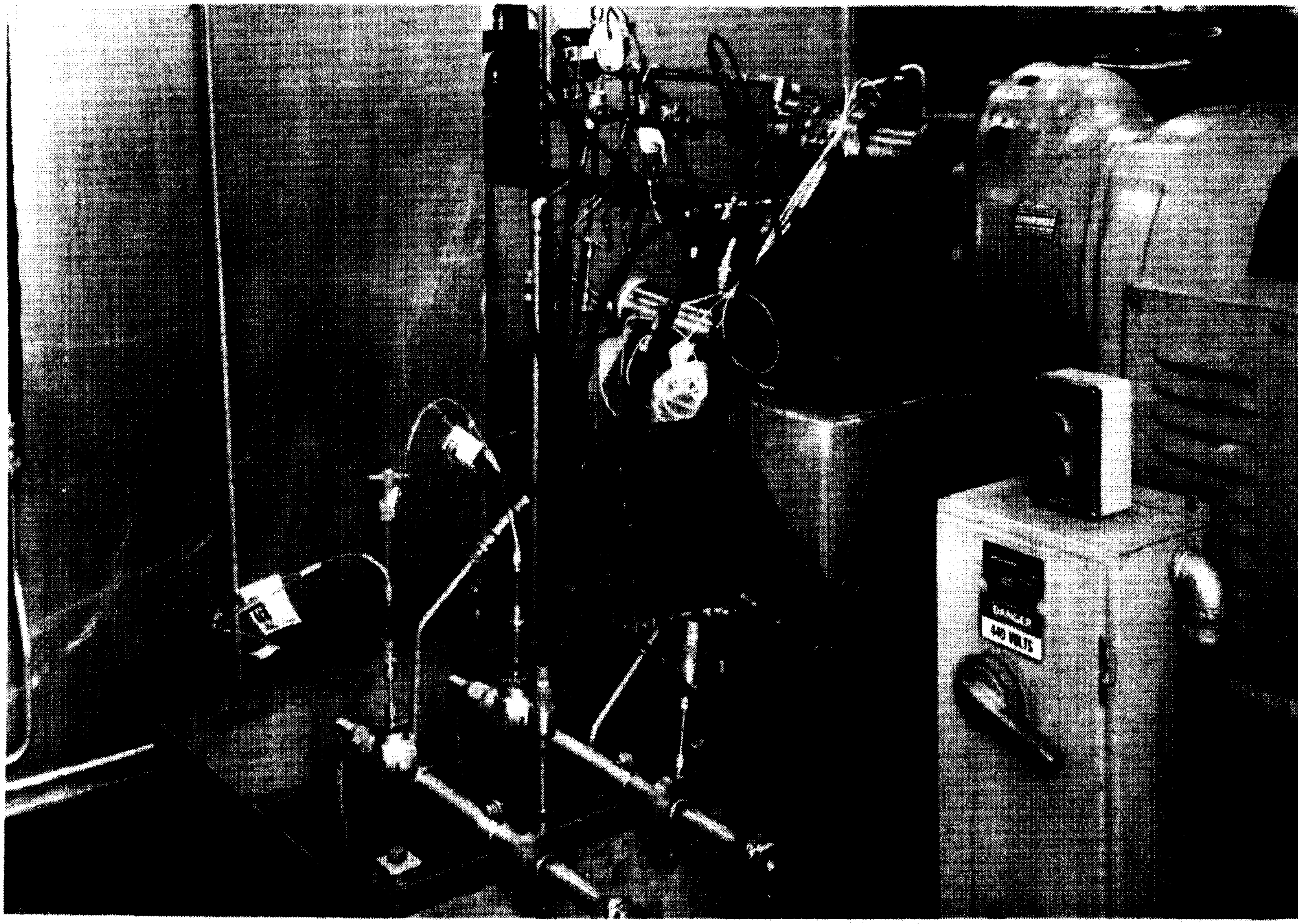

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Figure 54. Mark 37 Gear Pump Freon 12 Test Facility

so that the bridge measures shaft twist (strain) which is directly proportional to shaft torque. The torque shaft is positioned within the torquemeter housing by two high-precision preloaded ball bearings.

Instrumentation

Figure 55 presents the parameters which were measured on the gear pump tests with freon. Pump inlet temperature and pressure, discharge pressure, speed, torque, bearing coolant return flow, and both seal drain pressures were recorded on a Brush recorder, whereas the other parameters were registered on an oscillograph. Radial displacement of the spline coupling was measured in two planes with Bently transducers. The signal from the Bentlys and also the dynamic pressure measurements were recorded on tape.

Test Description

The objective of the freon test series was to determine the hydrodynamic performance and mechanical operating capability of the gear pump and correct any deficiencies which may appear before operation in liquid fluorine is undertaken.

Test #001. To optimize the volumetric efficiency of the pump, the gear tip and side clearances had been set at very low values. With the side clearance, in particular, the intention was to start out with essentially a line-to-line fit, and allow the gears to wear their way into the aluminum-oxide coating of the side plates. In line with the above approach, the objective of the initial test in freon was to conduct several starts and stops to wear-in the gears. Because of the nature of the drive system, the minimum operating speed was fixed at 2500 rpm and the speed buildup time was also fixed at approximately 2 seconds from zero to full speed.

Pump Inlet

Inlet static pressure

Inlet temperature

Inlet pressure, dynamic

Pump Discharge

Discharge static pressure

Discharge temperature

Discharge pressure, dynamic

Flow

Drain Lines

Bearing coolant return pressure

Bearing coolant return temperature

Bearing coolant return flow

Primary seal drain pressure

Primary seal drain temperature

Secondary seal drain pressure

Secondary seal drain temperature

Torquemeter

Torque

Speed

Other

Pump fluid ΔT

Bently #1

Bently #2

Figure 55. Mark 37 Gear Pump Instrumentation

On the first test, a sudden increase in torque was experienced immediately after the electric drive was started. The torque level reached 600 in.-lb (68 n.m), and at that point the shear pin in the coupling between the torquemeter and the pump sheared.

The pump was disassembled for inspection, and it was noted that rubbing had taken place between the idler gear outer diameter and the inner diameter of the center plate. A slight contact was evident also at the outer diameter of the driver gear. No damage was sustained by the pump components.

Hardware Modifications. The idler gear was reworked by removing approximately 0.0007 inch (0.0178 mm) radially from the tip. The driver gear tip was cleaned up of any contact marks; in the process, 0.0001 inch (0.00254 mm) was removed from the tip radially. The tip clearances resulting from the above modifications were: 0.0009 inch (0.0228 mm) radially at the idler gear and 0.0006 inch (0.0152 mm) radially at the driver gear.

The pump was assembled after the above modifications to provide additional clearance were incorporated. The shaft torque was 25 - 70 in.-lb (2.82 - 7.9 n.m) with the pump completely dry, and 25 - 40 in.-lb (2.82 - 4.51 n.m) with trichloroethylene in the pump. The spread in hand torque was attributed to skewing of the rollers in the bearings. It was believed that the degree of skewing would be reduced under operating loads.

Test #002. The objective of the second test was to operate at 2500 rpm and 500 psig pressure (3.45×10^6 n/m^2) discharge pressure for one minute to wear-in the aluminum-oxide coating on the side plates, opposite the gear faces. With the increased tip clearances, no significant contact was anticipated at the tips.

For approximately 0.5 second after the electric drive was started, the torque level was normal at 80 in-lb (9.05 n.m). At that point, the torque suddenly spiked to 580 in-lb (65.5 n.m), after which it dropped to 30 in.-lb (3.4 n.m). A half a second later, another sudden increase occurred which exceeded 1000 in.-lb (113 n.m). The test was terminated and the pump was disassembled for inspection.

Posttest examination revealed the following: The driver gear shaft was twisted off just forward of the driver spline. Apparently the pump incurred a high torque which caused the shear pin in the coupling between the electric motor and torquemeter to be sheared on the first torque spike; the internal parts of the coupling seized shortly thereafter, causing the second torque spike, which resulted in shearing of the pump shaft.

The driver gear teeth scored the aluminum-oxide coating on both housing plates. The driver gear teeth were discolored near the pitch diameter on both sides. indicating that a substantial amount of friction heating was present. The aluminum oxide opposite the idler gear teeth was not damaged. There was evidence of light rubbing at the OD of the driver gear. (See Figure 56.)

It appears that the friction heat generated during start caused the driver gear to expand sufficiently to eliminate the 0.0001 - 0.0002-inch axial clearance. When the fit became tight, the aluminum-oxide coating was scored which in turn further increased the interference until the pump bound up, causing the excessive shaft torque.

Hardware Modifications. From the results of the last test series, it appears that the aluminum-oxide coating on the side plates will not be able to take the scraping action of the gear teeth. To alleviate this problem, the gear faces were relieved 0.0005 inch (0.013 mm) on the side, opposite the gear teeth (Fig. 57). With this modification, any axial rubbing will take place

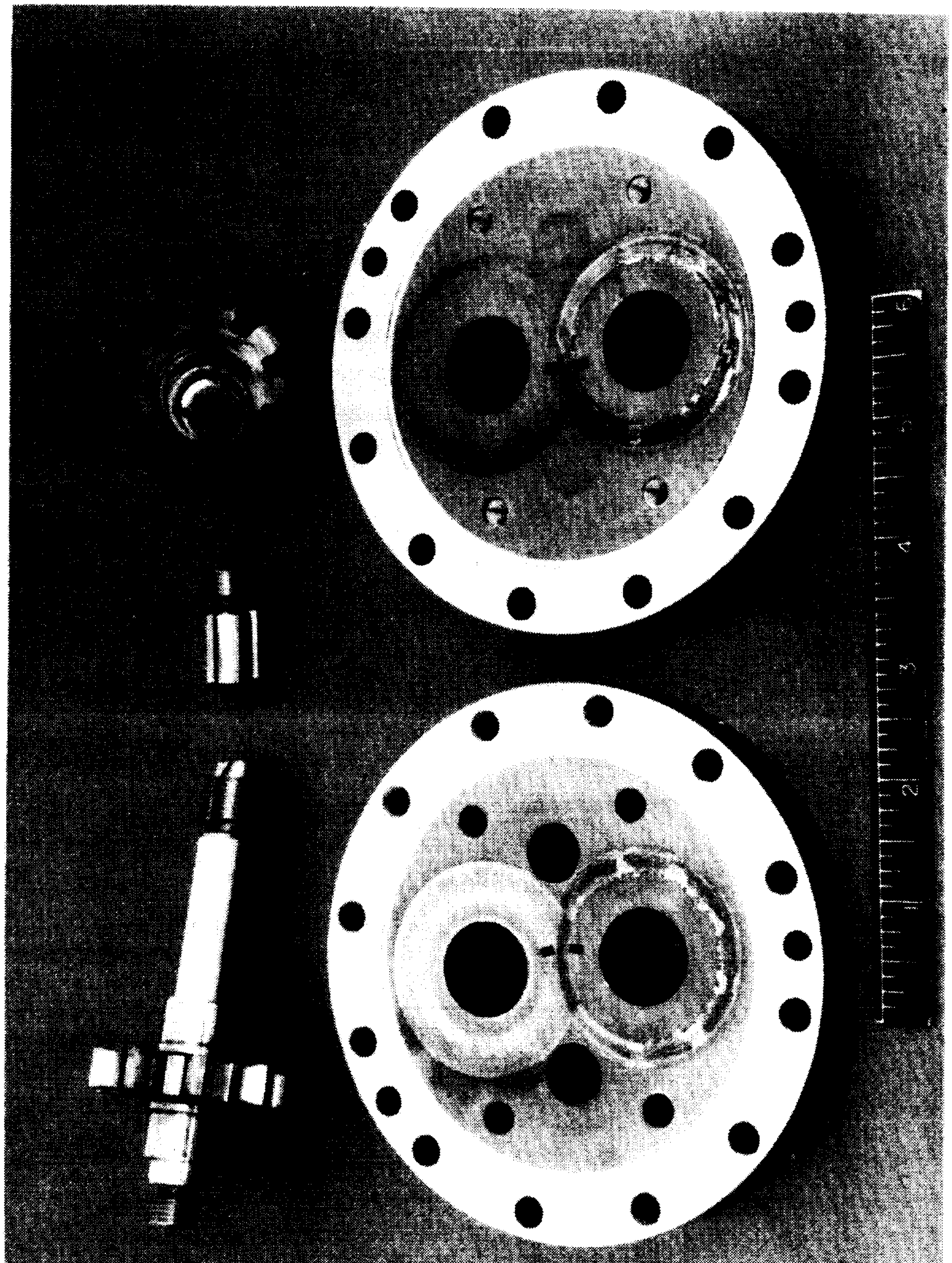

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Figure 56. Mark 37 Gear Pump After Test No. 002

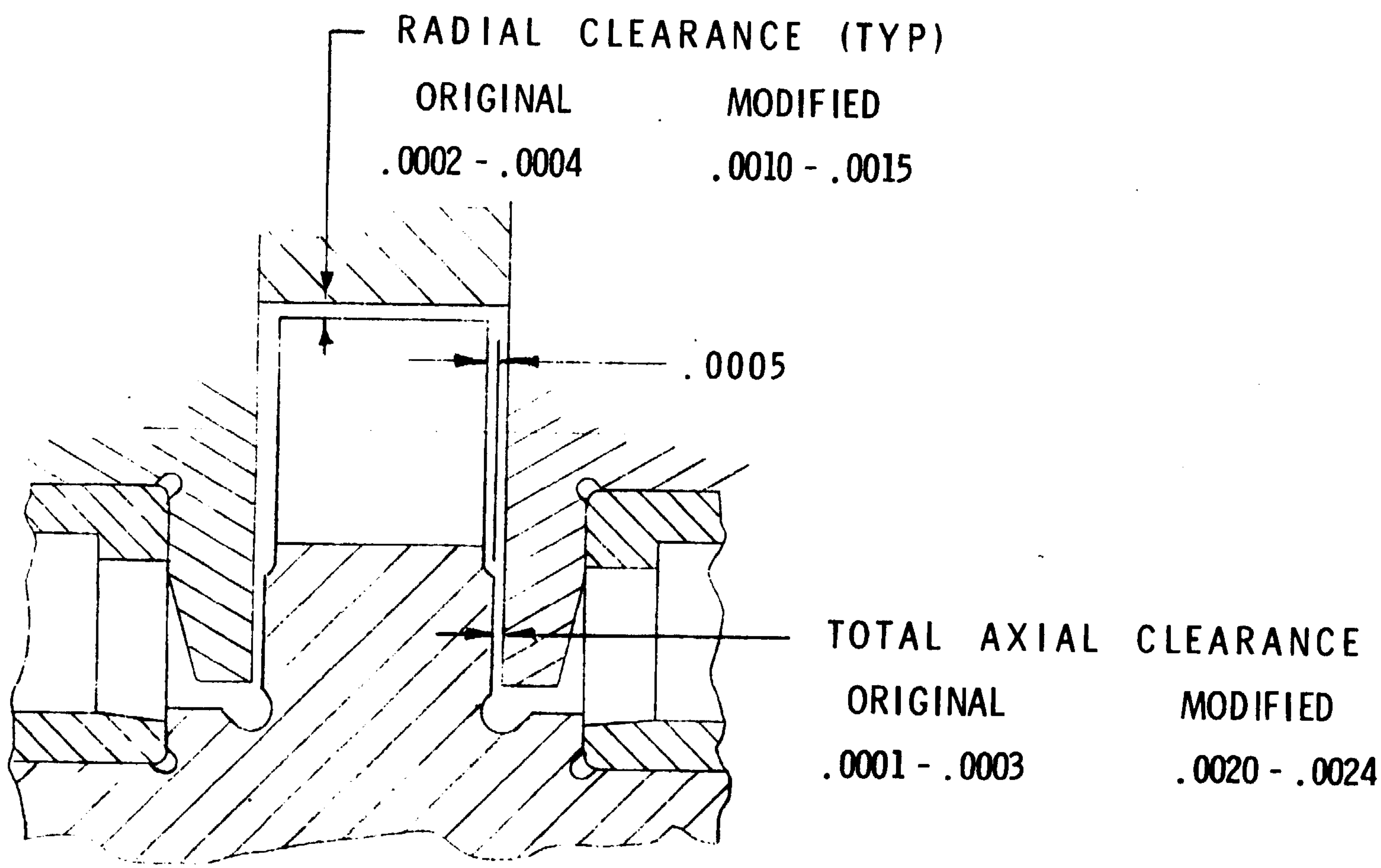

Figure 57. Mark 37 Pump Gear Tooth Relief

at the uninterrupted surface of the hub, and scraping action is eliminated. In addition, the clearance between the gear hub and the side plates was increased to provide a minimum of 0.001 inch (0.0254 mm) per side, resulting in a total axial clearance of 0.002-0.0024 inch (0.0508-0.061 mm).

To avoid all contact at the gear tips, the tip clearance was changed to a minimum of 0.001 inch (0.0254 mm).

These clearance increases will result in lower hydraulic efficiencies but should eliminate the mechanical problems uncovered by the first two tests, due to the extremely close initial design clearances. The clearances can be optimized later to improve efficiency.

A design change was also made to the mechanics of aligning the two side plates with the center plate. The original concept included a straight solid interference fit pin and a retaining ring at the outer diameter of the plates. Difficulty was experienced in installing and disassembling both parts. With the pin in particular, the problem was sufficiently severe to necessitate using a spring pin in lieu of the solid pin on the first two assemblies. To alleviate this problem the design was changed to delete the retaining ring and straight pin, and accomplish the alignment with two tapered solid pins.

The design changes outlined above were incorporated into one set of hardware. However, before any testing was conducted with the modified hardware, a program review was held in which it was concluded that the centrifugal pump offered a better potential of fulfilling the requirements of this program. As a result further testing and development of the gear pump was abandoned.

NOMENCLATURE

A_e	=	nozzle effective area (in^2)
C_p	=	specific heat at constant pressure Btu/lb/°F
D_m	=	mean diameter (in)
g	=	gravitational constant (ft/sec^2)
H	=	pump developed head (ft)
K	=	gas constant
N	=	speed (rpm)
N_{ss}	=	suction specific speed
$NPSH_{CRIT}$	=	net positive suction head at 2% head drop (ft)
P	=	pressure (psig)
P_s	=	pump inlet pressure (psfa)
P_v	=	vapor pressure (psfa)
Q	=	volume flow (gpm)
$\mathcal{T}$	=	turbine output torque (lb-ft)
T	=	temperature (°F)
T	=	pump inlet to discharge temperature rise (K°)
V	=	inlet fluid velocity (ft/sec)
η	=	efficiency
γ	=	specific weight (lb/ft^3)
$\dot{w}$	=	weight flow (lb/sec)

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