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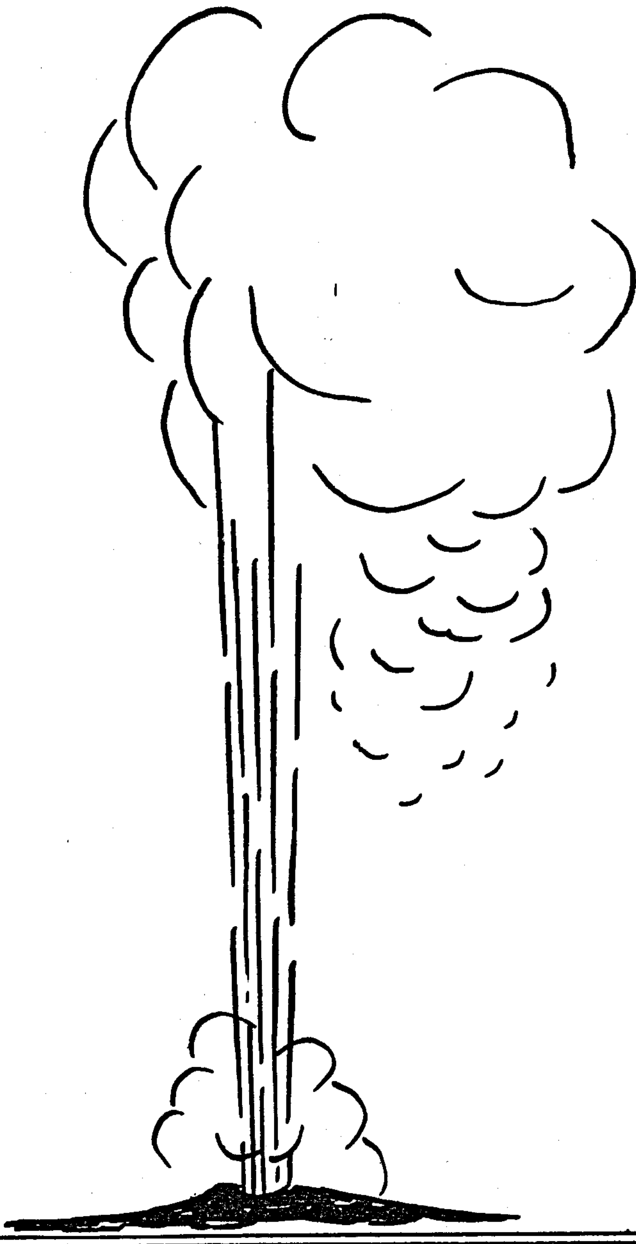

TESTS OF A TWO-STAGE, AXIAL-FLOW, TWO-PHASE TURBINE

By
David G. Elliott

December 15, 1982

Work Performed Under Contract No. AI01-80ER10614

Jet Propulsion Laboratory
California Institute of Technology
Pasadena, California

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ABSTRACT

A two-phase-flow turbine with two stages of axial-flow impulse rotors was tested with three different working-fluid mixtures at a shaft power of 30 kW. The turbine efficiency was 0.55 with nitrogen-and-water of 0.02 quality and 94 m/s velocity, 0.57 with Refrigerant 22 of 0.27 quality and 123 m/s velocity, and 0.30 with steam-and-water of 0.27 quality and 457 m/s velocity. The efficiencies with nitrogen-and-water and Refrigerant 22 were 86 percent of theoretical. At that fraction of theoretical, the efficiencies of optimized two-phase turbines would be in the low 60 percent range with organic working fluids and in the mid 50 percent range with steam-and-water. The recommended turbine design is a two-stage axial-flow impulse turbine followed by a rotary separator for discharge of separate liquid and gas streams and recovery of liquid pressure.

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The steam-and-water tests at Biphase Energy Systems were assembled by Charles Dame and directed by Phillip Maddox.

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I. INTRODUCTION

Turbines that can operate with two-phase flows having large liquid fractions would be useful in geothermal power, waste-heat conversion and other applications discussed in the mid-term report (Ref. 1). The purpose of the work discussed here and in Ref. 1 was to investigate the most efficient rotor designs for two-phase turbines.

It seemed clear at the outset that an impulse turbine must be used, with all of the pressure drop taken across nozzles and the rotor flow taking place at constant pressure. This is because there is only weak coupling between the liquid and gas in two-phase expansion, and long, straight nozzles are required for efficient acceleration of the liquid (Ref. 2). In addition, for maximum efficiency the nozzles must have liquid-gas mixers at the inlets to assure uniform flow distribution at the start of expansion.

The two phase turbine problem thus reduces to one of slowing down two-phase jets at constant pressure in some kind of rotor assembly. The desirable characteristics of the rotors are efficiency, erosion resistance, and low cost. Erosion resistance depends mainly on the rotor materials and liquid velocity, independent of rotor geometry. The cost of rotating components, no matter how complex, is small compared with other costs of a power system. Therefore, the basis for selecting rotor designs is chiefly maximum efficiency.

One approach to efficient two-phase rotor design is to separate the flow leaving the nozzles into liquid and gas streams and provide different rotor flow paths optimized for each phase (Ref. 3). The difficulty is that flow separation at the nozzle-exit flow speed, by any method proposed so far, has losses that are as high or higher than merely sending the unseparated two-phase flow through a conventional turbine wheel.

Based on present information, it appears that the most efficient two-phase turbine design is an axial-flow impulse turbine with the liquid impinging on the blades and forming a thin film but otherwise following the same path as the gas. A separator can be added at the exit if desired. Within this basic framework, much can be done through optimization of blade shapes and stage speeds to obtain good efficiency.

The chief loss is from the large friction of the liquid films on the blades. To reduce this loss a special staging method can be used. The first stage is operated at a blade speed that is a large fraction of jet speed, rather than at half the jet speed or less, as in conventional impulse turbines. This is possible without excessive rotor stress because the speed of two-phase jets is low. At the high rotor speed the relative speed of the incoming liquid is lower than it would be at conventional rotor speeds, and friction loss is reduced.

The flow leaving the first-stage rotor has considerable forward velocity, and a second stage is used to recover the leaving kinetic energy. The second stage rotates in the same direction as the first stage at a lower speed.

There are no intervening stationary nozzles or blades. More than two stages can be used, allowing still lower relative velocities in each stage, with further reduction in liquid friction losses.

A two-stage turbine was built to test this method of staging. Initially, the turbine had a single nozzle. In tests with water and nitrogen (Ref. 1) the turbine efficiency (shaft power divided by isentropic power) was 0.51, which was only an 8 percent gain over the efficiency of 0.47 achieved with a single-stage rotor. The limited gain was partly explained by increased windage loss (the power required to drive the rotors in air) with the two-stage turbine. Subtracting out the windage loss, the single-stage turbine had an efficiency of 0.49 and the two-stage turbine had an efficiency of 0.55, a 12 percent gain.

Theoretically, the two-stage turbine should have had a 20 percent gain in efficiency over the single-stage turbine, without windage loss. The reason for the smaller gain in practice appeared to be that about a third of the liquid became delayed in the first stage rotor, clinging to the blades and tip shroud as a slowly moving layer. This liquid eventually drained from the first stage rotor and entered the second stage rotor, but the first-stage torque was reduced by the delayed flow effect.

It was hoped that the amount of delayed liquid could be reduced, relative to the total flow, by adding more nozzles so that the blades of the first-stage rotor would always enter a new jet before the flow from the previous jet was able to slow down. A new nozzle assembly with six nozzles was built. Initially, only two additional liquid-gas mixers were made so that the gain with two additional nozzles could be measured. The tests with three nozzles showed only a 2 percent gain in efficiency attributable to reduction of the delayed flow effect. Therefore, additional mixers were not built, and the remaining tests were completed with three nozzles. The efficiency of the three-nozzle, two-stage turbine was 0.55 with water-and-nitrogen.

The turbine was then tested with Refrigerant 22 ($CHClF_2$) and it achieved an efficiency of 0.57. Finally, the turbine was tested with steam-and-water at Biphase Energy Systems, Santa Monica, California. The measured efficiency was only 0.30, due, in part, to limitations of the particular test conditions.

II. TURBINE CONSTRUCTION

Figure 1 is a photograph of the nozzle assembly used in the water-and-nitrogen and Refrigerant 22 tests. Six circular nozzles were mounted at a 20 degree angle from the plane of the rotors with as small a circumferential spacing between nozzles as could be fabricated. The converging-diverging walls of the nozzles were formed by casting epoxy resin around a mandrel that was split at the throat for removal from each end. Three of the nozzles had liquid-gas mixers installed; the other three nozzles were not used. The purpose of the mixers was to inject low-speed two-phase streams of equal liquid/gas ratios at 61 uniformly-distributed points to properly start the nozzle expansion. The nozzle contour and mixer construction were the same as shown in Figs. 10 and 11 of Ref. 1. The throat diameter of each nozzle was 13 mm, the exit diameter (at the upstream end of the exit ellipse) was 28 mm, and the length was 350 mm.

In the steam-and-water tests, two nozzles built by Biphase were used. They had throat diameters of 5.6 mm, exit diameters of 30 mm, and lengths of 275 mm.

The first-stage rotor is shown added in Fig. 2 and the second-stage rotor in Fig. 3. The first-stage and second-stage rotors were the same ones designated as Rotors 2 and 1, respectively, in Ref. 1; design details are given there.

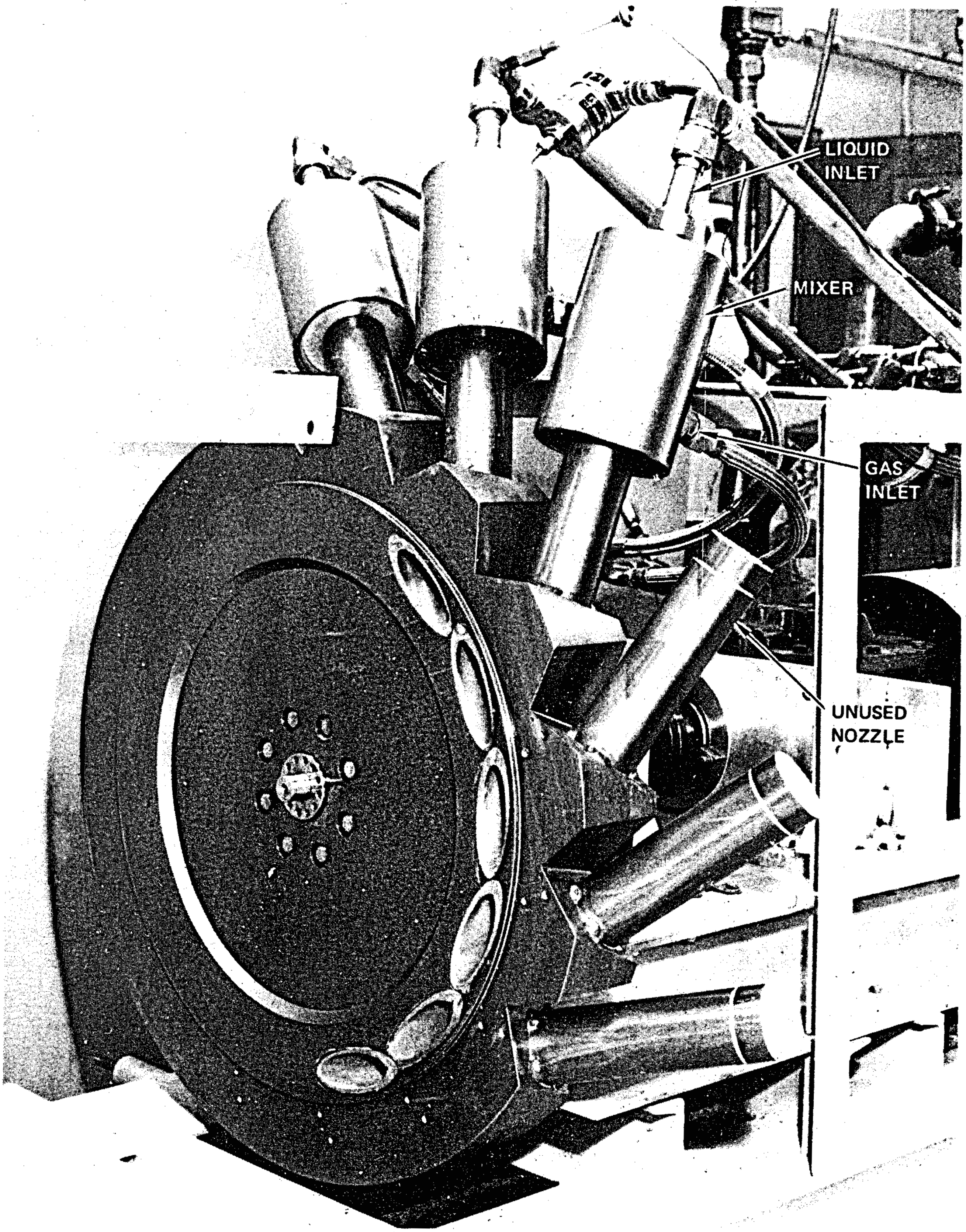

Figure 1. Turbine nozzle assembly

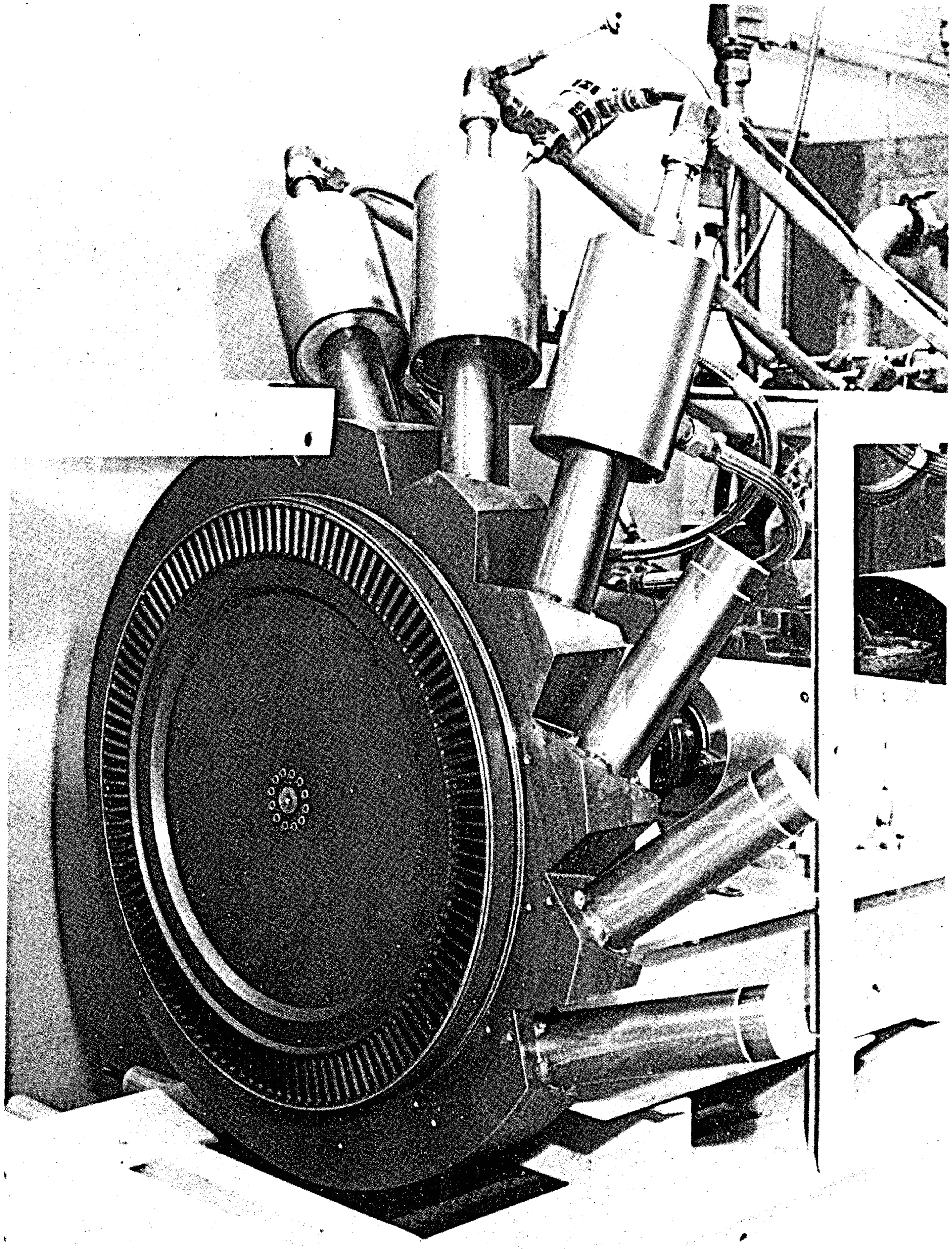

Figure 2. Turbine with first stage

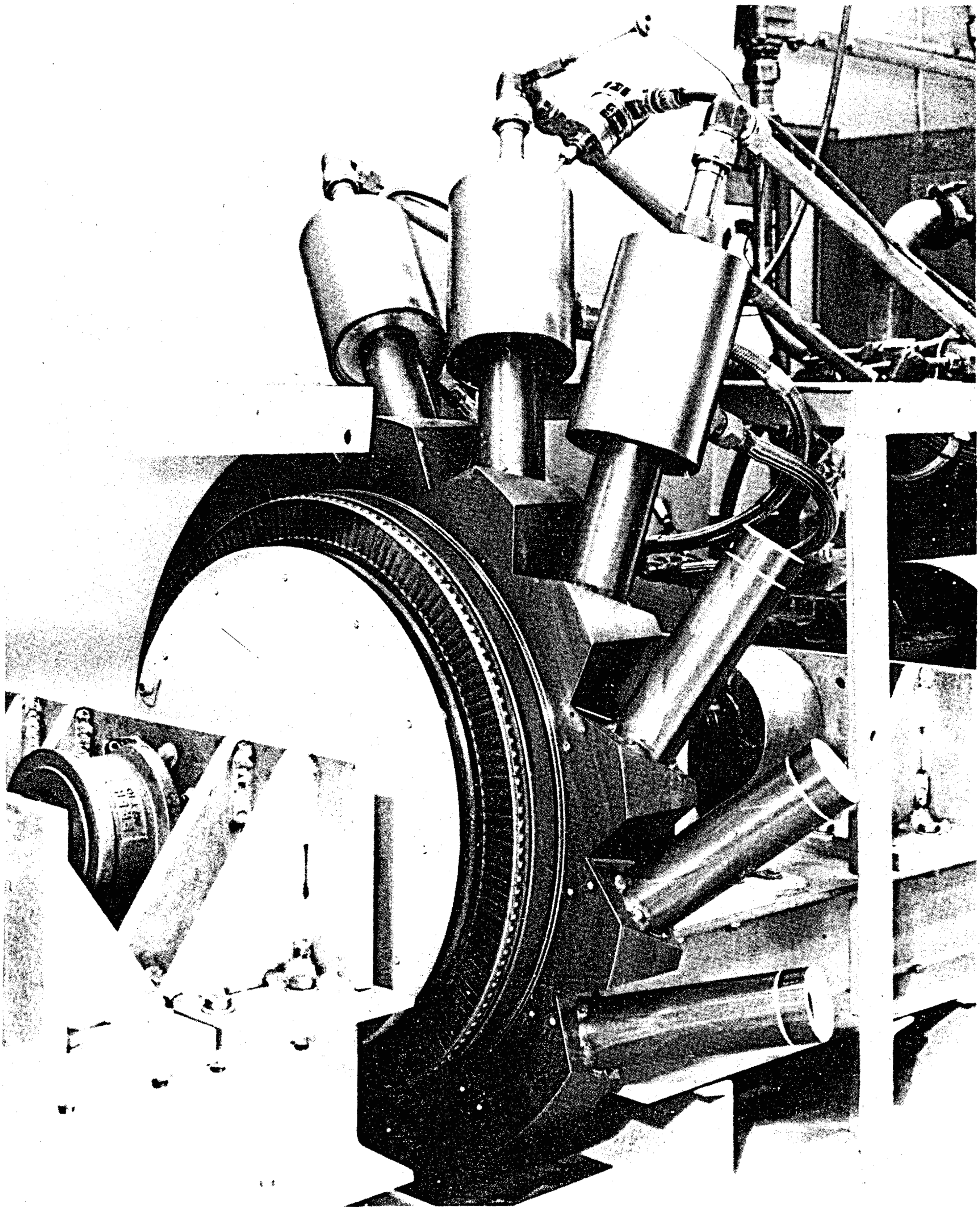

Figure 3. Turbine with first and second stages

III. TEST PROCEDURES

The method of loading the rotors and measuring shaft power in the tests at JPL is shown in Figs. 4 and 5. The first-stage rotor was connected to a 50 kW electric motor through a variable-speed belt drive, a step-up gear box, and a rotating torque transducer. The second-stage rotor drove a water brake which was pivoted for torque measurement.

During operation, the first-stage rotor drove the electric motor as a generator, and speed was controlled by adjusting the variable-speed belt drive. The second-stage rotor drove the water brake, and speed was controlled by adjusting the water flow rate. The first-stage power output was measured as the product of transducer torque and first-stage rotor speed. The second-stage power output was measured as the product of water-brake torque and second-stage rotor speed.

The feed system for Refrigerant 22 tests is shown in Fig. 5. Liquid Refrigerant 22 at room temperature was fed from a nitrogen-pressurized tank through a hand-operated throttling valve to the liquid inlets of the three mixers. A small amount of superheated Refrigerant 22 vapor at about 50 °C was fed by its own pressure from a Refrigerant 22 cylinder in a hot-water bath to the gas inlets of the mixers. The liquid and vapor flow rates were measured by turbine meters. The vapor flow rate was only set high enough (2 percent of the liquid flow) to provide a vapor volume flow equaling the volume flow of the 61 liquid streams entering the nozzle.

In the nitrogen-and-water tests, water was fed to the liquid inlets from pumps, and nitrogen was fed to the gas inlets from a bottle bank.

Figure 6 shows the two rotors installed at Biphase Energy Systems for steam-and-water tests. Saturated hot water was throttled through a valve to provide a flow of 3 percent quality in the feed lines to the nozzles. Mixers to assure uniform distribution were not used, but the nozzle exit velocity measured by Biphase nevertheless agreed with the Reference 2 theoretical velocity.

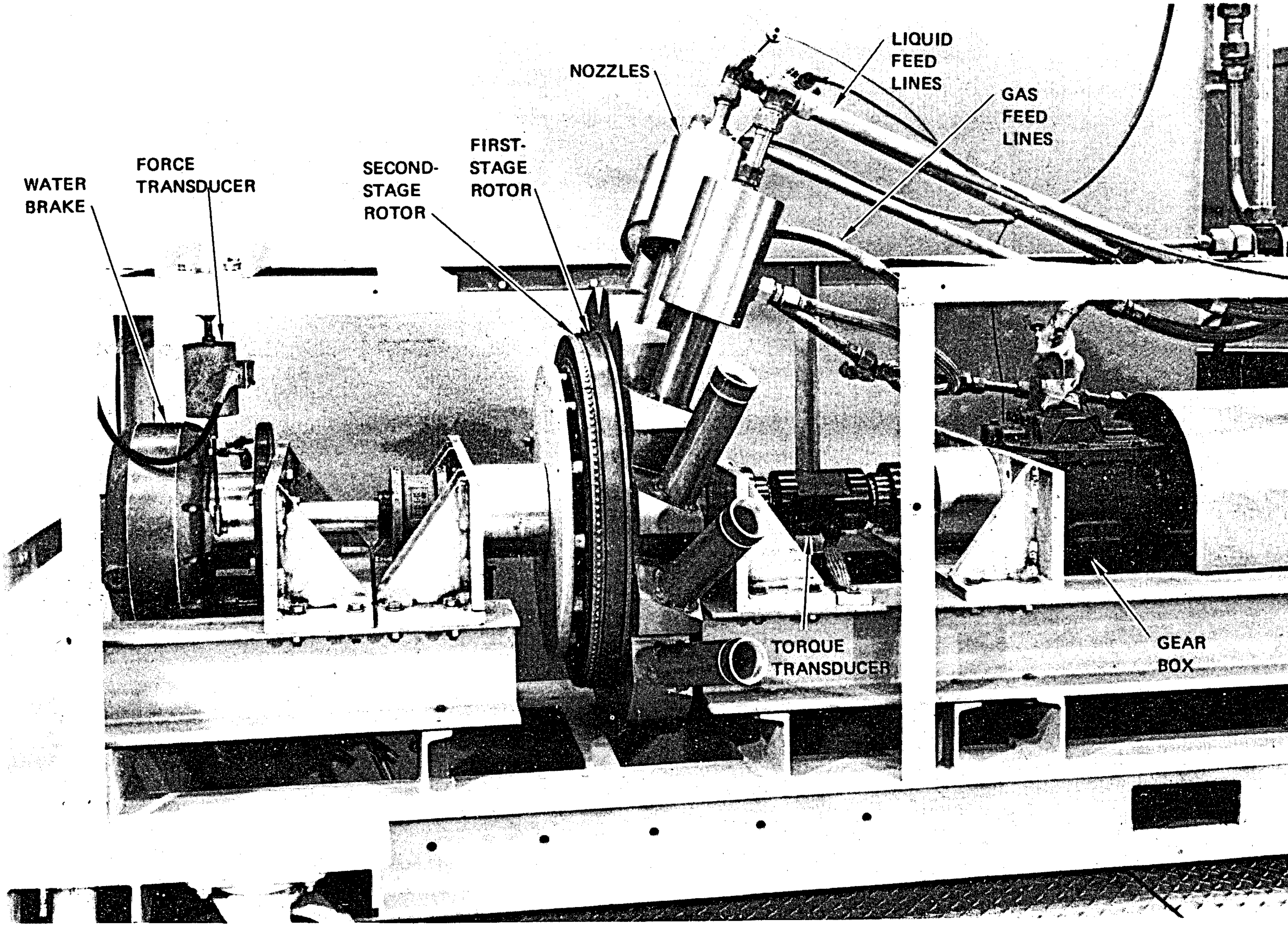

Figure 4. Rotor assembly and loading arrangement

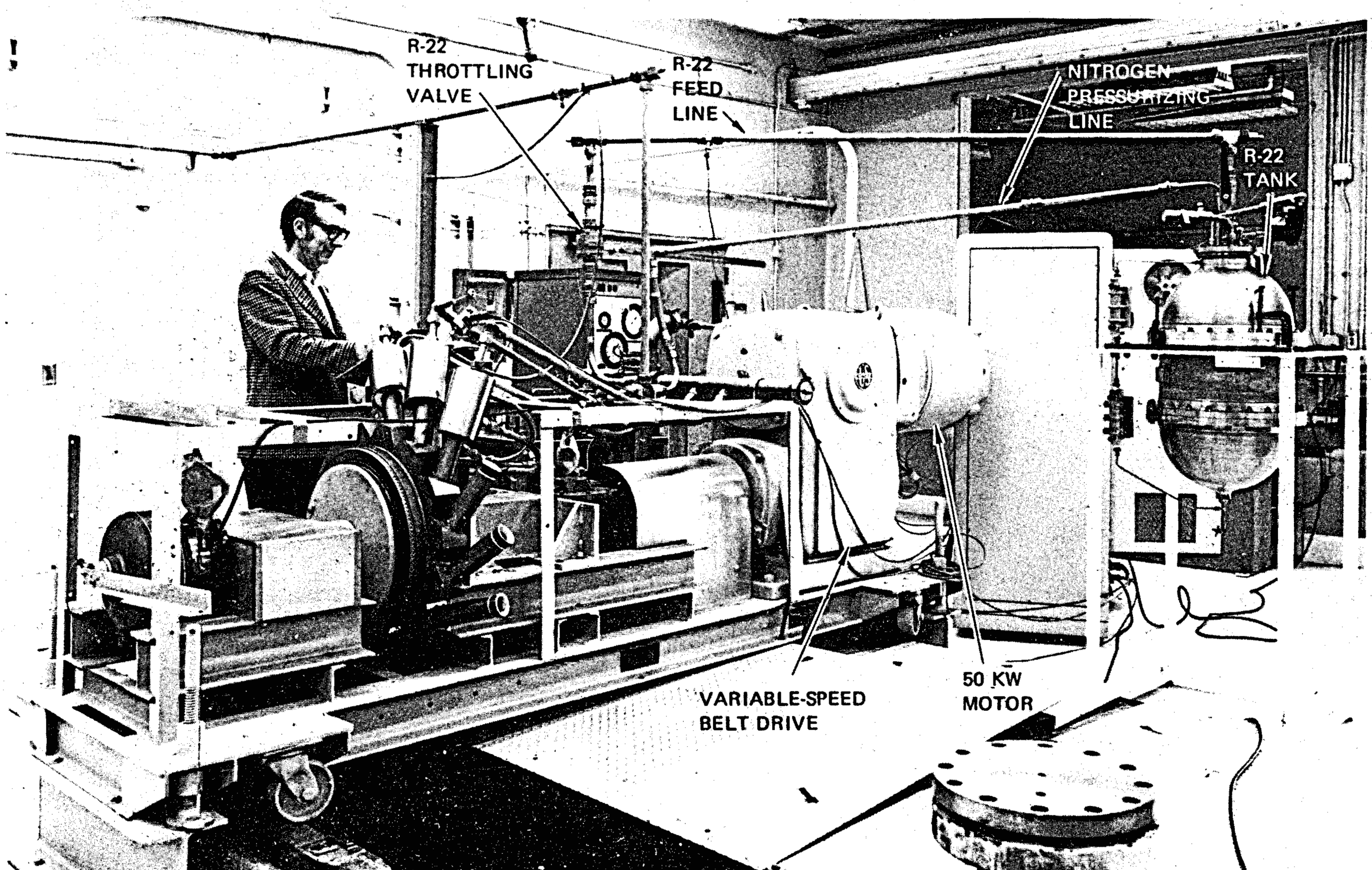

Figure 5. Turbine experiment with Refrigerant 22 feed system

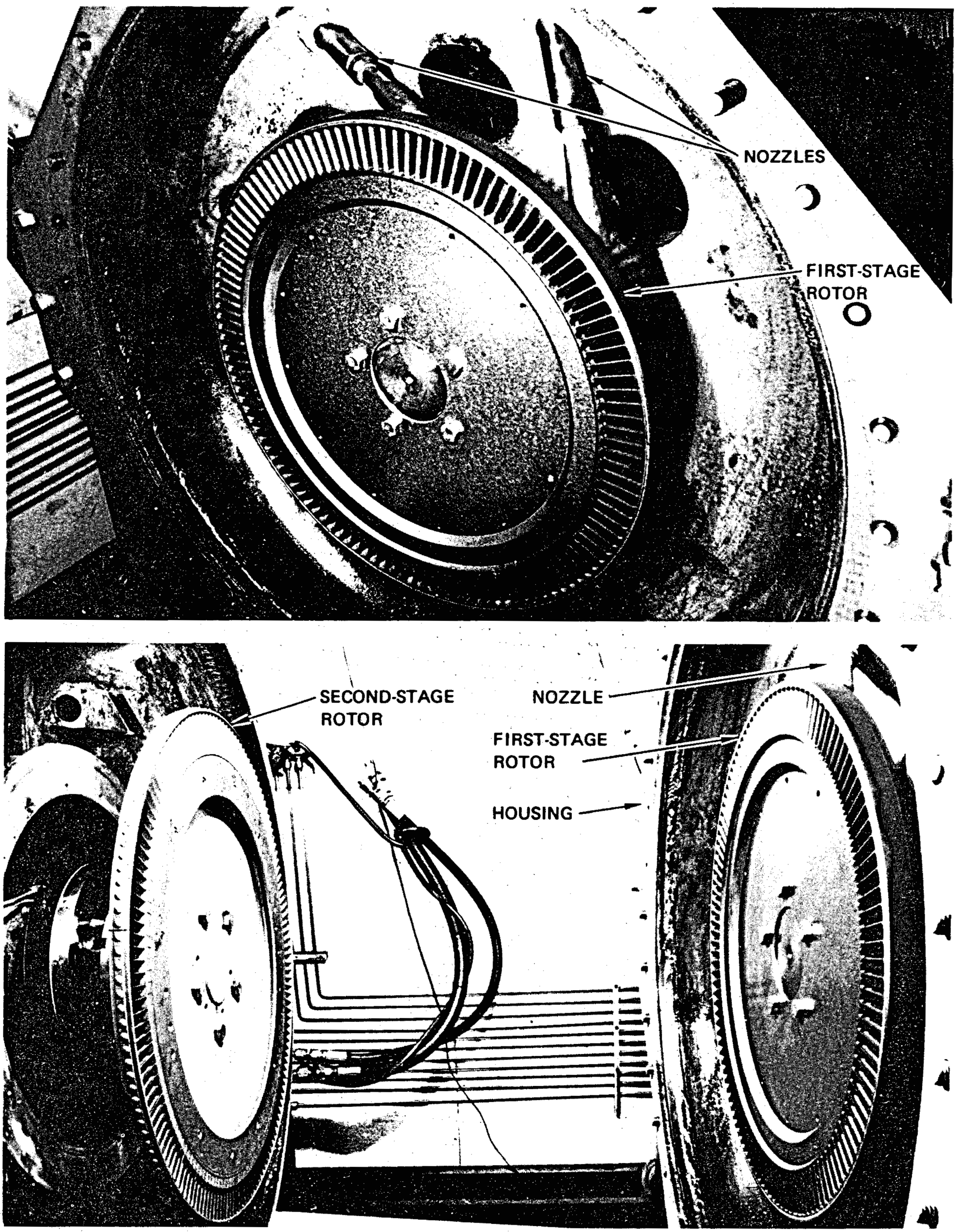

Figure 6. Rotors installed for steam-and-water testing at Biphase

IV. NOZZLE PERFORMANCE

Figure 7 is a photograph of the nozzles in operation with nitrogen-and-water. The nozzle operating conditions are given in Table 1, together with the operating conditions that were used for the Refrigerant 22 and steam-and-water tests. The Refrigerant 22 conditions are representative of conditions in closed-cycle two-phase turbines for waste-heat recovery; working fluids such as Refrigerant 113 or toluene would be used at elevated temperatures, but the velocities and qualities would be similar to those of Refrigerant 22. The water-and-steam conditions are representative of geothermal or hot-water energy-storage applications. The nitrogen-and-water conditions are only of interest for research purposes, permitting low-cost testing at low speeds.

The measured exit velocity $\overline{V}$ is the weighted average of the liquid and gas velocities measured as the ratio of nozzle thrust to flow rate. Using the gas/liquid velocity ratios calculated by the two-phase nozzle computer program (Ref. 2), the individual liquid and gas velocities can be estimated and the jet power calculated.

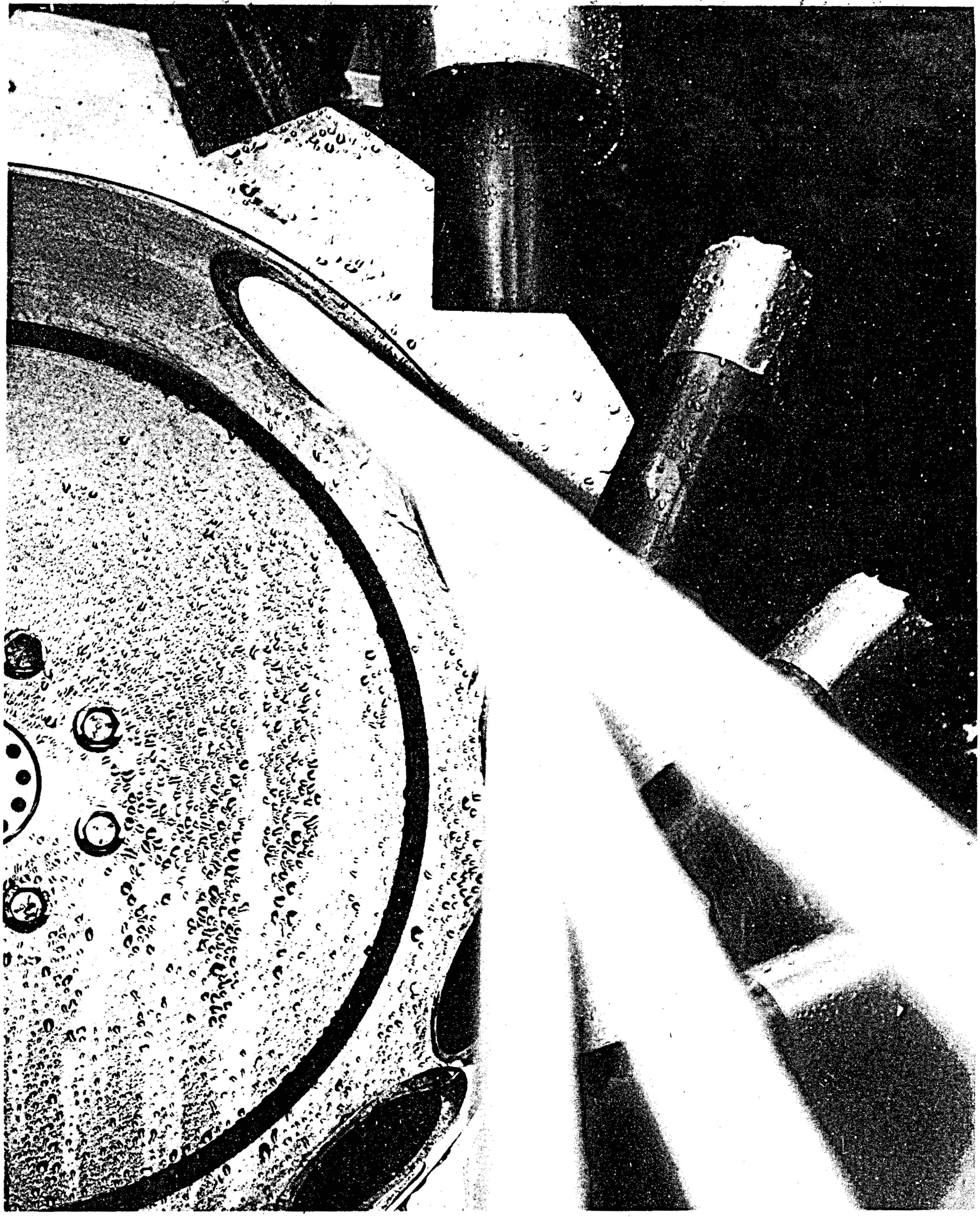

Figure 7. Nozzle flow with nitrogen-and-water

Table 1. Nozzle performance

Item	Working fluid: Nitrogen and Water	R-22	Steam and Water
Flow rate, kg/s	10.82	4.81	0.78
Inlet pressure, kPa	2000	840	2930
Inlet temperature, °C	17	17	230
Inlet quality, percent	1.45	2.0	3.0
Exit pressure, kPa	98	98	46
Isentropic velocity, m/s	107	131	517
Isentropic power, P_i, kW	61.9	41.3	104
Exit quality, percent	1.48	26.7	26.5
Measured exit velocity $\overline{V}$, m/s	94.3	123	457
Gas/liquid velocity ratio	1.46	1.16	1.57
Liquid velocity, m/s	93.7	118	397
Gas velocity, m/s	137.0	137	622
Jet power, P_n, kW	48.3	36.7	85.2
Efficiency P_n/P_i	0.78	0.89	0.82

V. TURBINE PERFORMANCE

A. Nitrogen-and-Water

The purpose of the nitrogen-and-water tests was to study the behavior of the liquid phase in a two-phase turbine. The nitrogen flow rate was only 1.5 percent of the water flow rate; thus almost all of the torque and power was due to the water. This permitted comparison with a computer model (Ref. 1) that calculates the ideal behavior of liquid passing through turbine rotors.

Figure 8 is a photograph of the turbine operating with nitrogen-and-water at peak-efficiency conditions. The water leaves the second-stage rotor at a position displaced about 25 degrees past the nozzles and at about a 45 degree forward angle. Smaller amounts of water escape between the nozzle and the first-stage rotor and (visible near the bottom) between the first-and second-stage rotors.

Theoretical efficiency curves for nitrogen-and-water calculated by the computer model for the Table 1 nozzle flow conditions are plotted in Fig. 9. Measured windage torques are used in the theoretical curves. The turbine efficiency is theoretically highest with first-stage rotor speeds between 2000 and 2400 rpm and a second-stage rotor speed of about 750 rpm. The highest theoretical efficiency is 0.65. The theoretical curves end at second-stage speeds of 750 rpm or less, because the liquid impinges on the backs of the second-stage rotor blades at higher speeds, and the model does not handle that case.

The measurements were made at a first-stage rotor speed of 2200 rpm. The lowest second-stage speed that could be held with the water brake was 740 rpm. However, the second stage could be locked to obtain a zero-speed point. The highest measured efficiency was 0.55 at a second-stage speed of 740 rpm. Efficiency decreased at higher and lower second-stage speeds, as shown in Fig. 9.

A few tests made at higher and lower first-stage speeds verified that the highest efficiencies occur at 2200 rpm.

Table 2 compares the theoretical and experimental efficiencies at a first-stage speed of 2200 rpm and second-stage speed of 740 rpm. The measured efficiency is 86 percent of the theoretical efficiency. The difference between the measured efficiency of 0.55 and the theoretical efficiency of 0.64 represents the gain that might be possible though improvements in liquid flow path in the turbine rotors; this should be an area of future research.

B. Refrigerant 22

The purpose of the Refrigerant 22 experiments was to demonstrate the highest possible efficiency with a two-phase turbine and also to investigate operation at practical flow conditions where the gas phase is a significant fraction. The velocity and quality obtained with Refrigerant 22 expanding

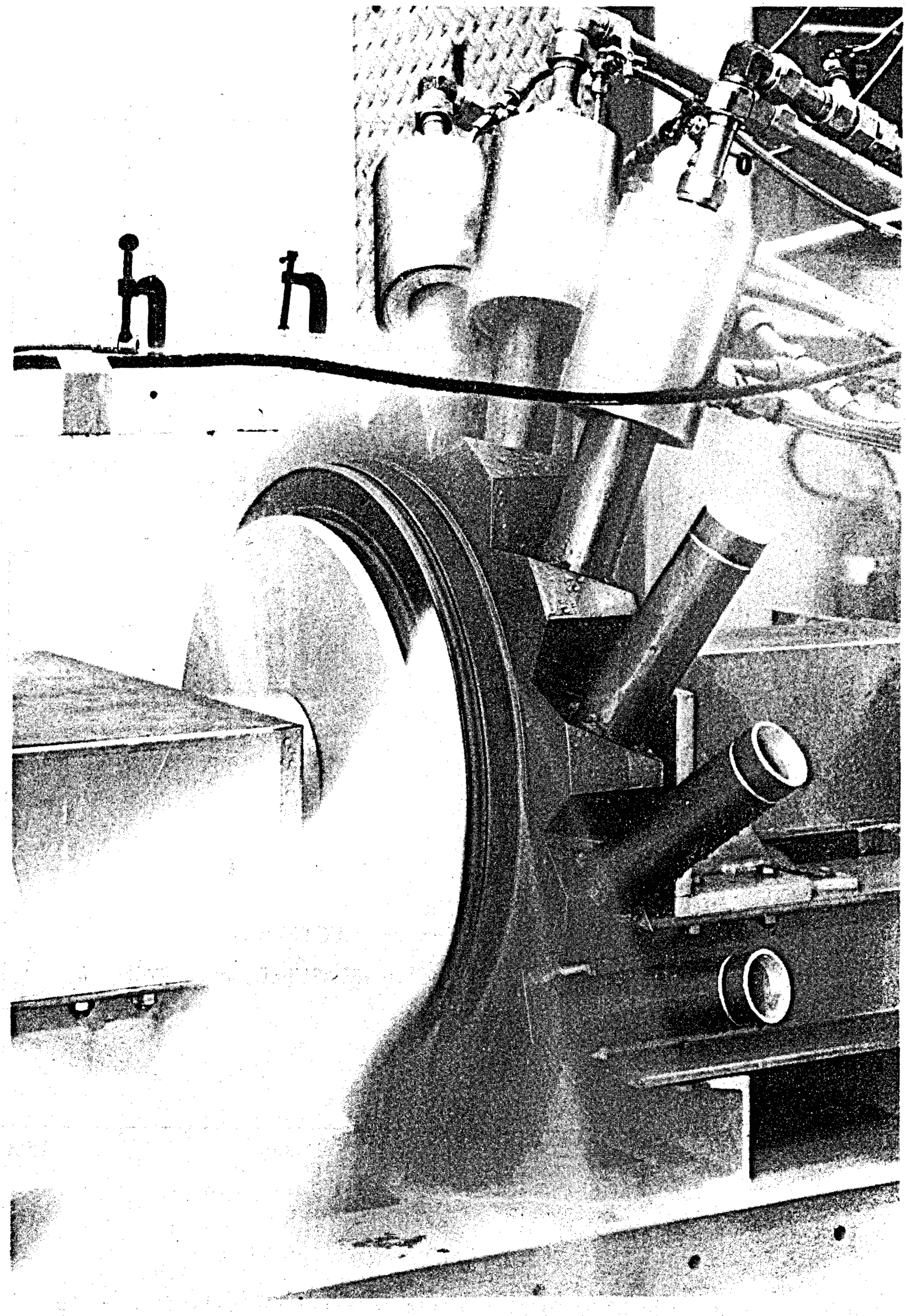

Figure 8. Turbine operating with nitrogen-and-water (Table 2 conditions)

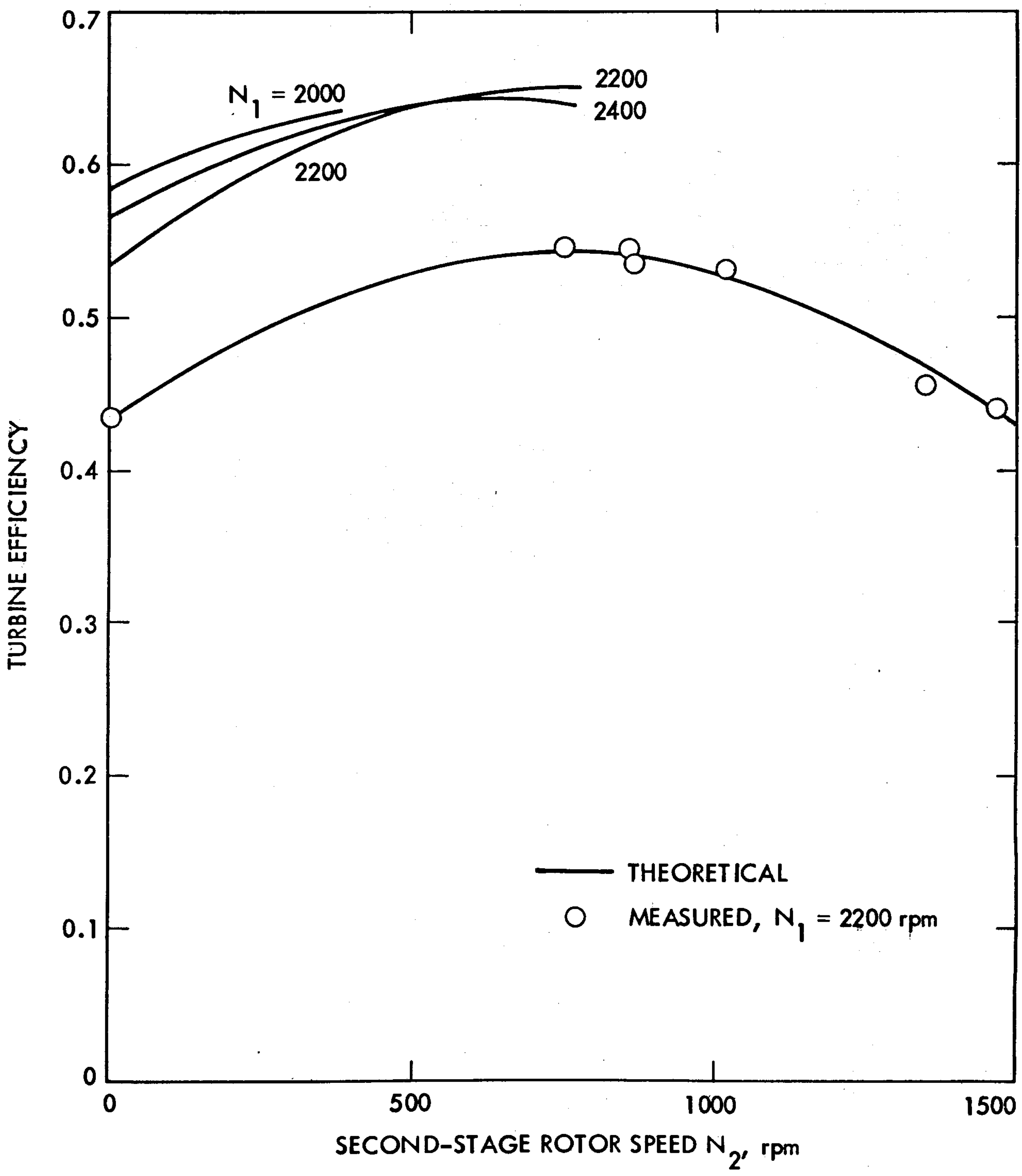

Figure 9. Comparison of theoretical and experimental efficiencies for nitrogen-and-water

Table 2. Turbine performance with nitrogen-and-water

Item	Experimental			Theoretical		
	First stage	Second stage	Total	First stage	Second stage	Total
Speed, rpm	2200	740		2200	740	
Torque, N-m	118	88		154	57	
Shaft power, P, kW	27.2	6.8	34.0	35.5	4.4	39.9
Rotor efficiency, P/Pn			0.70			0.83
Turbine efficiency, P/P_i			0.55			0.64

from saturation pressure at room temperature to atmospheric pressure (a ten-to-one pressure ratio) are similar to those obtained with heated organic working fluids in waste-heat and bottoming cycles. Because of the high molecular weight of organic fluids the efficiency of two-phase nozzles is relatively high, as shown in Table 1, leading to high turbine efficiencies. Theoretically, the highest efficiency of the turbine with Refrigerant 22 is 0.66 at a first-stage speed of 3000 rpm and a second-stage speed of 1100 rpm, as shown in Fig. 10.

The Refrigerant 22 feed tank permitted only about a 20-second run duration, during which I had to set the liquid throttling valve for a nozzle inlet pressure close to saturation, set the vapor flow valve for a flow rate near 2 percent, set the water-brake flow valve for a second-stage speed near 1100 rpm, and do this quickly enough to leave a few seconds for recording a steady data point. On the fourth run it was possible to achieve steady operation near the desired conditions. Figure 11 is a photograph of the turbine operating at that time. The forward angle of the flow leaving the second-stage rotor is steeper than in the nitrogen-and-water tests, and the liquid escaping between the first- and second-stage rotors is visible.

The measured turbine efficiency with Refrigerant 22 was 0.57. This is the highest efficiency achieved so far with a two-phase turbine.

Table 3 compares the theoretical and experimental efficiencies. The theoretical efficiency contains an arbitrary assumption of 0.70 efficiency for the gas phase; only the liquid behavior is calculated by the computer model. The measured efficiency is 86 percent of the theoretical efficiency.

C. Steam-and-Water

The efficiency of two-phase turbines with steam and water is severely limited by two effects: the low molecular weight of steam in the nozzles, which limits nozzle efficiency; and the large volume of steam at typical condensing pressures which forces the turbine blades to have a large area and the liquid to form very thin films with large velocity loss.

The theoretical efficiency of the turbine at the Biphase nozzle conditions is plotted in Figure 12, and the data points are shown. The maximum theoretical efficiency is 0.43 at a first-stage speed of 7500 rpm and a second-stage speed of 3500 rpm. This calculation assumes 70 percent efficiency for the steam and uses windage losses scaled from measurements in air at 3000-rpm first-stage speed.

The efficiency is limited by the large windage loss, equal to about 30 percent of the output power, at the small power available from the two nozzles used for the tests. If nozzles were mounted all the way around the turbine the theoretical efficiency would be 0.52.

The highest measured efficiency was 0.30, only 70 percent of the theoretical efficiency. Table 4 compares the theoretical and experimental performance at the conditions of highest measured efficiency.

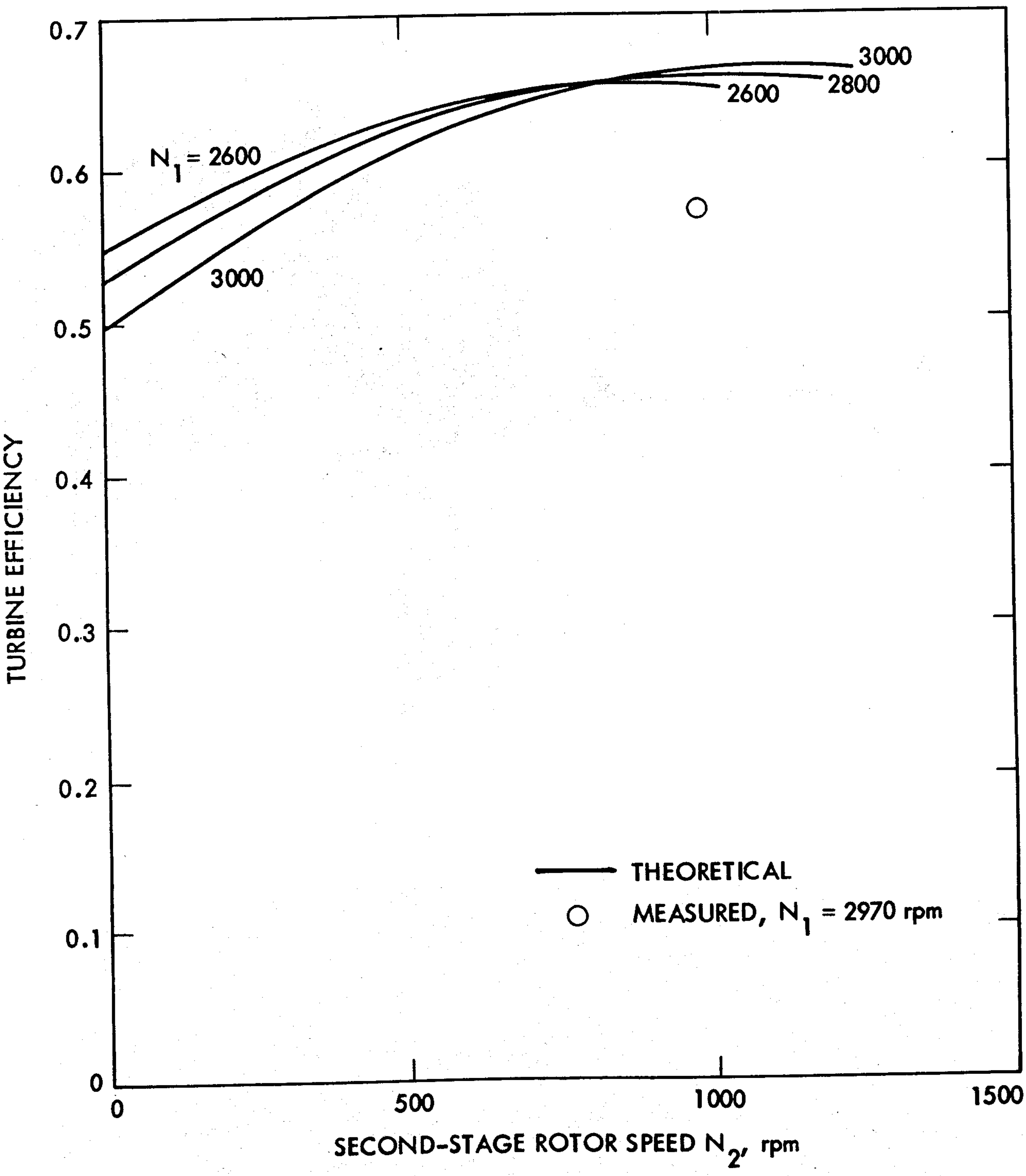

Figure 10. Comparison of theoretical and experimental efficiencies for Refrigerant 22

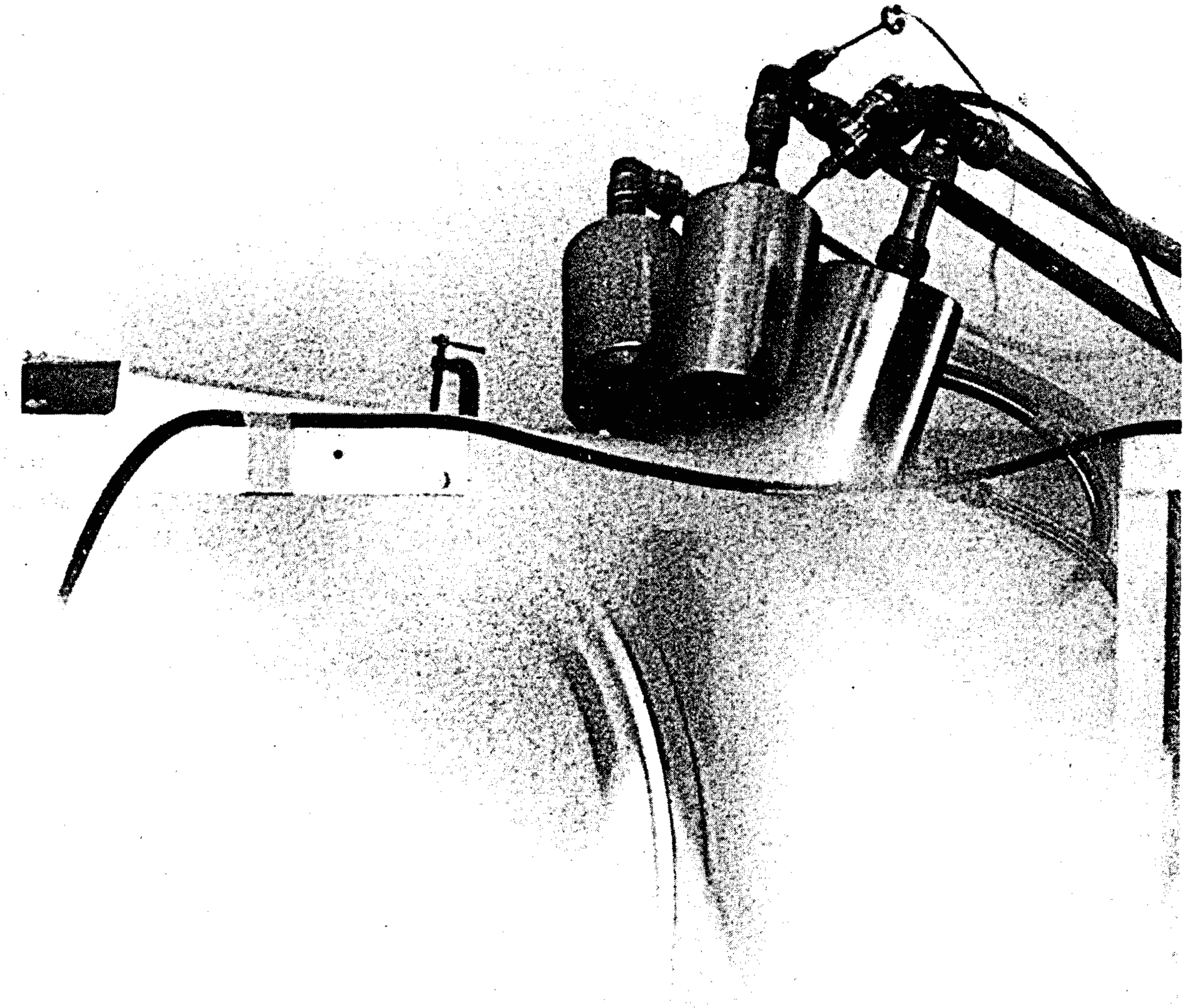

Figure 11. Turbine operating with Refrigerant 22 (Table 3 conditions)

Table 3. Turbine performance with Refrigerant 22

Item	Experimental			Theoretical		
	First stage	Second stage	Total	First stage	Second stage	Total
Speed, rpm	2970	980		2970	980	
Torque, N-m	53.3	68.6		68.8	57.0	
Shaft power, P, kW	16.6	7.0	23.6	21.4	5.8	27.2
Rotor efficiency, P/Pn			0.64			0.74
Turbine efficiency, P/P_i			0.57			0.66

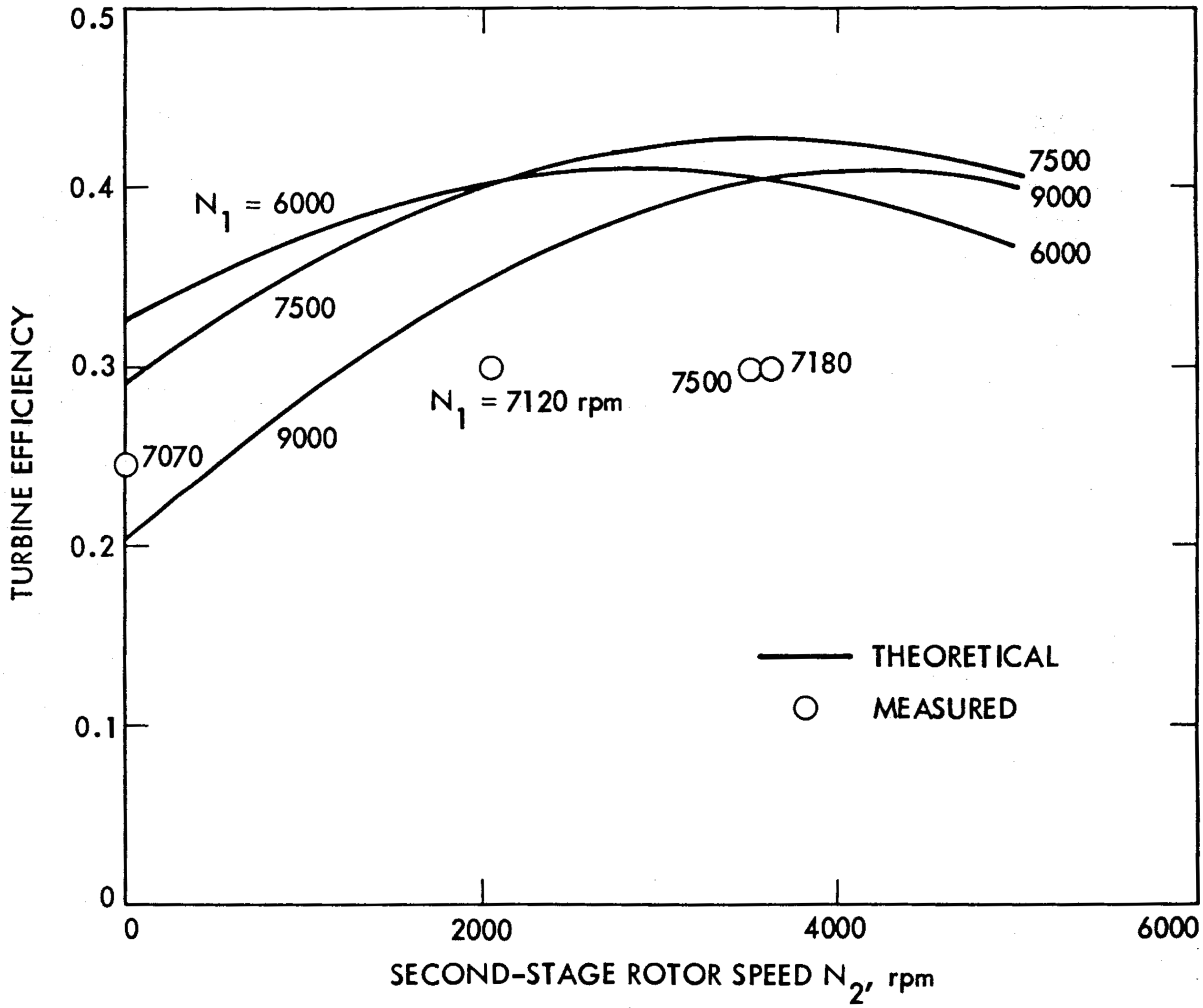

Figure 12. Comparison of theoretical and experimental efficiencies for steam-and-water

Table 4. Turbine performance with steam-and-water

Item	Experimental			Theoretical		
	First stage	Second stage	Total	First stage	Second stage	Total
Speed, rpm	7500	3520		7500	3520	
Torque, N-m	34.5	10		45.4	24.5	
Shaft power, P, kW	27.1	3.7	30.8	35.7	9.0	44.7
Rotor efficiency, P/Pn			0.36			0.52
Turbine efficiency, P/P_i			0.30			0.43

The test schedule did not permit investigating the reasons for the large difference between the theoretical and experimental efficiencies. Either the rotors performed poorly with steam-and water, or there may have been test problems such as splash-back of water into the rotor inside the closed housing.

VI. RECOMMENDED TURBINE DESIGN

In most applications it would be desirable for the flow to leave the turbine as separate liquid and gas streams. This could be accomplished by enclosing the rotors in a rotary separator, with the exit flow from the second-stage rotor impinging inside the separator drum. This type of separator has been used by Biphase Energy Systems in their two-phase turbine designs where the nozzle flow impinges inside a rotary separator.

The recommended two-phase turbine design is shown in Figure 13. The rotary separator collects the liquid leaving the second-stage rotor and also collects the liquid escaping between the nozzles and first-stage rotor and between the first- and second-stage rotors. The liquid forms a liquid layer inside the spinning separator drum. A scoop removes the liquid and delivers it to the liquid outlet of the turbine.

The scoop could incorporate a diffuser to recover liquid pressure. By taking less power from the rotors the diffuser exit pressure could be increased to the point where the liquid could be returned to the nozzle inlets, permitting closed-loop operation without a liquid pump after starting.

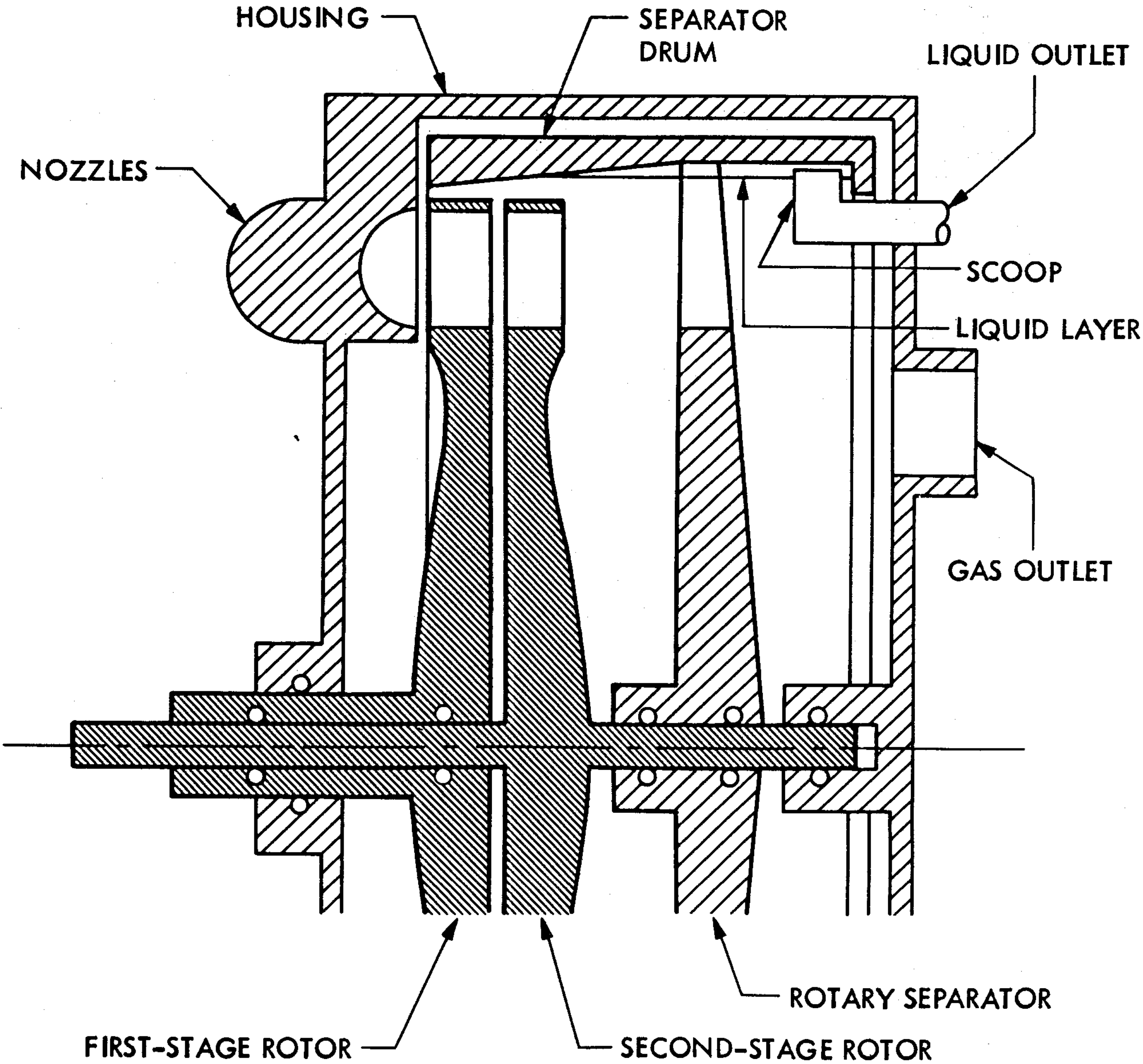

Figure 13. Recommended two-phase turbine design

VII. CONCLUSION

The results of the two-stage, two-phase turbine tests showed that efficiencies can be achieved that are about 86 percent of theoretical with nitrogen-and-water mixtures and with organic working fluids. The same is probably also true of steam-and-water although this was not achieved in the tests. Based on theoretical efficiencies calculated for optimized turbines, (Ref. 1, Fig. 44), the attainable efficiencies of two-stage two-phase turbines would be in the low 60 percent range for organic fluids and in the mid 50 percent range for steam-and-water. With improvements in the liquid flow path, the efficiency of two phase turbines could reach the mid 60 percent range with organic fluids and the high 50 percent range with steam-and-water.

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