NASA Contractor Report 180885

Study of Toluene Rotary Fluid Management Device and Shear Flow Condenser Performance for a Space-Based Organic Rankine Power System

(NASA-CR-180885) STUDY OF TOLUENE ROTARY FLUID MANAGEMENT DEVICE AND SHEAR FLOW CONDENSER PERFORMANCE FOR A SPACE-BASED ORGANIC RANKINE POWER SYSTEM Final Report (Sundstrand Energy Systems) 103 p CSCL 10B G3/20

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ACRONYM LIST

NASA	National Aeronautics and Space Administration
NC	Noncondensible (e.g. noncondensible gases)
OD	Outside Diameter
ORC	Organic Rankine Cycle
PLR	Parasitic Load Radiator
RFMD	Rotary Fluid Management Device
SD	Solar Dynamic
SD-ORC	Solar Dynamic Organic Rankine Cycle
TPTMS	Two-Phase Thermal Management System
WP02	Work Package-02 (Johnson Space Center)

1.0 Summary

Management of two-phase fluid and control of the two-phase heat transfer process in microgravity is a technical challenge that must be addressed for an orbital Organic Rankine Cycle (ORC) application. A test program was performed that satisfactorily demonstrated the two-phase management capability of the RFMD and shear flow condenser in a simulated ORC test bed. Operational tests of the RFMD and shear flow condenser in adverse gravity orientations confirmed that the centrifugal forces in the RFMD and the shear forces in the condenser were capable of overcoming the adverse gravity forces of the ground environment. Successful operation in an adverse gravity environment demonstrated that the centrifugal and shear forces are capable of controlling the two-phase fluid interfaces. In a micro-gravity environment these same forces would not have to compete against gravity and would be dominant, thus indicating the RFMD and condenser would be capable of maintaining control of the two-phase fluid interfaces in the ORC heat rejection zone. The specific test program performed covered the required operating range of the Space Station Solar Dynamic Organic Rankine Cycle power system. Review of the test data verified that the system successfully performed all of the critical functions of the planned experiment including: fluid was pumped from the RFMD in all attitudes, sub-cooled liquid states were achieved in the condenser, condensate was pushed uphill against gravity forces, and noncondensible gases were swept through the condenser to be accumulated in the RFMD for periodic overboard venting.

A series of tests were performed with the key components (RFMD and condenser) horizontal. Review of the data shows that the measured

component performance was comparable to the pre-test predicted performance for all test conditions. The RFMD hydraulic performance was 25% lower than predicted. This was overcome by operating at higher speeds and a deeper liquid annulus in the rotating drum. The measured RFMD motor power was 4% lower than predicted at design point. The condenser pressure drop was measured to be 7% less than predicted at design flow. Detailed analysis of the exact experimental conditions measured during a given test was not performed. Pre-test predictive models were provided for comparison purposes only. No assessments were made on the model adequacy. Detailed analysis and model assessment would be the subject of a follow-up program.

2.0 Introduction

The Heat Engine Technology Development Program was initiated August 29, 1985 under Contract No. NAS3-24663 as part of the Space Station Advanced Development Program. The contract was awarded to Sundstrand Energy Systems by NASA-Lewis Research Center. Task 2 of the program, reported in this document, consisted of the design, fabrication, instrumentation, calibration, assembly, checkout and operation of an organic Rankine cycle (ORC) condensation and two-phase fluid management concept for potential application to the Space Station Solar Dynamic Power Generation System.

The primary technical objective of this test program was to demonstrate that the Rotary Fluid Management Device (RFMD) and shear flow condenser are a viable approach to two-phase fluid management in micro-gravity ORC applications. A secondary objective was to compare results from

Sundstrand's analytical methods to experimental results. To achieve the technical objectives, a laboratory ground test rig was designed and constructed using toluene as the working fluid. Major components in the rig included a Rotary Fluid Management Device (RFMD), a shear flow condenser, an inventory control accumulator, and a vaporizer that simulated the power generation portion of an ORC system. The basis for all of the component design and sizing was the Sundstrand Solar Dynamic Organic Rankine Cycle Electric Power Generation System for the Space Station, developed during the Phase B portion of the Space Station Program.

3.0 Hardware Description

3.1 Loop Description

The Solar Dynamic Organic Rankine Cycle (SD-ORC) is a candidate for the Power Generation System that will convert solar energy into electrical power for the Space Station. The Rocketdyne Space Station Phase B concept consists of an offset parabolic concentrator that focuses solar energy on a receiver.[6] The receiver absorbs the solar energy as heat. The power conversion unit (or heat engine) converts the thermal energy into electrical power. A heat pipe radiator rejects waste heat from the system. A functional block diagram of the SD-ORC is shown in Figure 1.

The heart of the SD-ORC is the heat engine that converts thermal energy into electrical power. This is accomplished by pumping a supercritical working fluid (toluene) through the solar receiver to absorb heat. Superheated toluene exits the receiver and drives a single-stage axial

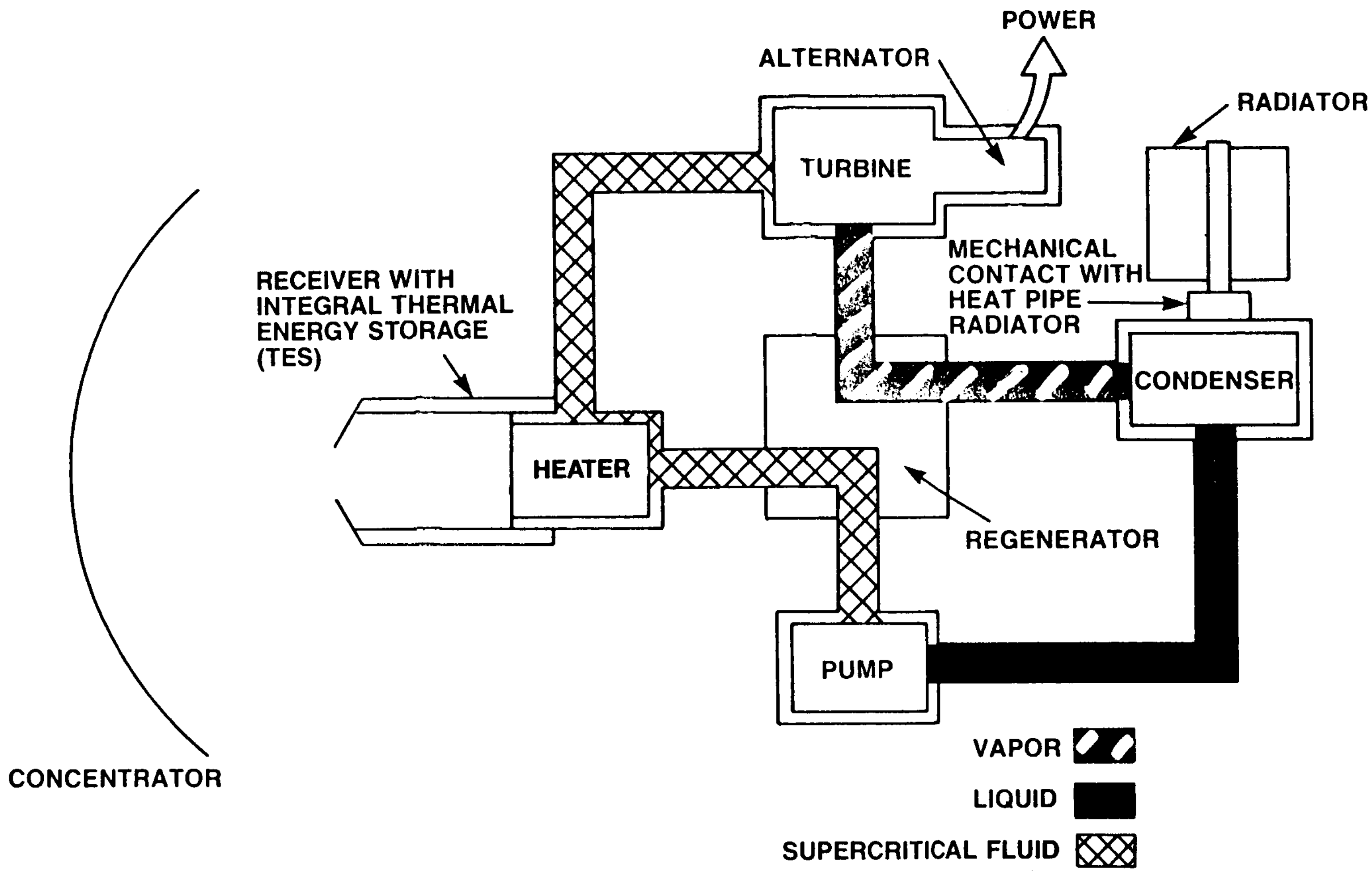
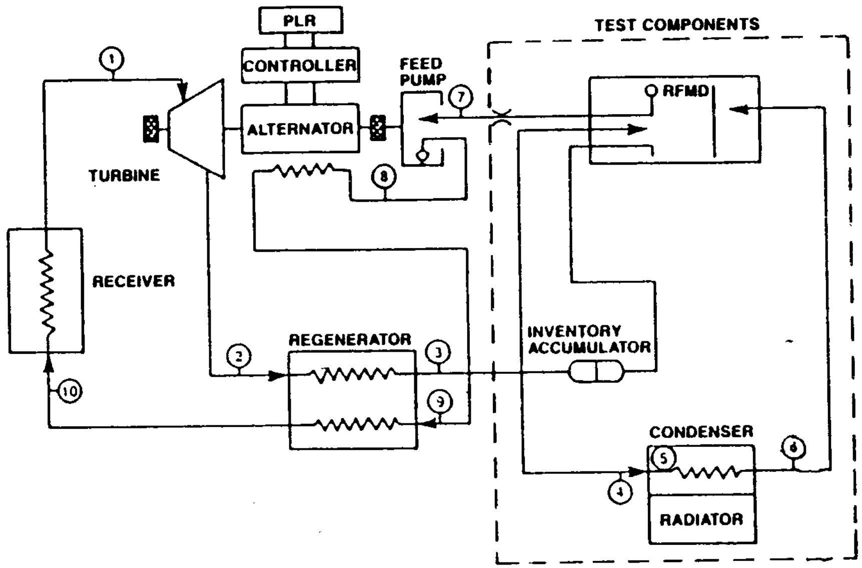


FIGURE 1 - SD-ORC FUNCTIONAL BLOCK DIAGRAM

impulse turbine. An alternator is integrally mounted to the turbine shaft and provides electrical power to the user. The balance of the system components condense and pressurize the working fluid to complete the cycle. The nature of a Rankine cycle dictates that vapor and liquid phases are present. Due to the selection of a supercritical heat addition process, the only two-phase interfaces in the SD-ORC are in the heat rejection zone. Demonstration of the effective control of the two-phase fluid interfaces in the ORC heat rejection zone is the technology challenge being addressed in this project.

Figure 2 shows the entire SD-ORC block diagram and state points, with the ORC heat rejection zone and the components that were tested in this project identified. The major components of the experimental program included a Rotary Fluid Management Device (RFMD), a shear flow-controlled condenser, and an inventory control accumulator, as shown in Figure 3. These three components were designed and fabricated to be functionally prototypic of the Space Station SD-ORC system. The SD-ORC state points and flow rates were to be exactly duplicated representing a 25 Kwe engine simulation. The maximum SD-ORC heat rejection is 113 Kw_{th}. The test loop was designed for 150 Kw_{th} capacity. The balance of the SD-ORC cycle was simulated by test facility support equipment as shown in Figure 4. A pumped electrical resistance heated oil loop was used to vaporize the liquid toluene from the RFMD. The outlet conditions of the vaporizer were controlled to match the state point of the regenerator vapor outlet conditions of the SD-ORC. The heat pipe radiator and condenser interface were replaced by a water cooling jacket to remove waste heat from the condenser. The RFMD and toluene condenser were mounted on support

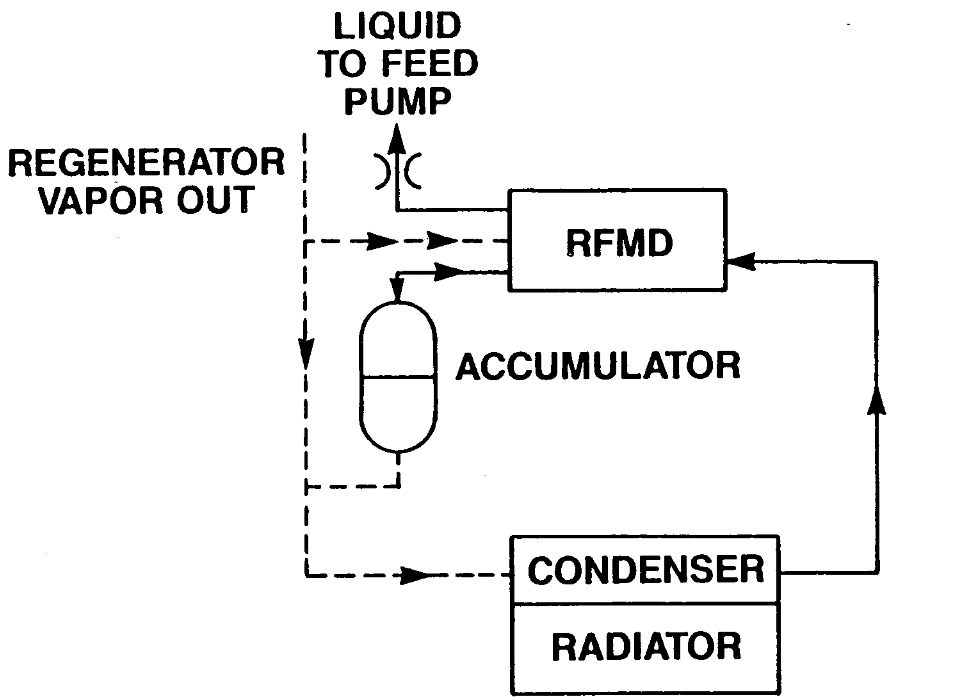


STATE POINTS

Location	Description	Temperature °C	°F	Pressure kPa	psia	Mass Flow kg/s	lb/s
1	Turbine Inlet	674	755.0	4233	614.00	.198	.436
2	Regenerator Vapor In	560	548.8	21.3	3.09	.198	.436
3	Regenerator Vapor Out	353	176.2	20.3	2.94	.198	.436
4	Condenser Vapor In	353	176.2	16.8	2.44	.188	.415
5	Condenser Saturated Vapor	331	135.7	16.8	2.44	—	—
6	Condenser Exit	307	92.5	7.3	1.06	.188	.415
7	Feed Pump Inlet	329	132.4	15.6	2.27	.198	.436
8	Feed Pump Exit	335	143.0	4944	717.10	.198	.436
9	Regenerator Liquid In	341	154.6	4888	709.10	.198	.436
10	Regenerator Liquid Out	504	448.1	4654	675.00	.198	.436

FIGURE 2 - ORC SCHEMATIC AND STATE POINTS

SPACE STATION SD-ORC



<u>FIGURE 3</u> - HEAT REJECTION SYSTEM FUNCTIONAL DIAGRAM

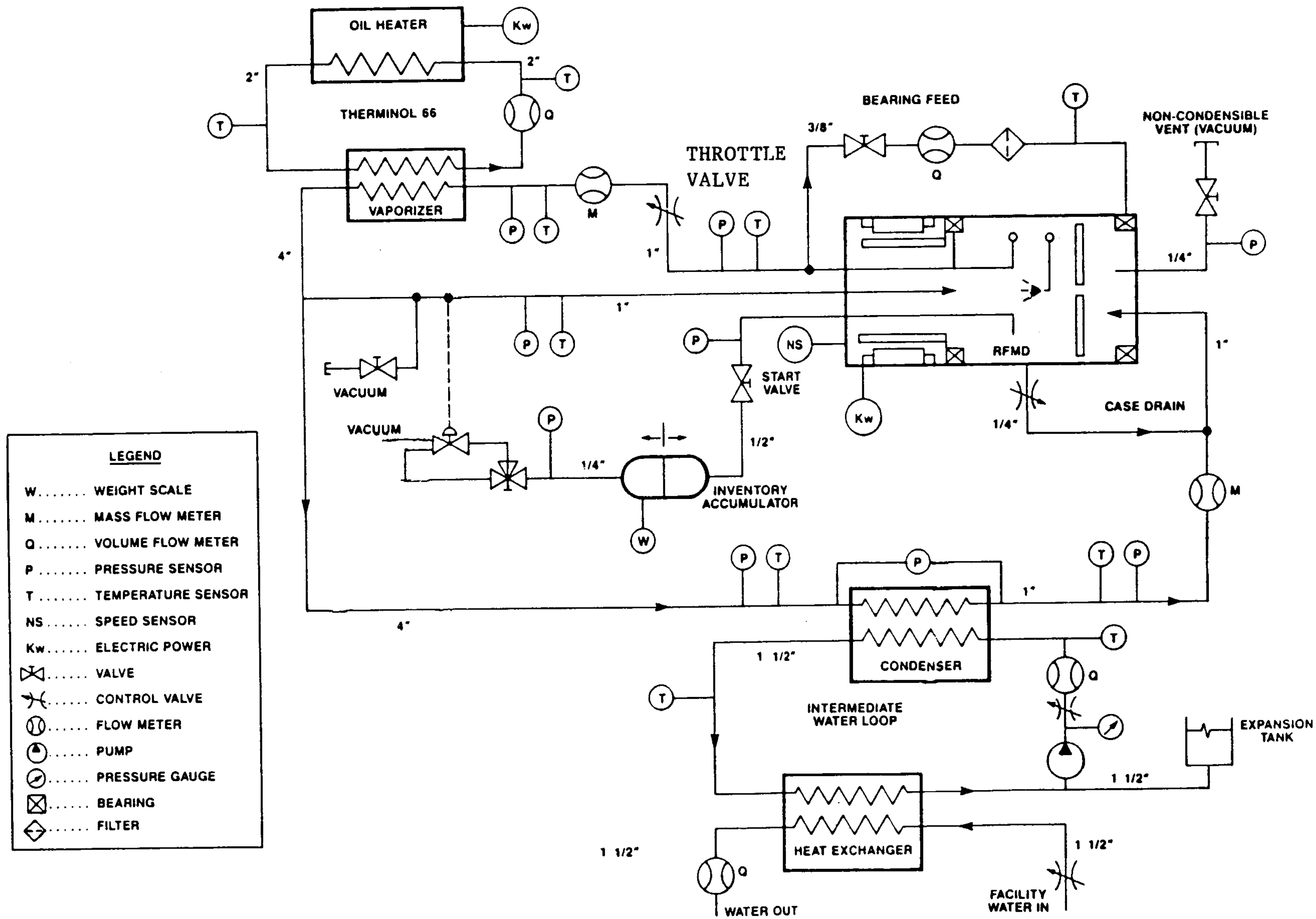


FIGURE 4: TOLUENE RFMD AND CONDENSER TEST LOOP SCHEMATIC

structures that allow changing the orientation of each respective component to achieve positive, neutral, and adverse gravity effects.

The Task 2 test stand was constructed and operated in the Sundstrand Plant 6 test lab, Cells 61 and 62. The toluene system is located in Cell 62. A floor plan of Cell 62 is shown in Figure 5 and identifies the key components. The toluene stability loop test stand constructed for Task 1 of the subject contract is also located in Cell 62. Figures 6 and 7 are photos of the Task 2 components in Cell 62. The controls and data acquisition systems were located in an adjacent room, Cell 61. Figure 8 is a photo of the analog recorder and control console constructed for Task 2. Figure 9 is the data acquisition system and printer.

A primary consideration in designing the support structures for the RFMD and the condenser was to allow the components to be rotated and operated in neutral gravity, adverse gravity and gravity-aided orientations.

RFMD adverse gravity capability can be demonstrated by changing the rotational axis of the RFMD from horizontal to vertical. Operation in the vertical orientation is the most severe adverse gravity orientation. This is because the pumping pitot is located at the upper end of the rotating drum in the vertical position. Insufficient adverse gravity capability would be indicated by a deterioration in pump performance due to the liquid annulus "slumping" to the bottom of the rotating drum. Verification that RFMD pitot pump performance is unaffected by the external forces of gravity would indicate that RFMD performance in a micro-gravity environment would also be unaffected. The RFMD was mounted

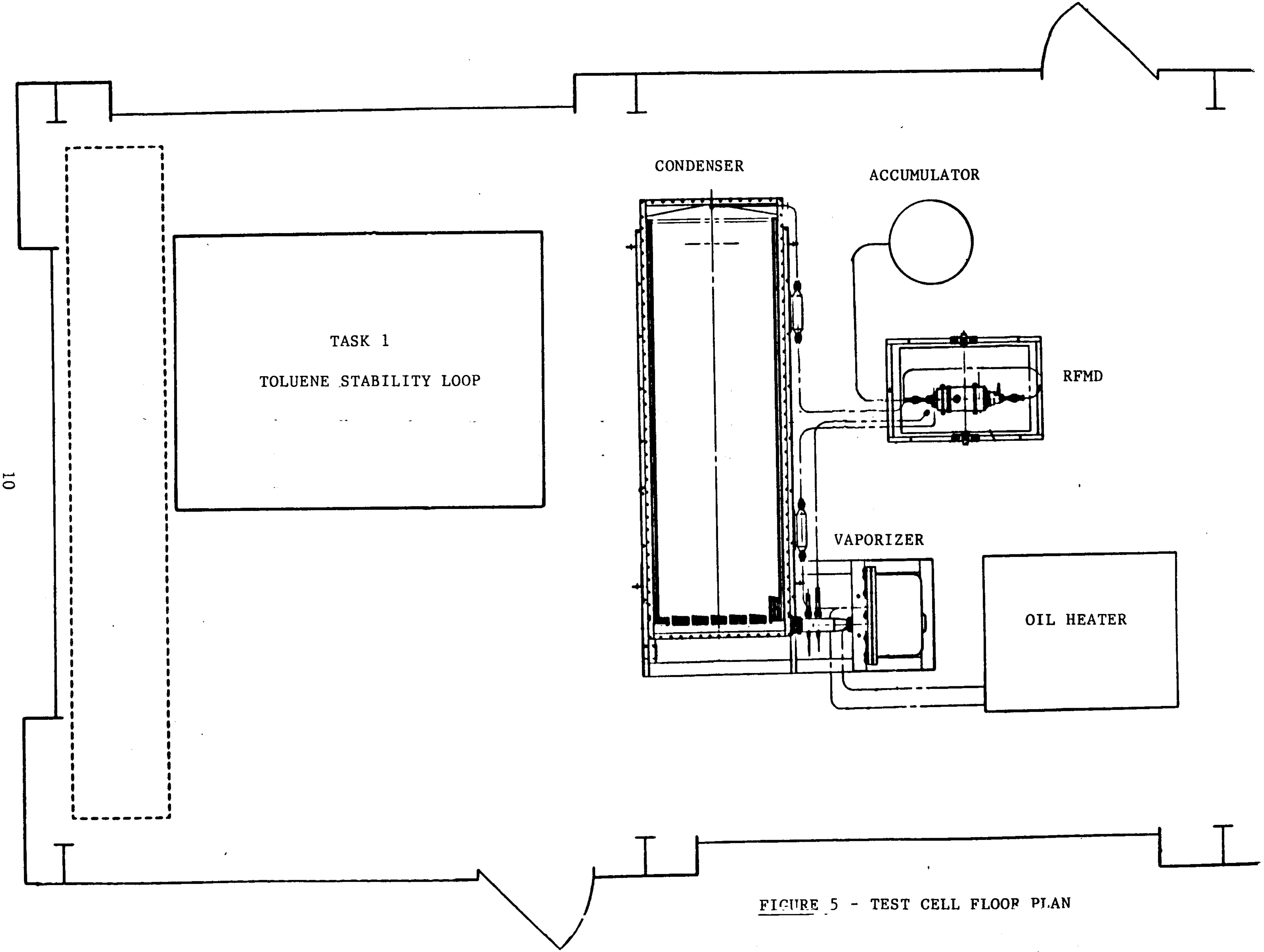


FIGURE 5 - TEST CELL FLOOR PLAN

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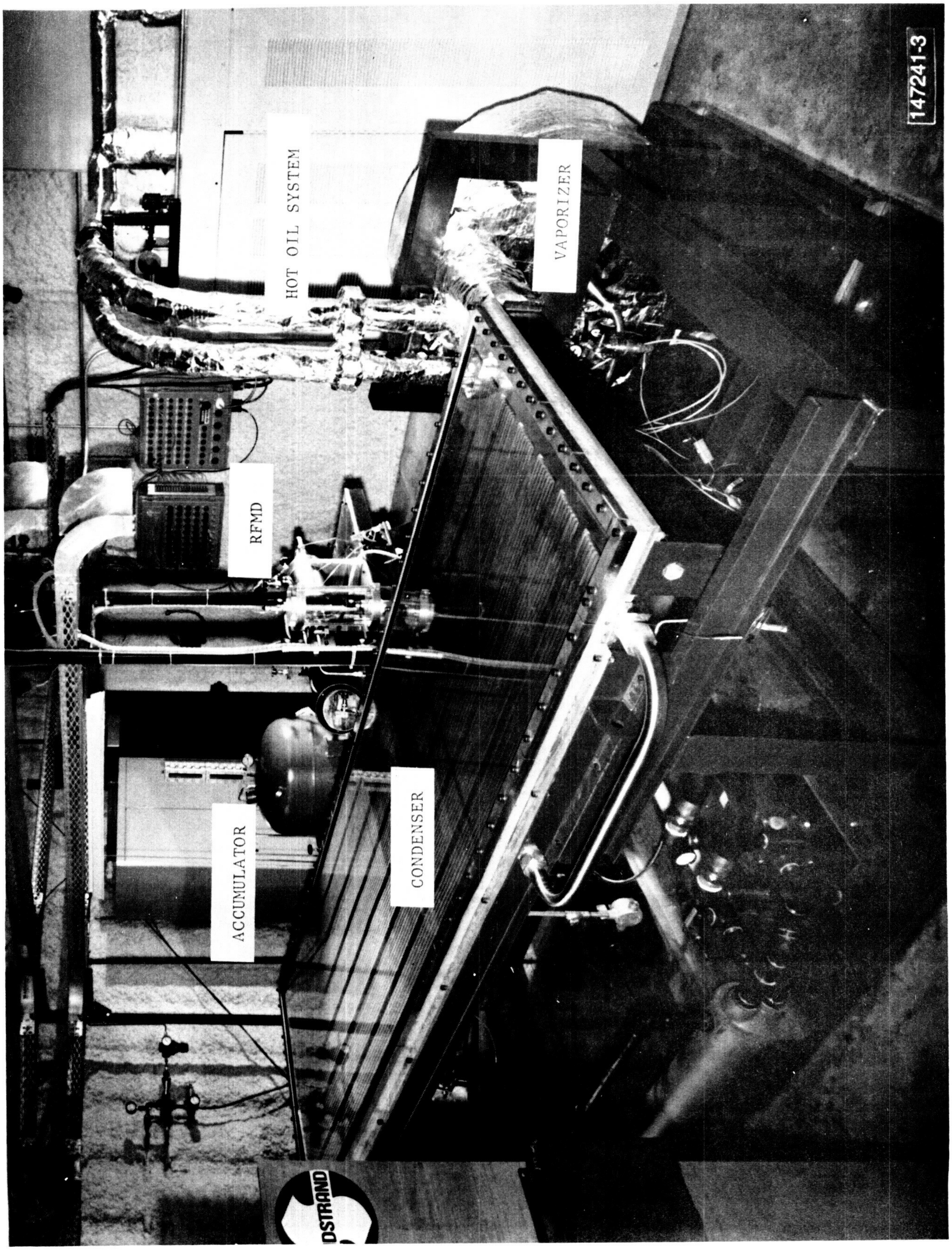

FIGURE 6: CELL 62, VIEW 1

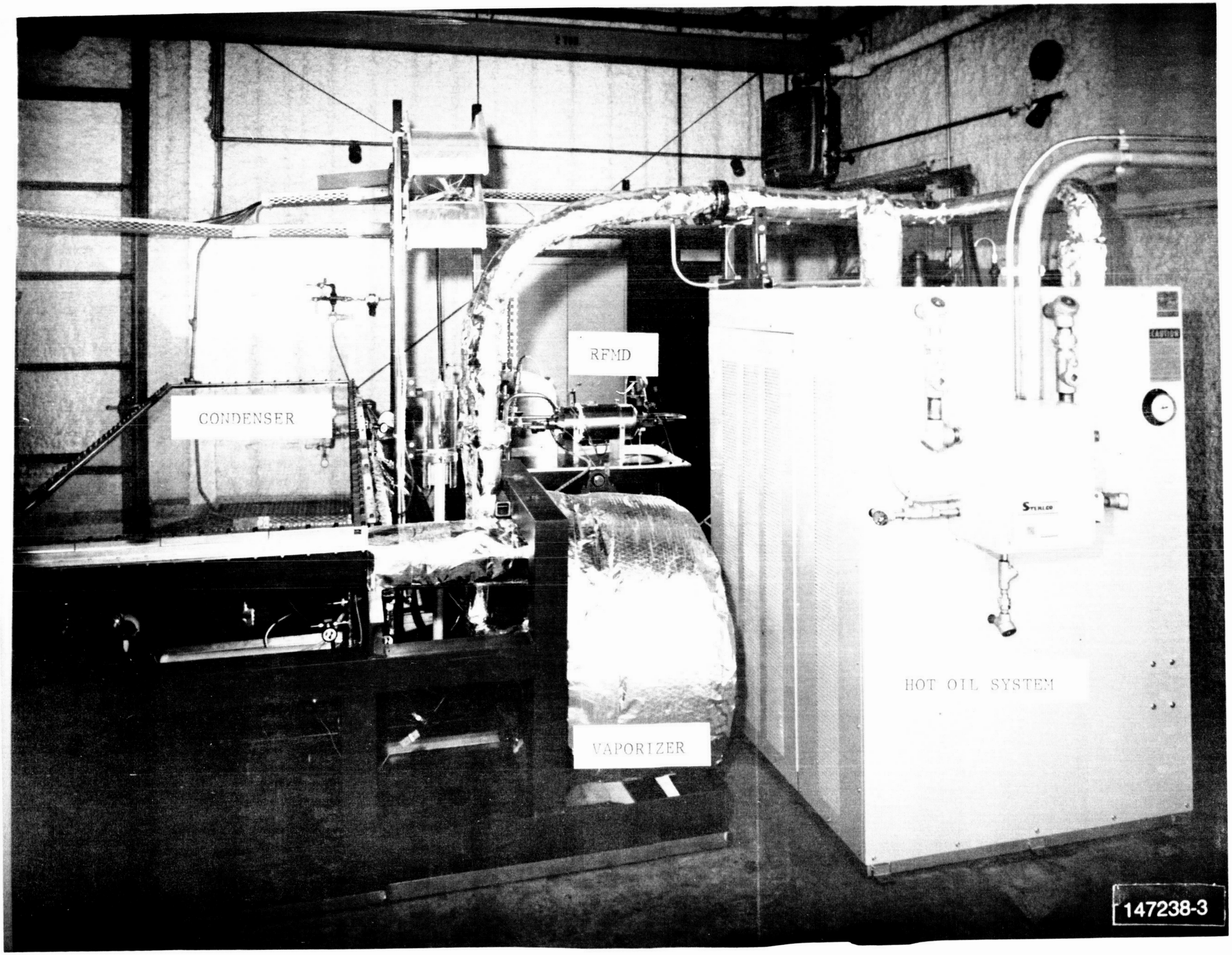

FIGURE 7 - CELL 62, VIEW 2

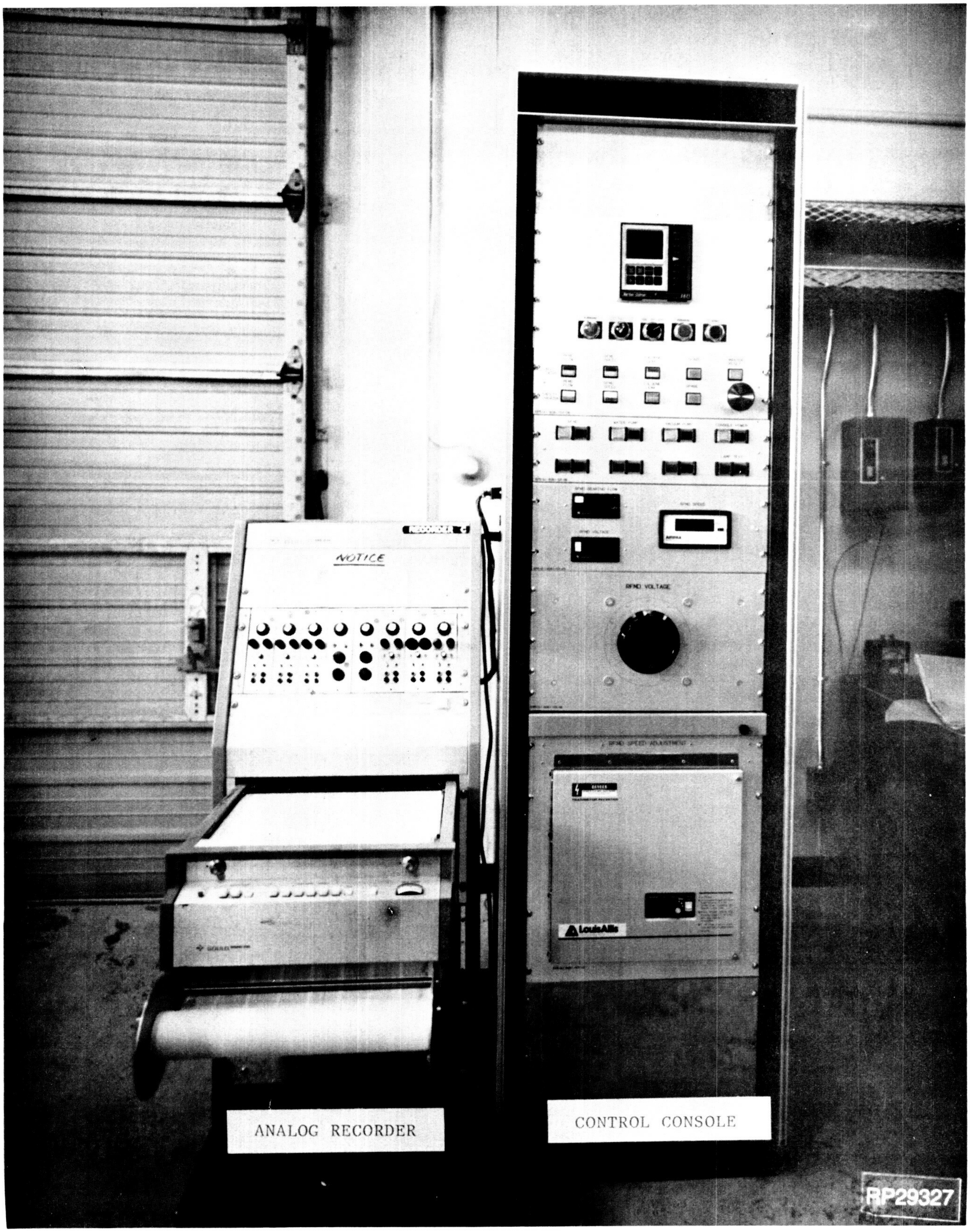

FIGURE 8 - CONTROL CONSOLE

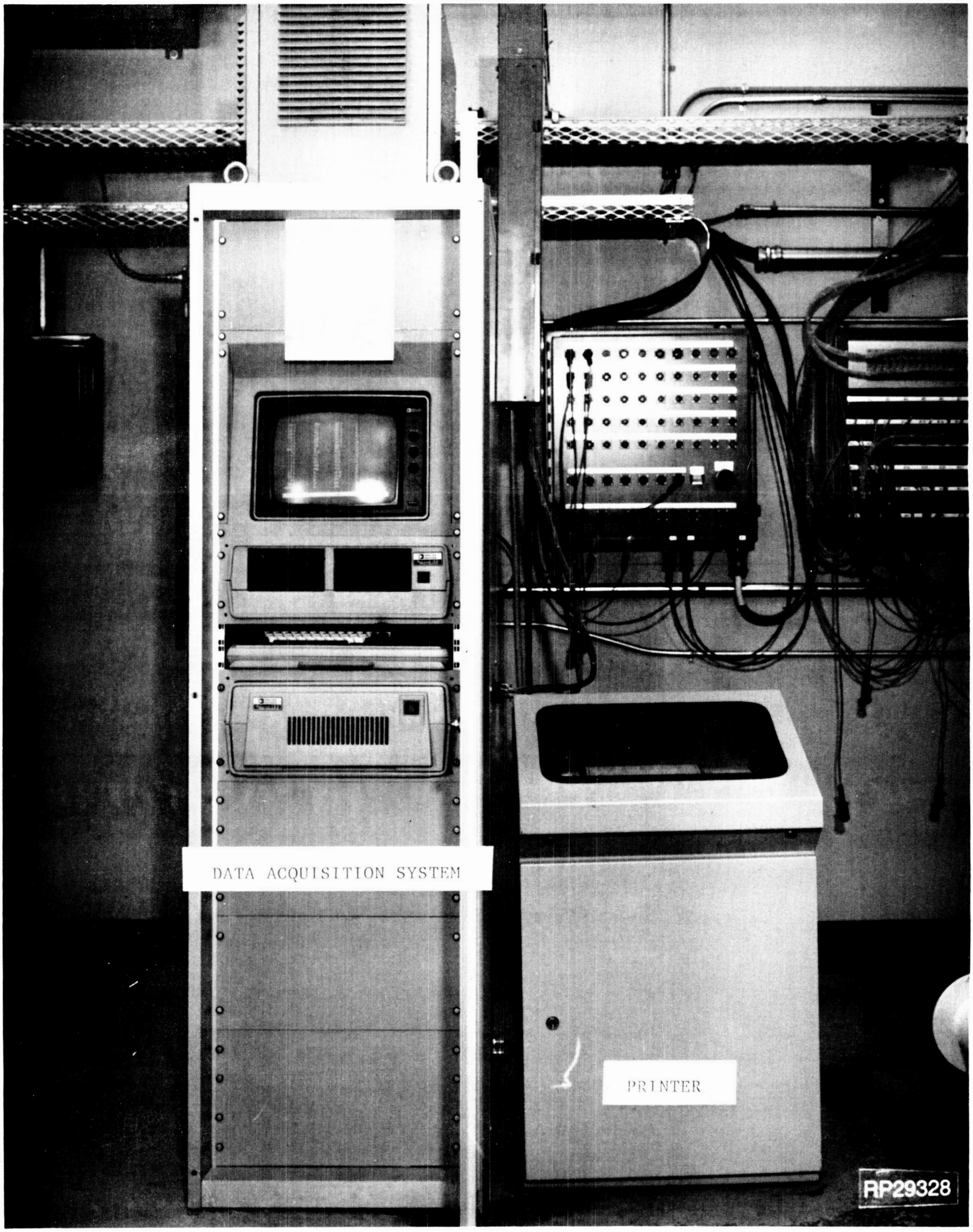

FIGURE 9 - DATA ACQUISITION SYSTEM

on the support structure shown in Figure 10. The rotational axis of the RFMD could be positioned horizontally to the floor as shown (neutral gravity) or perpendicular to the floor (adverse gravity). Figures 11 and 12 show the RFMD support structure in the horizontal and vertical attitudes, respectively. Flex hose connections between the RFMD and condenser structures allowed for changing component orientations without having to change any plumbing or shutdown the test stand.

Condenser adverse gravity capability would be demonstrated by stable condenser operation at uphill condenser inclinations. Insufficient adverse gravity capability would be represented by liquid flowing downhill, against the normal flow direction of the condensing vapor. Liquid run-back would result if the gravity forces due to the inclination of the condenser are greater than the shear forces in the condensing channels. The absence of liquid run-back would verify that the shear forces are greater than the gravity forces in the direction of flow. This would indicate that the shear forces acting on the fluid would dominate as well in a micro-gravity environment. The condenser and vaporizer were mounted on a common support structure shown in Figure 13. There is no relative movement between the vaporizer and the condenser. The structure was designed to rotate around a pivot foot on the floor, see Figure 14. The toluene flow in the condenser channels could be subjected to positive, neutral, or adverse gravity forces by varying the slope of the support structure.

Further information on the design, construction and checkout of the test rig are on file at Sundstrand.

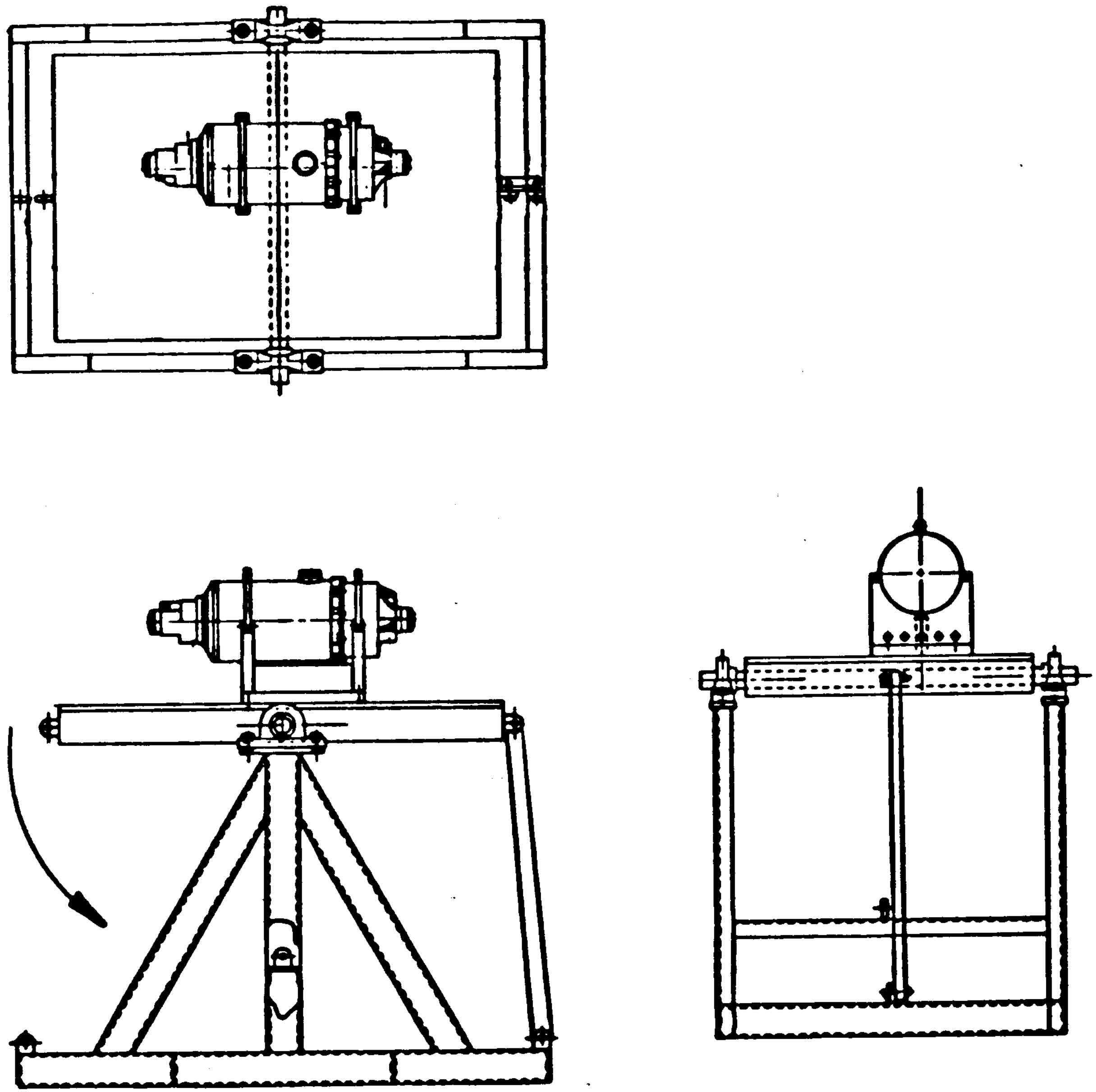

FIGURE 10 - RFMD SUPPORT STRUCTURE

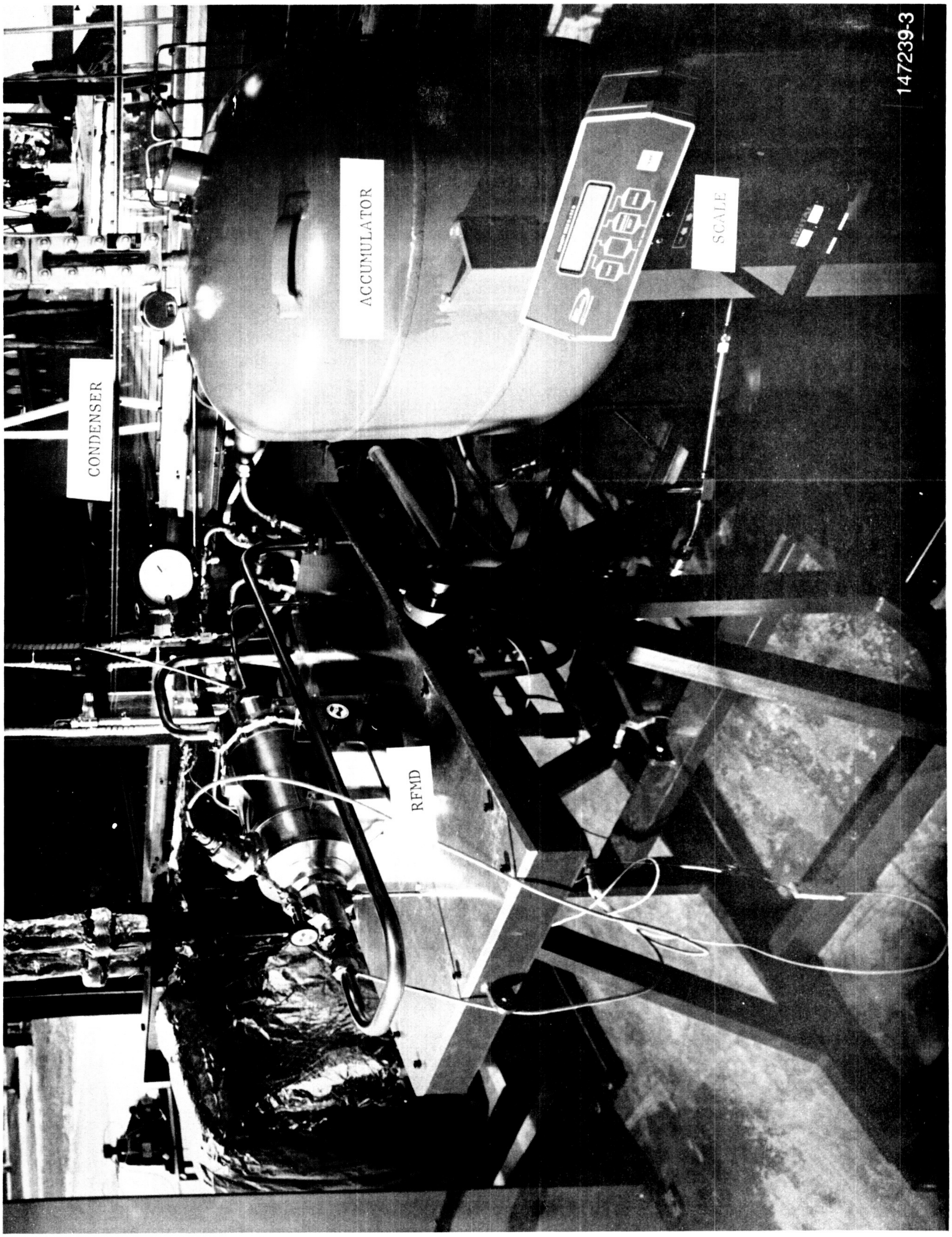

FIGURE 11 - RFMD IN HORIZONTAL POSITION

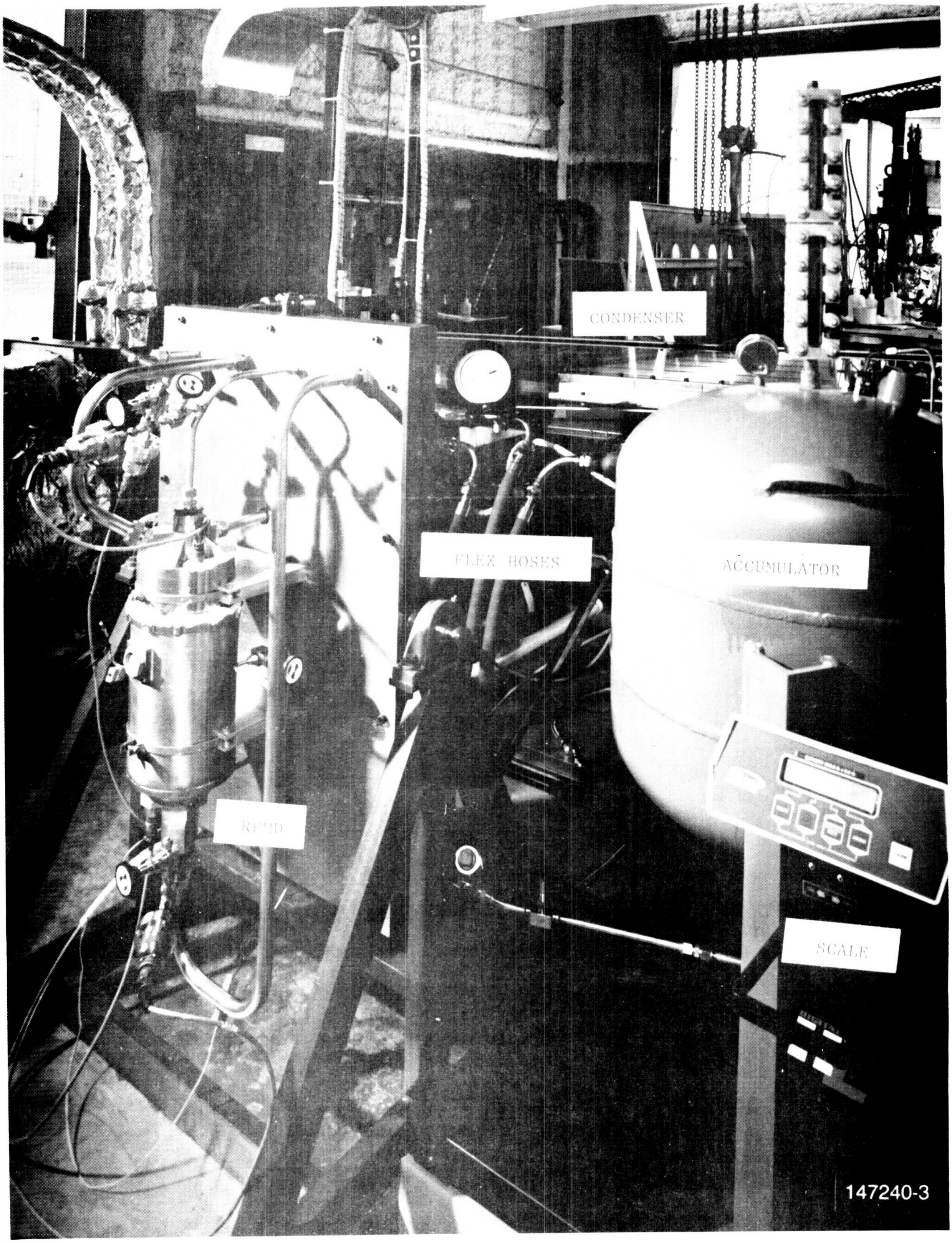

FIGURE 12 - REMD IN VERTICAL POSITION

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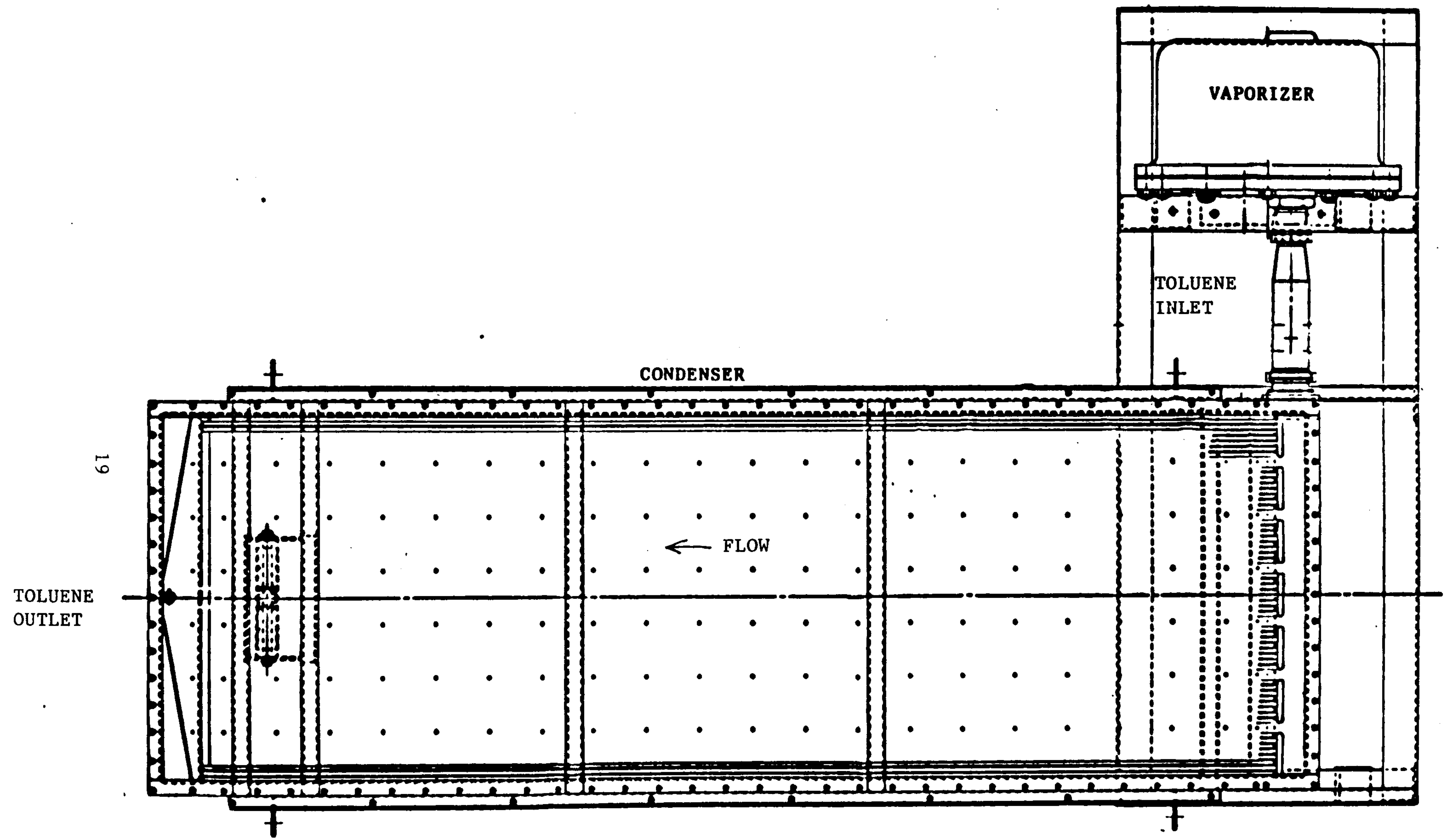

FIGURE 13 - CONDENSER/VAPORIZER STRUCTURE-TOP VIEW

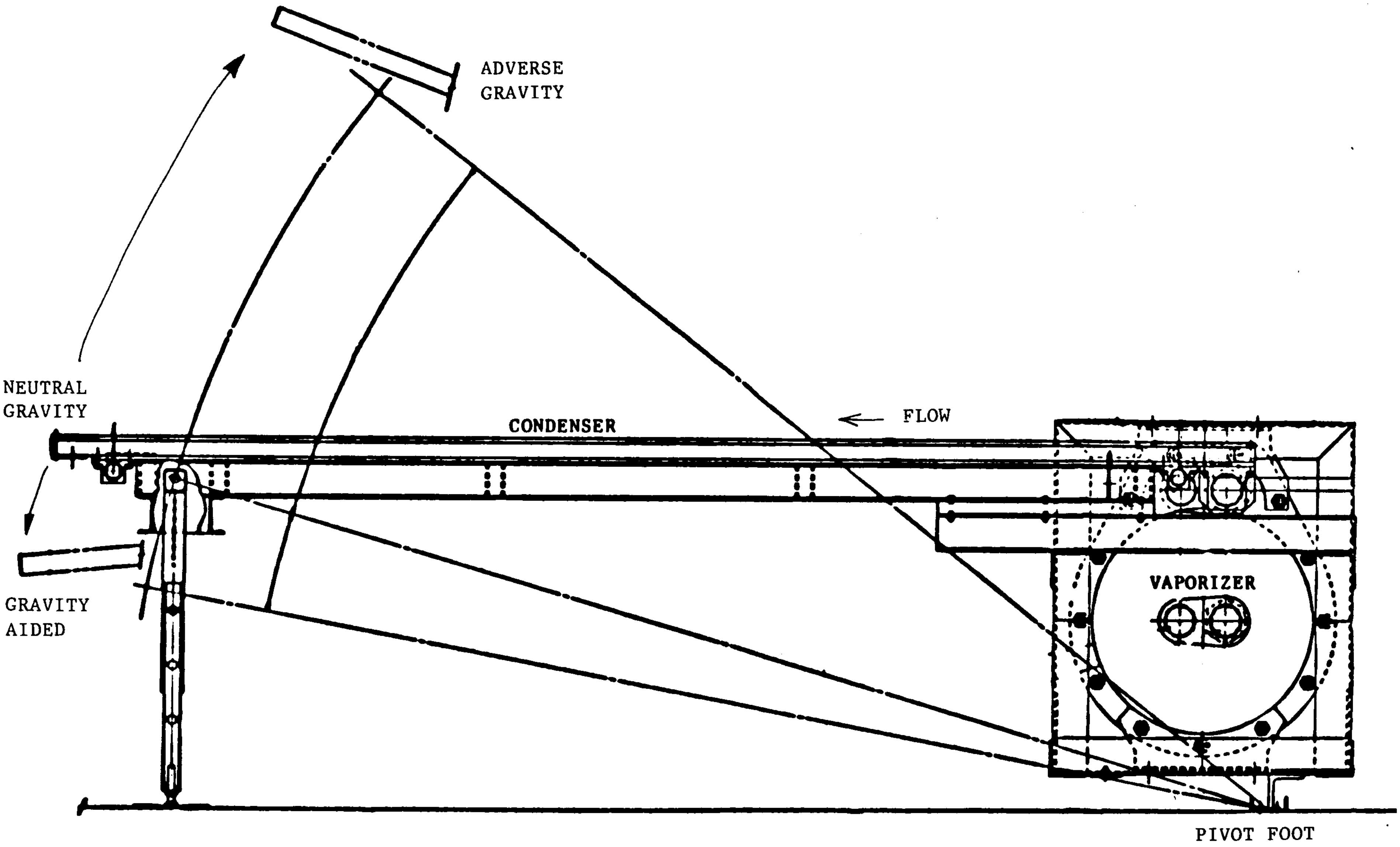

FIGURE 14 - CONDENSER/VAPORIZER STRUCTURE-SIDE VIEW

4.2 RFMD Description

In discussing the basic technology challenges for orbital organic Rankine cycle systems, a number of interrelated functions must be provided; regarding inventory control, net positive suction head for the pump, and heat rejection control. In addition, organic working fluids undergo pyrolytic degradation when cycled to high temperatures. Fluid degradation produces small quantities of noncondensible gases that can accumulate and ultimately affect system performance. Noncondensibles must be collected and periodically removed from the system. Sundstrand has developed a pitot pump based two-phase fluid management system that provides all of the required functions for the orbital application. The pitot pump based device is called the Rotary Fluid Management Device (RFMD). The RFMD utilizes the centrifugal forces of a rotating drum to maintain separation of liquid and vapor phases. Stationary pitot probes positioned in the liquid annulus of the rotating drum fulfill the pumping requirements.

Incorporation of the RFMD into the SD-ORC heat rejection system is shown in Figure 15. The RFMD performs the functions of: controlling system flow by varying the speed of the rotating drum, controlling system inventory when coupled with an accumulator, assuring positive two-phase fluid interface control in micro-gravity by use of centrifugal forces, raising condenser subcooled liquid to saturation conditions, providing saturated liquid to the high pressure feed pump, and providing for noncondensible gas segregation and removal. The RFMD design consists of a motor-driven rotating drum supported by working fluid lubricated

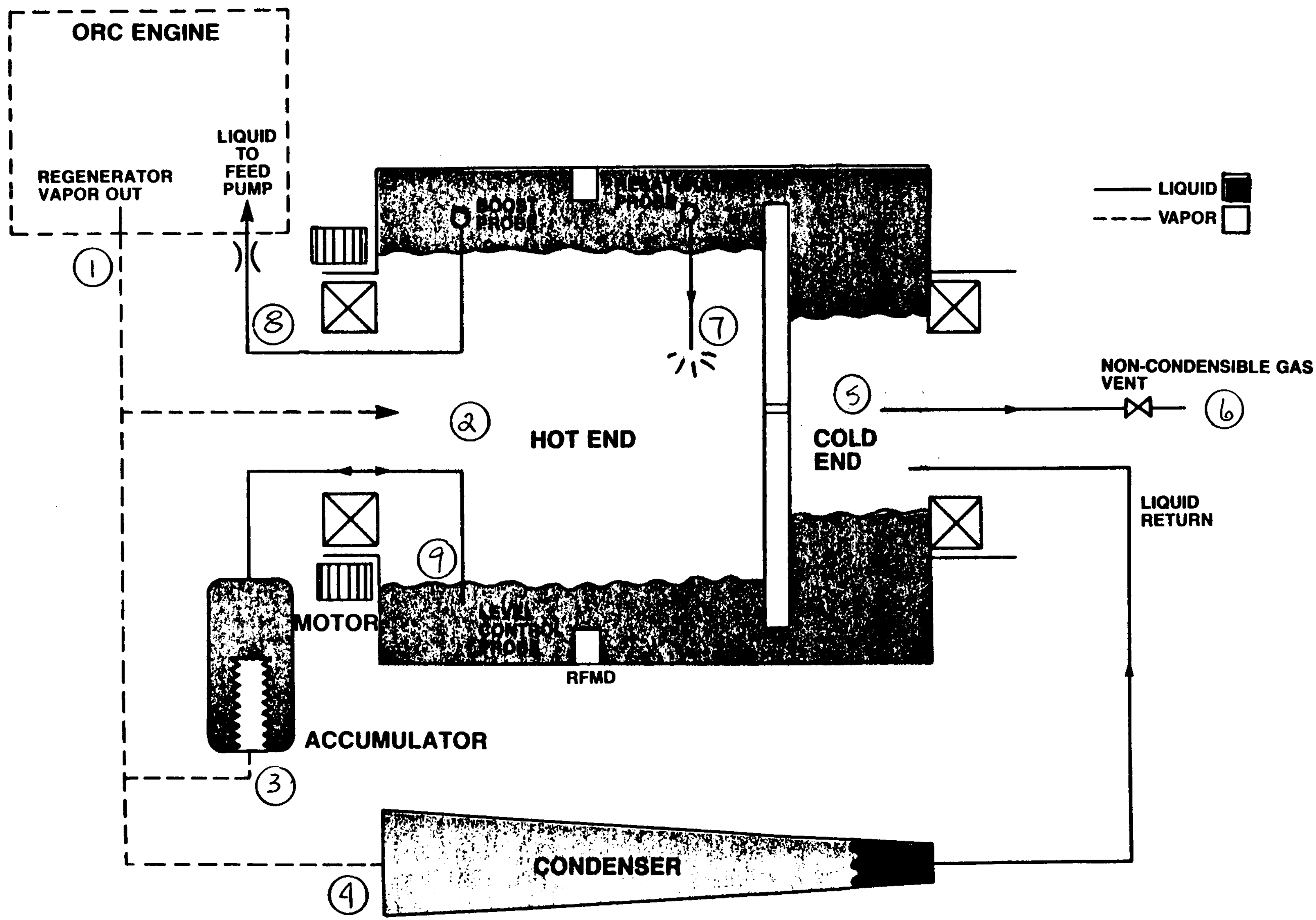

FIGURE 15 - TWO PHASE FLOW MANAGEMENT CONCEPT

journal bearings. Stationary shafts enter the rotating drum at both ends along the rotational axis. The shafts contain concentric fluid passages to bring fluid into and out of the rotating drum. A stationary outer housing surrounds the entire assembly. All of these features, less the stationary outer housing, are shown schematically in Figure 15.

The number of state points indicated in the following description of the operation of the RFMD in an ORC application are shown in Figure 15. In the SD-ORC application, the regenerator delivers dry toluene vapor to the heat rejection system(1). The vapor is referenced to the RFMD hot end(2) and also to the inventory control accumulator bellows(3). The majority of the superheated toluene vapor is delivered to the inlet of the shear flow condenser(4) where the superheat is removed, condensation is completed, and sufficient subcooling is performed to assure the liquid is returned in that state to the RFMD cold end(5). Due to the high vapor velocity in the condenser, any noncondensible gases entrained in the toluene vapor are carried through the condenser and back to the RFMD. In the RFMD, the high density liquid is separated by the centrifugal action of the rotating drum, leaving the low density, noncondensible gases to accumulate in the core. An accumulation of these gases can be externally detected by monitoring the core pressure. When this pressure increases above the vapor pressure associated with the liquid temperature, non-condensible gases are present. The partial pressure of the non-condensible gas causes the increase in pressure. The gas can be periodically vented from the system by evacuating the core of the rotating annulus. The subcooled liquid, which has been centrifuged to the wall, is ported at the perimeter through the central barrier in the

RFMD. This fluid is then collected by the resaturation pitot probe and sprayed into the hot end chamber as a mist[7]. The reference vapor entering the warm end chamber from the regenerator condenses directly on the subcooled mist, raising the bulk fluid to saturation conditions. Subcooled liquid (the surface is at saturation equilibrium, the liquid at the probe is subcooled due to the centrifugal pressurization) is collected by the boost pump pitot probe, and delivered to the system feed pump through a flow control orifice[8]. System inventory control is performed by a static pitot probe in the warm end chamber[9]. This probe is directly connected to the inventory control accumulator. The pressure developed by this static probe is a function of the depth of submergence of the probe in the liquid annulus. The pressure developed by the static probe is balanced against the accumulator spring load to maintain a constant annulus depth. As the system demands more inventory, the annulus depth in the RFMD decreases and the static head on the level control probe also decreases. The accumulator bellows then forces liquid out of the accumulator and into the RFMD to restore balance. Conversely, if inventory is being displaced from the system, the excess liquid accumulates in an increased annulus depth in the RFMD. A deep annulus increases the static probe pressure, and forces liquid back into the accumulator until the equilibrium condition is reached.

The RFMD designed, fabricated, and tested for this program is shown in Figure 16. The rotating drum is driven by an integral motor assembly (Figure 17). The rotating drum is supported by hydrodynamic radial and thrust bearings (Figure 18). The radial journal bearings are mounted in spherical balls to provide for self-alignment of the two shaft design.

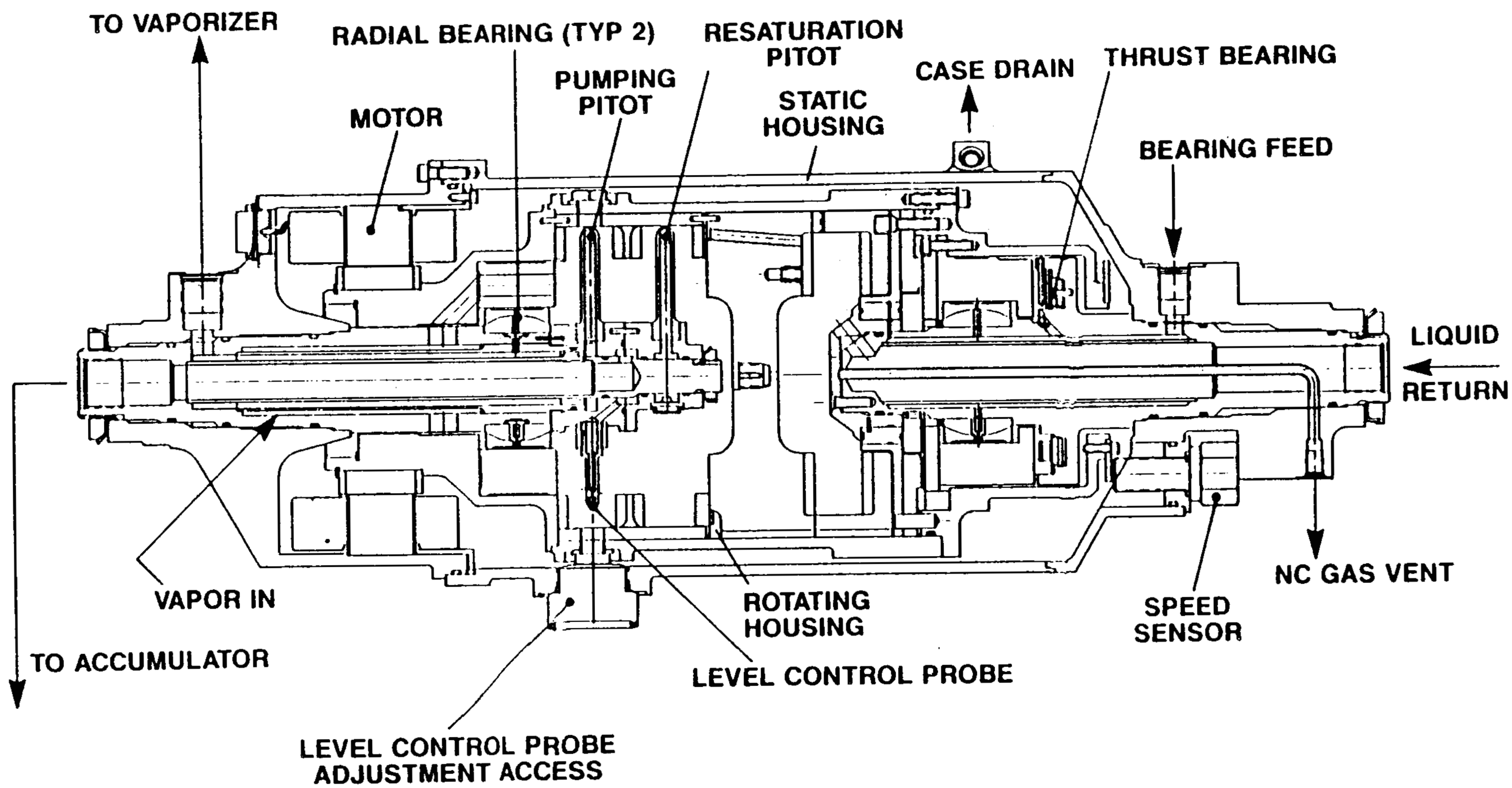

FIGURE 16 - TOLUENE RFMD FEATURES

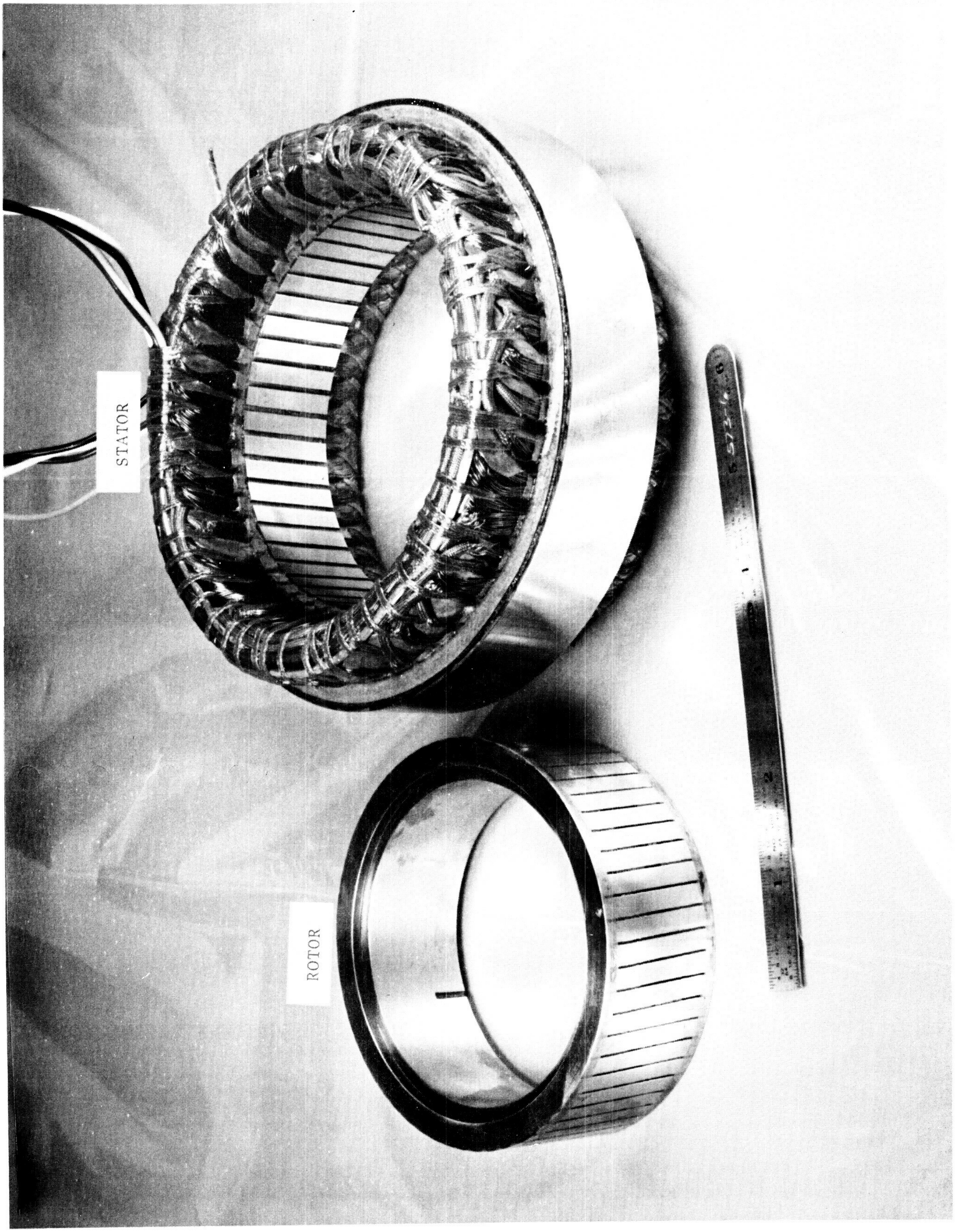

FIGURE 17 - RFMD MOTOR PARTS

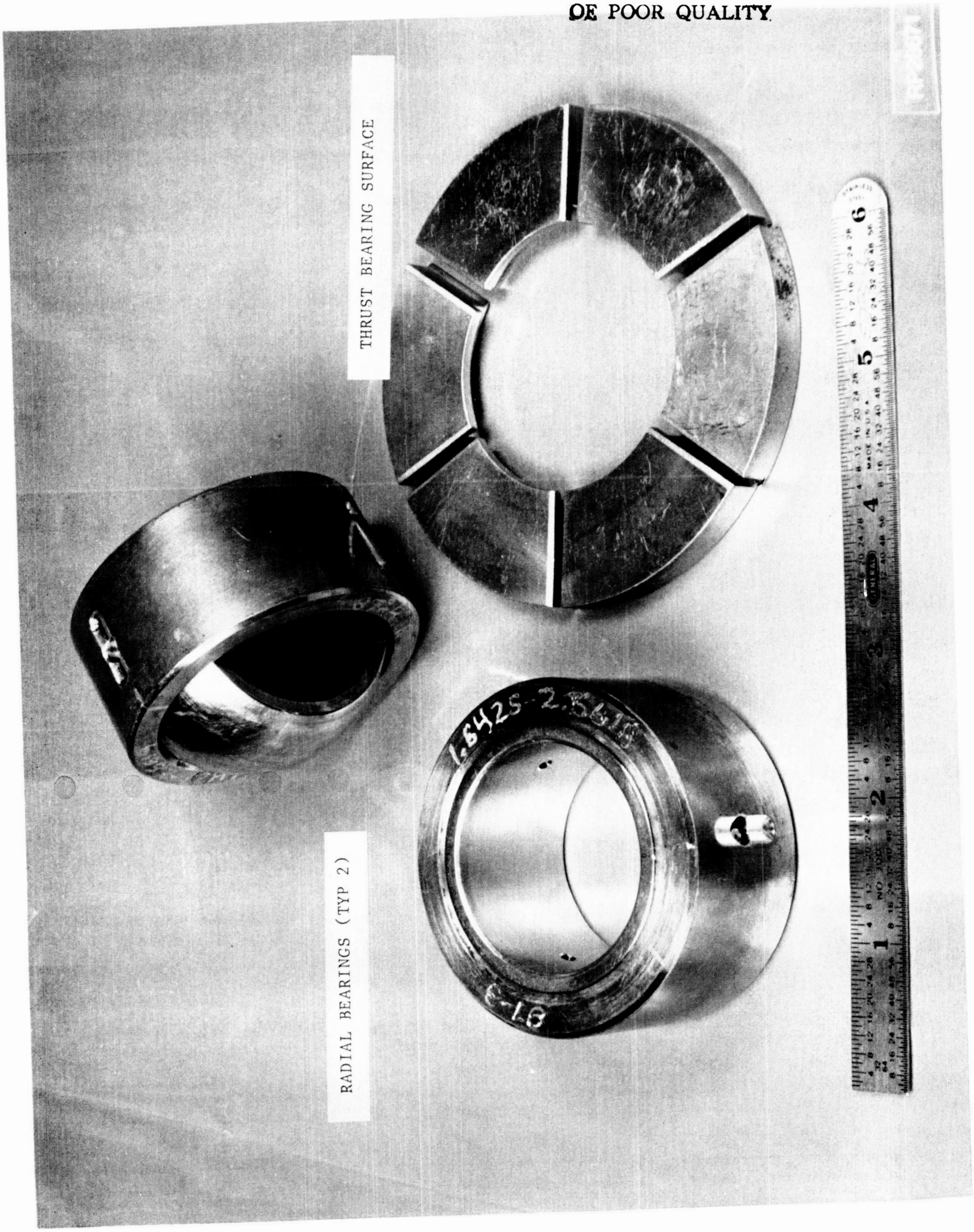

FIGURE 18 - RFMD BEARINGS

The bearings are mounted on two stationary shafts, one at each end of the assembly. Relative motion between the stationary radial bearings and the rotating housing is at the OD of the bearing. The rotating housing rides on a film of toluene lubrication. There is no metal-to-metal contact to cause wear. The pitot probes are mounted on the motor end (or hot end) shaft (Figure 19). The inboard probe is the resaturation probe, with the spray orifices mounted to the hub. This probe sprays a mist of subcooled liquid into the hot end vapor core. The outboard probes are the boost probe (pumping pitot) and the level control probe oriented 180° apart (Figure 20). The boost probe and resaturation probe are total pressure probes with the throat oriented tangentially. The level control probe is a static pressure probe with the throat oriented radially. Concentric fluid passages within the hot end shaft provide for the vapor reference, the level control, and the boost probe flows. The hot end radial bearing is lubed with an internal feed from the boost probe. The second radial bearing and the thrust bearings are located on the opposite shaft, or cold end shaft (Figure 21). The shaft has concentric passages for the bearing lubrication flow (radial, thrust and wind back seal), condensate return, and noncondensible gas vent. The two shaft assemblies and the rotating housing are assembled as a unit designated the Balance Assembly (Figure 22). The rotating housing is surrounded by a static outer housing (Figure 23). The outer housing provides mounting locations for the two stationary shaft assemblies, the motor stator, and the magnetic speed pickup. All of the dynamic seals are contained within the stationary outer housing, which provides a configuration in which all overboard seals are static. Due to the low temperatures and low rotating speeds of this application, conventional materials and standard

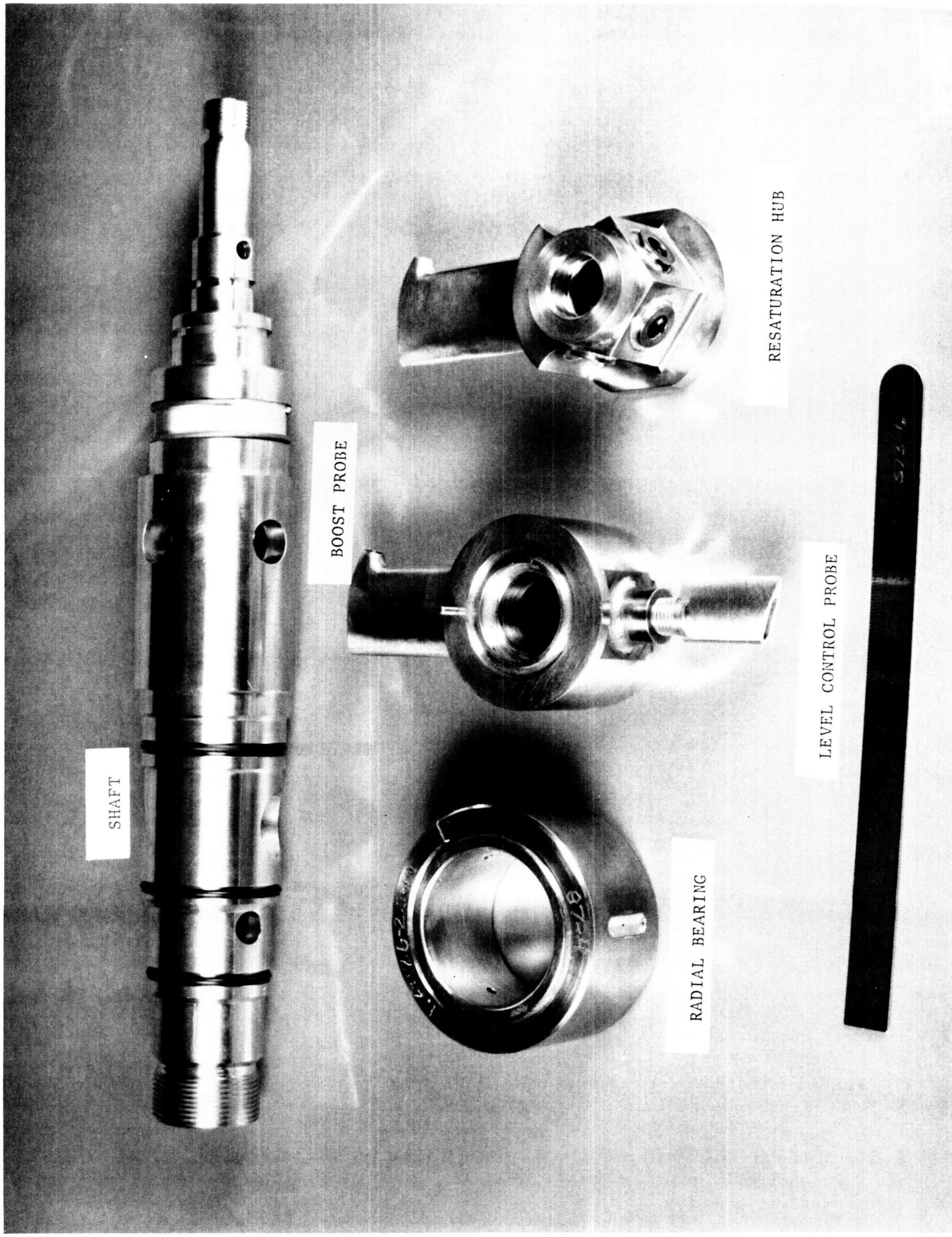

FIGURE 19 - RFMD HOT END SHAFT ASSEMBLY PARTS

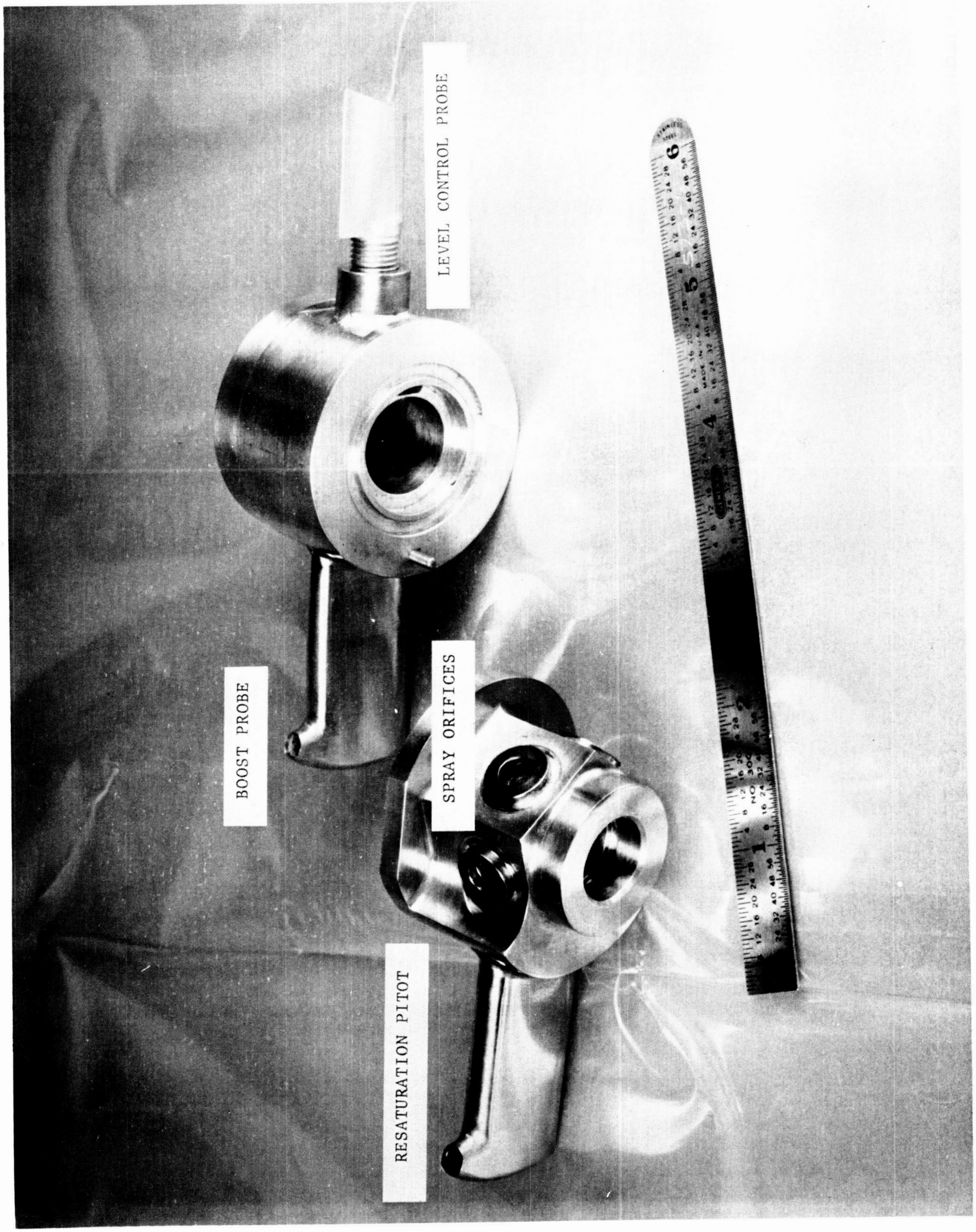

FIGURE 20 - RFMD PITOT HUBS

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FIGURE 21 - RFMD COLD END SHAFT ASSEMBLY PARTS

FIGURE 22 - RFMD BALANCE ASSEMBLY PARTS

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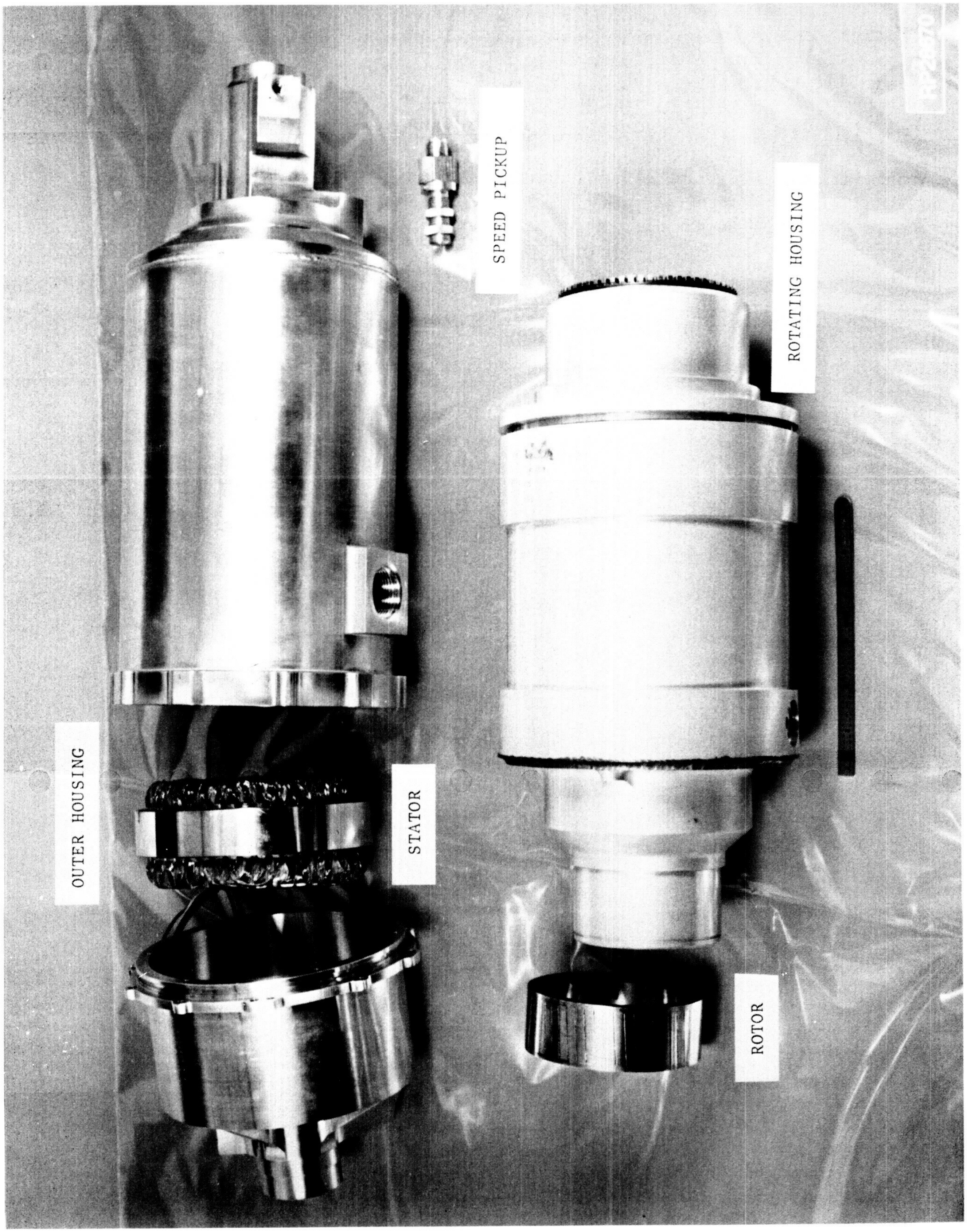

FIGURE 23 - RFMD ASSEMBLY PARTS

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fabrication techniques were used throughout the design. Ryton RF-41 was used as a liner for the rotating drum and for the isolation wall between the two chambers of the RFMD (Figure 24). Ryton was selected due to its toluene compatibility, excellent thermal insulation properties, high strength, and ability to cast to final size.

The design approach chosen for this project was to modify an existing Rotary Fluid Management Device (RFMD) design for the Space Station ORC application. This approach was chosen because it reduced the development risk involved with a new design, which resulted in a reduced schedule and cost risk to the overall program. The existing RFMD design was from the Two-Phase Thermal Management System (TPTMS) program that was developed for the Space Station thermal bus application (WP02)which uses freon and ammonia working fluids at approximately 1/10 the flow rate of the ORC application. The specific modifications to the TPTMS design are summarized below and shown in Figure 25.

1. Toluene working fluid compatible materials

2. A more powerful motor operating at higher speeds and larger pitot probes were needed to deliver 10 times the TPTMS flow rates

3. In the TPTMS application, mixed two-phase fluid is brought into the warm chamber of the RFMD and separated. Dry vapor is returned to the condenser. The ORC application simply references the dry vapor line to the condenser. This

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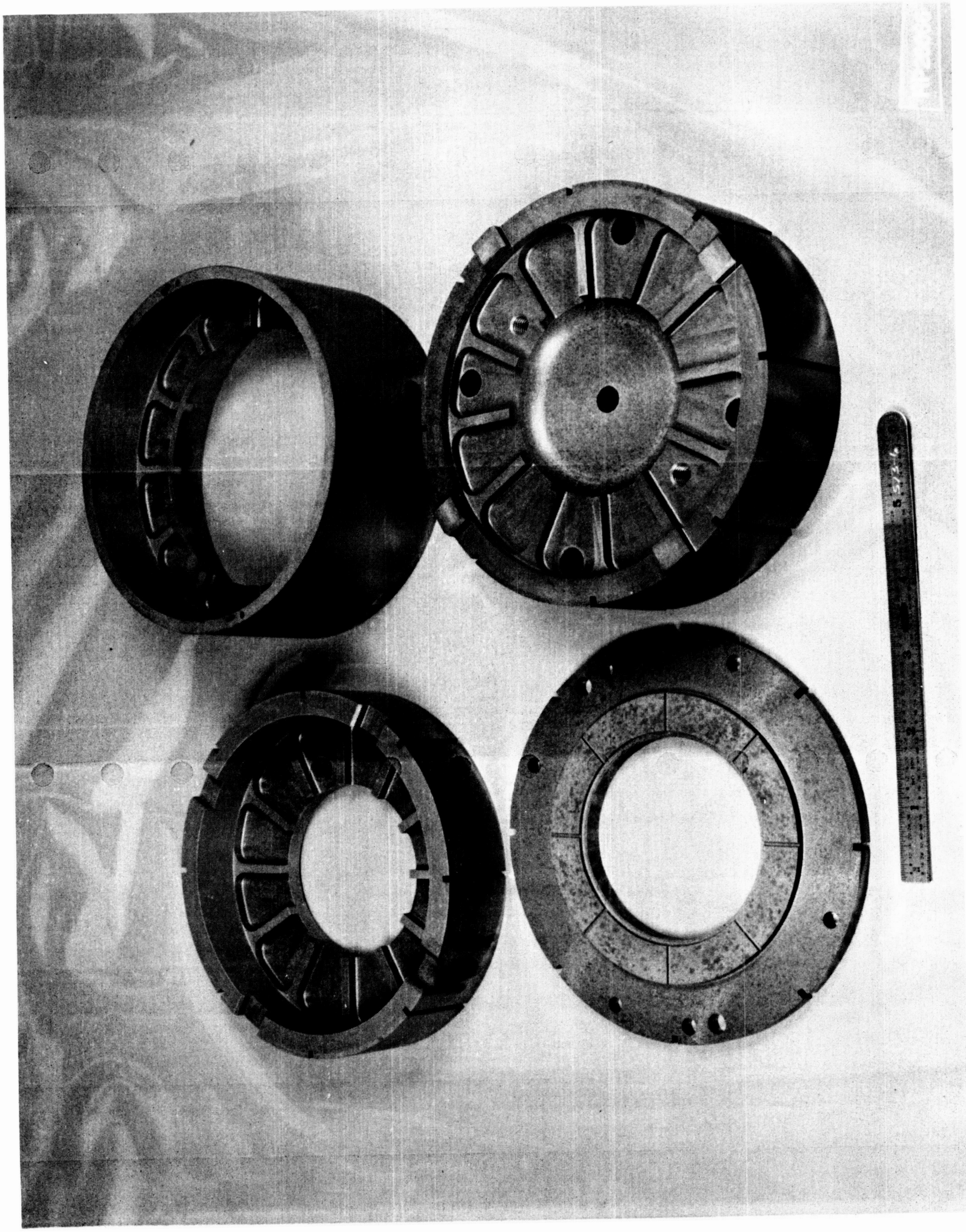


FIGURE 24 - RFMD RYTON RF-41 INSULATION DISCS

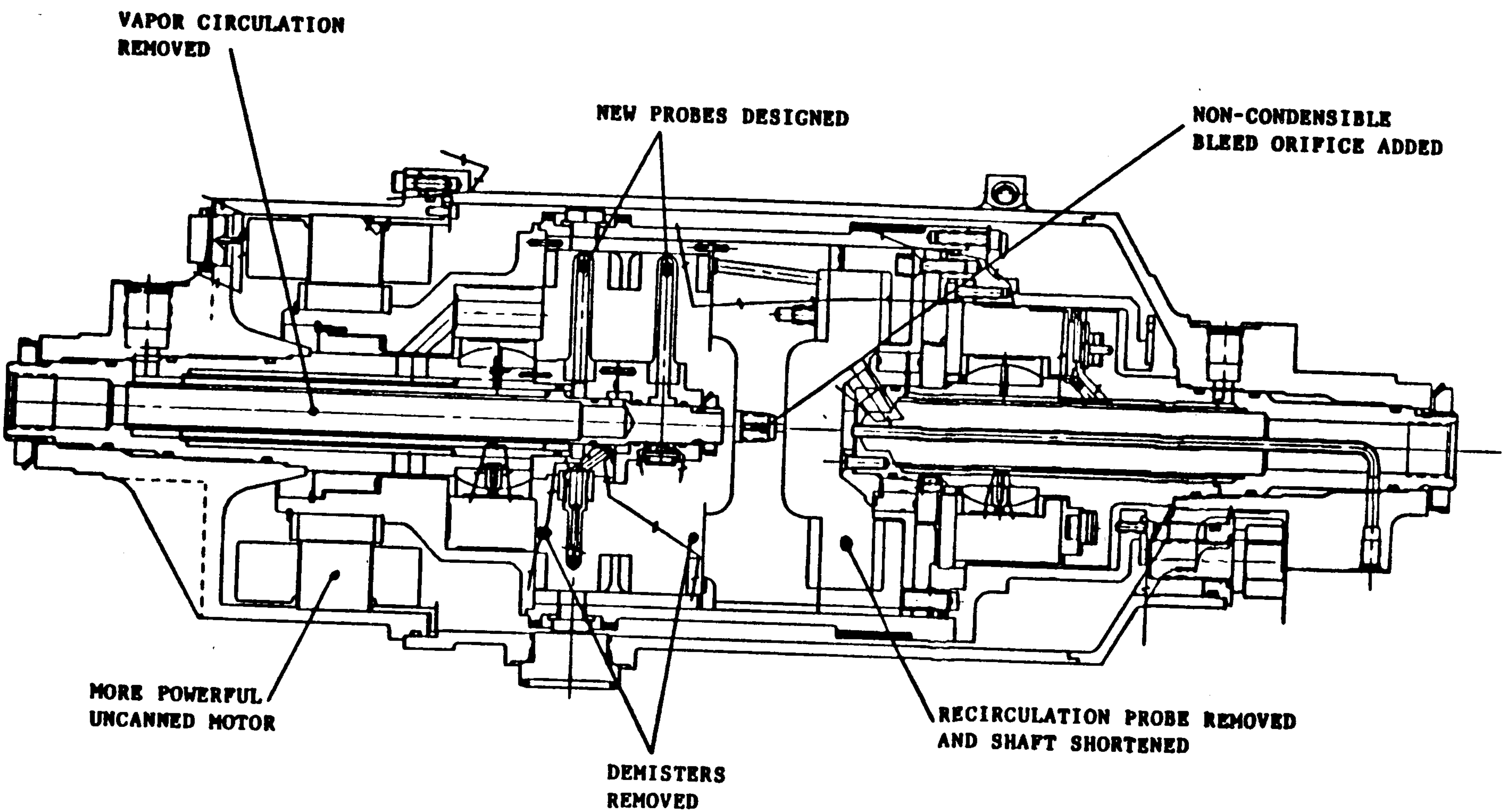


FIGURE 25 - MODIFICATIONS TO TPTMS RFMD

difference in system design allowed for removal of the demisters in the hot RFMD chamber and the removal of the dry vapor outlet line. The final hot end change driven by the difference in system design is that an orifice was added between the RFMD hot and cold chambers. A continuous bleed was needed from the hot to the cold chambers so that noncondensible gases (swept into the chamber by the dry reference vapor) would not accumulate in the hot end.

4. The ORC application does not require the condenser liquid recirculation feature employed by the TPTMS application. The recirculation pitot probe was removed from the cold end chamber.

5. Based on the previous TPTMS fabrication experience, several modifications were incorporated that improved the manufacturability and simplified the assembly of the RFMD.

Key features of the RFMD fabricated for this test program are summarized in Table 1.

Prior to installing the RFMD in the test stand, the dynamic balance of the rotating drum was measured at 2500 rpm. The measured imbalance was 0.33 in-oz. in the left plane and 0.47 in-oz. in the right plane. These values were for the "as-built" configurations, i.e. no material was added or removed to change the amount of imbalance. At this measured level of imbalance, there was no detectable vibration or movement.

TABLE 1
ROTARY FLUID MANAGEMENT DEVICE (RFMD)
KEY FEATURES

Functional Description	o Stationary outer housing o Stationary pitot probes o Motor-driven rotating inner housing o Centrifugal forces within the rotating drum separate liquid and vapor o Pitot probes utilize the dynamic head of the rotating liquid annulus to pump liquid to feed pump inlet o Subcooled liquid is returned to the RFMD from the condenser o Liquid level control in the rotating drum is maintained by a static pressure probe that balances a spring force in an accumulator
Working Fluid	Toluene
Layout drawing	EP2809-3101
RFMD assembly drawing	EP2809-3110
Performance (max. insolation, 115% power level)	
Flow rate	2200 lbm/hour
Motor shaft power	750 watts
Speed	3300 rpm
Dimensions	
Overall length	23.5 inch
Overall OD	7.3 inch
Rotating drum ID	5.934 inch
Rotating drum OD	6.315 inch
Weight	80 lbs
Pitots	
Number	2
Pitch diameter	5.0 inch
Throat diameter	0.200 inch
Bearings	Working fluid lubricated
Radial	Monoball journal
Thrust	Tilting pad (one direction only - motor end up)
NC gas orifice size	0.005 inch equivalent diameter
Materials of construction	Stainless steel Ryton Aluminum Viton

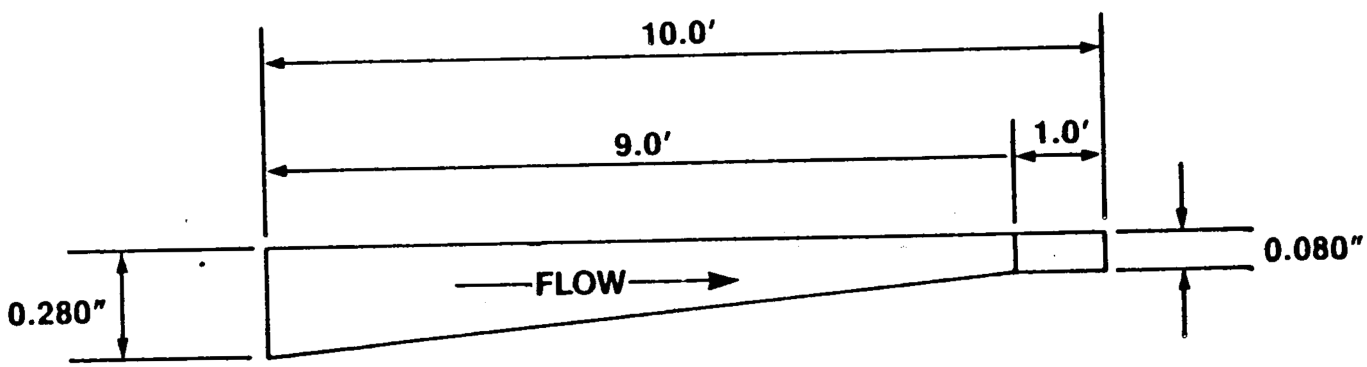
4.3 Condenser

The condenser of an organic Rankine cycle power generation system interfaces with the space radiator to provide the waste heat rejection function for the thermal cycle. This heat rejection results in a change in working fluid phase from vapor to liquid, with the latent heat of vaporization being rejected in the process. Sufficient subcooling at the condenser outlet is necessary to provide adequate net positive suction head to the pumping system. While numerous modes of condensation and condenser types exist, only three different types of condensers have been seriously considered for use in space: capillary control (as in heat pipes), shear flow-controlled, and direct contact condensation (jet condenser). The combination of high power which implies high mass flows, the selection of toluene working fluid with a relatively high vapor pressure, and the need for on-orbit start and restart indicate that a shear flow-controlled condenser would be the best approach for this application.

Shear-flow surface condensers consist of a number of parallel tubes or channels tapered in the direction of flow to maintain high vapor velocities. High velocities sweep the condensate and noncondensibles in the flow direction, even when subjected to adverse gravity forces. Vapor enters the large cross-sectional area end of the channel, and subcooled condensate exits the small cross-sectional area end of the channel. The channels are cooled to a temperature below the saturation temperature of the vapor by some means, and the vapor condenses by interaction with this subcooled surface. By using shear-controlled flow, noncondensible gases can be swept through the condenser along with the condensate. As with any condensing process, an accumulation of noncondensibles will tend to block the condensing surface by adding additional thermal resistance, resulting in higher condensation temperatures and effectively lower heat transfer coefficients.

The condenser for the Task 2 test loop was designed to be functionally prototypic of a flight Space Station condenser operating at the proper state points, condensing up to 2100 lb/hr of toluene, and rejecting up to 113 Kw_{th}. The condenser comprises a 1-inch thick aluminum plate approximately 4 feet wide and 12 feet long. The current flight design condenser is approximately 45 feet long. The 12 foot length was arbitrarily chosen as a compromise between a reasonable size to handle in the existing test facility and the desire to be as prototypic as possible. Cost and manufacturability were also major considerations. The toluene channels were designed to have the same pressure drop and mass flow as the Space Station application. The top face of the plate contains 55 tapered channels used for toluene condensation. The width of the channels was constant. The toluene channels decreased in depth in the flow direction. This resulted in a decrease in cross-sectional area which maintained adequate vapor velocities for shear-controlled condensation, see Figure 26. Heat rejection necessary for toluene condensation was accomplished by passing cooling water through 70 milled channels on the bottom face of the 1-inch thick aluminum plate. The coolant flow rate could be varied to achieve the desired toluene condenser exit conditions. The toluene channels were covered with tempered glass (note subatmospheric operation) to provide full

- TYPICAL OF 55 CHANNELS
- CONSTANT 0.5 INCH WIDTH



Drawing Is Not to Scale

FIGURE 26 - TOLUENE CHANNEL DIMENSIONS

visualization of the condensation process. The water channels were covered with an aluminum plate. Both the glass and aluminum plates were sealed with perimeter o-ring seals, see Figure 27.

Key features of the condenser constructed for this test program are listed in Table 2.

4.4 Accumulator Description

The working fluid inventory control accumulator provides several important functions in an orbital Organic Rankine Cycle power conversion system. When the system is not operating, the working fluid inventory is stored in the accumulator. The accumulator accommodates the changes in the amount of inventory in the system and the distribution of inventory during startup and shutdown transients. When the system is operational, the accumulator is used as a makeup device to compensate for system component inventory changes due to orbital and seasonal variations. The accumulator in the test loop served these same functions: storage during periods of nonoperation, provide accumulation of fluid for loop startup/shutdown, and accommodate inventory swings due to changes in operational conditions.

A flight design accumulator would consist of a spring loaded bellows that is referenced to the condenser inlet pressure. Since the condenser inlet pressure is common with the core of the RFMD hot end chamber, the accumulator spring load is balanced against the static pressure developed by the RFMD level control probe. A flight design accumulator was not required for a ground demonstration and a commercially available, welded

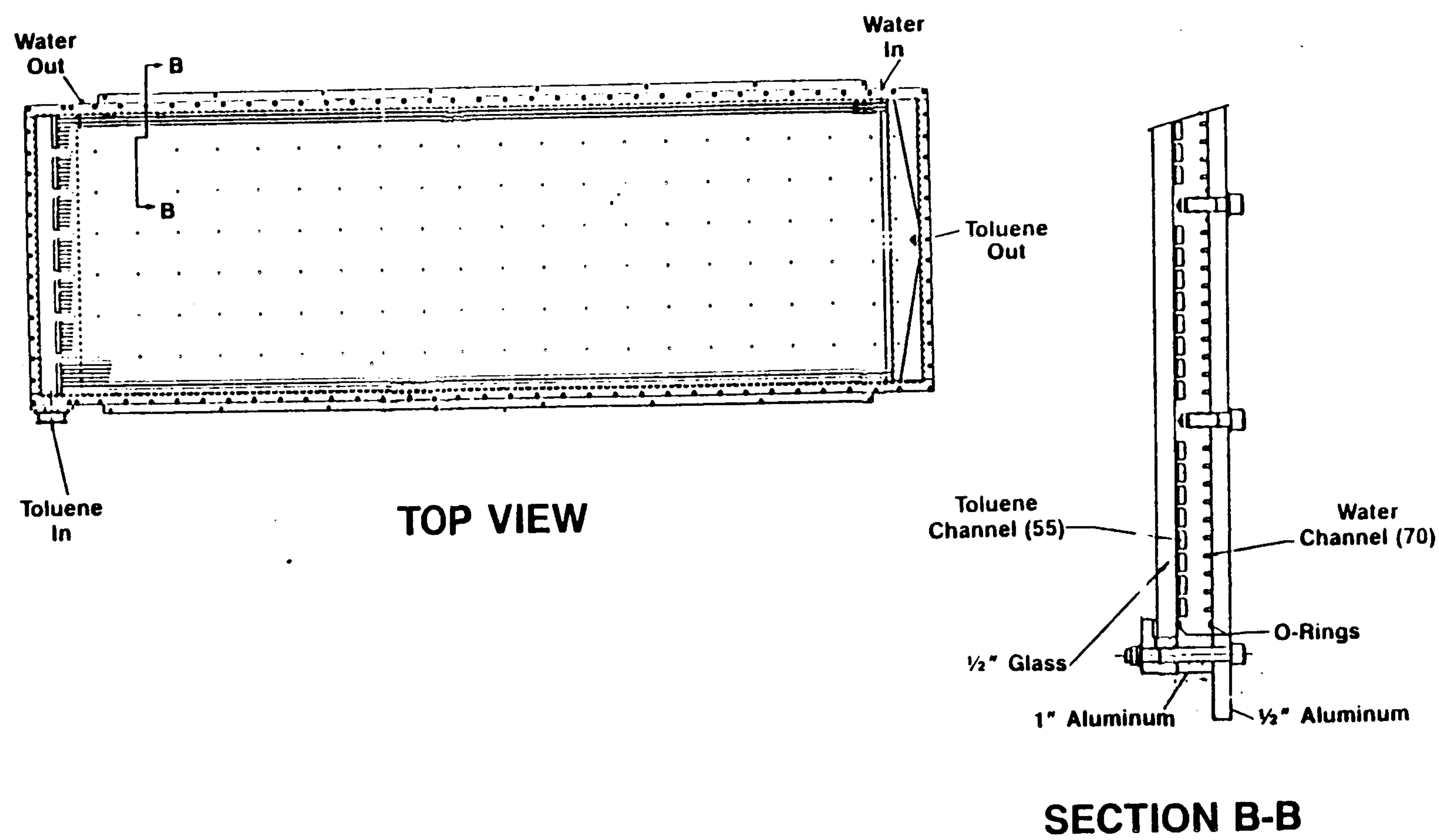


FIGURE 27 - CONDENSER FEATURES

TABLE 2
CONDENSER KEY FEATURES

Functional description	o Single sided heat exchanger o 1-inch aluminum plate core o Toluene channels cut in top face - decreasing channel cross-sectional area keeps vapor velocity high (shear control) o Water channels cut in bottom face. Constant cross-sectional area o Glass cover over toluene channels for full visualization of the condensation process
Layout drawing	EP2809-3901
Overall dimensions	
Length	o 134 inches
Width	o 44 inches
Thickness	o 2 inches
Materials of construction	o Aluminum o Tempered glass o Viton o-rings (for toluene) o EPR o-rings (for water)
Design Performance (maximum insolation)	
Toluene flow	o 2090 lbm/hour
Water flow	o 30 gpm
Toluene ΔP	o 1.6 psid (at max flow)
Toluene nominal inlet pres	o 3 psia-always subatmospheric
Water ΔP	o 6.4 psid
Water maximum operating pres	o 60 psig
Q	o 409,896 BTU/hour (120 Kw_{th})
Toluene Channels	
Number	o 55
Width	o 0.500 inch
Length	o 10.0 feet
o Condensing section	o 9.0 ft
o Subcooler	o 1.0 ft
Depth	
o Condensing section	o Straight taper over 9 ft
Inlet	o 0.280 inch
Outlet	o 0.080 inch
o Subcooler	o 0.080 inch depth constant
Channel-to-channel pitch	o 0.625 inch (in 7 groups; 6 of 8 each and 1 of 7)
Water Channels	
Number	o 70 (7 groups of 10)
Width	o 0.100 inch
Depth	o 0.220 inch
Channel-to-channel pitch	o 0.487 inch

metal bellows accumulator was purchased (Flexicraft P/N 84121-F2) that met all the requirements of the application. The accumulator can be seen in the right foreground of Figure 11. The inherent spring rate alone, associated with the metal bellows of the purchased accumulator, was not sufficient for this application. A gas regulator was chosen that was capable of sensing a reference pressure (the condenser inlet pressure) and adding an adjustable offset to the referenced pressure. The gas regulator performed the function of the spring load in a flight design accumulator. The ability to change the amount of offset effectively created an adjustable and variable spring load for the accumulator. This feature allowed for varying the depth of the liquid annulus in the RFMD rotating drum (the higher the gas offset pressure, the deeper the fluid annulus). The gas pressurized accumulator configuration also eliminated startup and shutdown transient concerns associated with condensation on the vapor side of the accumulator. The accumulator was sized to provide a working inventory capacity of 15 gallons.

4.5 Facility Support Equipment

The RFMD, condenser and accumulator were the key components in the test stand. The balance of the facility support equipment is summarized in Table 3.

5.0 System Operation

5.1 Procedures

Developing procedures for startup, steady-state operation and shutdown were an important aspect of this task. Previously developed procedures

TABLE 3
FACILITY SUPPORT EQUIPMENT

Test Loop Layout Drawing	EP2809-4501
Condenser Support Structure	EP2809-4511
RFMD Support Structure	EP5531-24 EP5531-27
Oil Heater	
Vendor	Sterling, Inc.
Capacity	150 Kw_{th}
Fluid	Therminol 66
Vaporizer	
Vendor	Graham
Model	Heliflow 20-28L
Data Acquisition System	Analog Devices MACSYM 150 and 250 Digital Printer 8-Pen Analog Brush Recorder
Instrumentation	
Temperature	Calibrated Type K Thermocouples
Pressure	Strain Gauge Transducers
Flow, Toluene	Micromotion Model D100
Flow, Oil and Water	Turbine Flow Meters
Safeties	Oil Over-temperature RFMD Bearing Under Flow RFMD Under Speed Toluene Vapor

from the TPTMS experience were used as the starting point for developing procedures unique for the ORC application. There are several system differences between the two applications that dictated the development of different procedures. Two additional constraints of the test facility exist that are not representative of a flight system.

The first constraint was that the internal pressure on the large glass cover of the condenser must remain at or below atmospheric pressure to prevent rupture of the glass. The glass is capable of containing only 1 psi superatmospheric pressure inside the condenser. Normal operation of the condenser is always subatmospheric (3 psia). TPTMS startup calls for pumping liquid through the condenser. For this loop that held the potential of overpressuring the condenser end thus the procedure was modified to facilitate startup with a partially full condenser. A flight design would not have a glass cover and would be adequately pressure rated to accommodate either start technique.

The second constraint was also a startup constraint. The TPTMS RFMD was originally designed for 1/10 of the flow required for the ORC application. Incorporation of a more powerful motor and larger pitot pumps into the existing design provided the required increase in system flow capacity. The rate the accumulator could supply makeup fluid during startup, however, remained at 1/10 capacity. This constraint dictated a "bootstrap" start procedure in which system flow was initiated at 1/10 design flow until the test loop was operating with normal fill levels. Flow could then be increased to design levels. A flight design would be capable of filling the RFMD from the accumulator as fast as the pitot pump could remove the fluid and thus obviate the low flow bootstrap start requirement.

Operation of the test rig was accomplished with, remote controls and manually positioned valves. A short summary of the procedures developed for start, steady-state operation and shutdown are presented. Refer to the test stand schematic in Figure 4.

Start

1. Status:

   - o Toluene system at room temperatures saturated conditions (if not, open vacuum valve to reduce pressure to ambient saturation)
   - o RFMD axis horizontal
   - o Condenser downhill slightly
   - o Throttle valve closed
   - o Start valve closed
   - o Case drain valve open
   - o Bearing feed valve open
   - o Accumulator pressure offset = 6 psi

2. Set facility water flow at 60 gpm

3. Therminol pump and heater 'on' (temperature controller = 180^{o}F)

4. Verify RFMD start is 'off', RFMD power supply 'on' (400 Hz and 115 Vac)

5. Override "RFMD bearing underflow" and "RFMD underspeed" safety circuits

6. Start brush recorder and printer. Set digital data acquisition to automatic at one minute intervals

7. Open start valve
   - o Verify weight decreasing
   - o Visually verify fluid entering condener at outlet

8. Start RFMD
   - o Verify speed = 3300 rpm
   - o Verify bearing flow > 0.4 gpm
   - o Activate safety circuits

9. Open throttle valve slowly to initiate system flow, filling vaporizer and condenser (Note: maintain bearing flow above 0.4 gpm). After the loop is filled, adjust throttle valve to desired flow (maintain bearing flow above 0.4 gpm).

10. Intermediate water pump 'on'
   - o Adjust flow to 30 gpm

Steady-State:

Steady-state operation of the test rig consisted primarily of fine-tuning the state points of interest by adjusting different parameters. The parameters that can be varied and the approximate range of capability are summarized below.

o	Facility Water Flow	0 to 60 gpm
o	Intermediate Water Flow	0 to 40 gpm
o	Oil Temperature	70 to 300°F
o	RFMD Speed	1000 to 3500 rpm
o	Toluene Mass Flow	0 to 2200 lb/hour
o	RFMD Axis	Horizontal or Vertical
o	Condenser	-6 to +22° from Horizontal

Periodic noncondensible gas venting was performed by verifying vacuum at vent valve, open vent valve until no further decrease in non-condensible gas vent pressure (PNCV) is noted, close vent valve.

While operating steady-state, RFMD and/or condenser could be rotated to adverse orientations without any system adjustments due to flex line interconnects.

Shutdown:

1. Return condenser to slight downhill, return RFMD to horizontal
2. Close throttle valve
   - o Verify no system flow
3. When no further weight increase in accumulator is indicated, RFMD motor power off
4. Close start valve
5. Oil heater and pump off

6. Water pump off

7. Facility water off

8. Turn off brush and digital data acquisition

5.2 Test Results

A test plan was defined and implemented that addressed the technical objectives of this experimental program. Table 4 summarizes the specific test matrix performed. The primary technical objective was to demonstrate that the RFMD and shear flow condenser are a viable approach for controlling two-phase interfaces in micro-gravity ORC applications. This objective was achieved by successfully operating the key components in neutral and adverse gravity orientations.

The secondary technical objective was to compare results from Sundstrand's analytical methods to experimental data. The test rig was operated over the entire range of the S.D. organic Rankine cycle operating conditions. RFMD and condenser performance at key test points were measured and compared to pre-test performance predictions.

5.2.1 Primary Technical Objective

To verify that the RFMD and condenser are a viable approach to two-phase fluid management in a micro-gravity environment, a ground demonstration test program was defined.

A series of tests were performed at a constant thermal load, while changing the RFMD and condenser orientations. The RFMD and condenser

TABLE 4

TEST MATRIX

EQUIVALENT ELECTRICAL POWER RATING	COMPONENT ORIENTATION			
	RFMD → CONDENSER →	RFMD → CONDENSER ↗ 2°	RFMD ↓ CONDENSER ↗ 2°	RFMD ↓ CONDENSER →
80%	X	X	-	X
94% (TES STATUS CHECK)	X (1) (2)	X (2)	X (2)	-
100% (NOMINAL INSOLATION)	X (1) (2)	X	X	-
115% (MAXIMUM INSOLATION)	X (1) (2)	-	-	-

(1) PERFORMANCE PREDICTIONS COMPARED TO MEASURED PERFORMANCE

(2) TEST PLAN POINTS

were mounted on manually positioned structures that allowed the orientation of each component to be changed. The rotational axis of the RFMD was positioned parallel and perpendicular to the floor. The condenser channels were positioned horizontal (neutral gravity) and uphill (adverse gravity) orientations. The adverse gravity tests were performed at the lowest toluene mass flow rate condition and represented a the most severe test for shear flow control. This is because lower flow rates in the channel geometry results in lower velocities.

RFMD performance was compared at horizontal versus vertical orientations. The chart below summarizes key RFMD parameters for the same heat load and different RFMD orientations.

	Horizontal	Vertical
Test Point	#1	#3
Total Flow (lb/hr)	1754	1700
Head Rise (psi)	14.9	15.1
Speed (rpm)	2811	2803

(Refer to Appendix 1 for full data set)

This chart demonstrates that the performance change between horizontal and vertical orientations was negligible. The key point of this procedure was to determine if the centrifugal force keeping the liquid annulus on the walls of the rotating drum was affected by the direction of the gravity force. The vertical position is the most severe test. If the centrifugal forces were insufficient, the annulus would "slump" away from the pitots (the pitots are at the top of the assembly in this orientation) and pumping performance would drastically deteriorate. The data indicates that this did not occur. This demonstration verifies that the centrifugal forces of the rotating drum are dominant in the RFMD and the annulus is insensitive to the external gravity orientations.

Stable condenser operation was obtained at all of the test points. Shear flow controlled condensation was observed at one-half the design flows during the startup procedure. The various flow regimes documented in Figures 28 through 33 were very uniform from individual channel to channel. Stable condenser operation was maintained when the channel flow passage was changed from horizontal to uphill 2^{o}. The visual appearance of the condensation and the axial location of each flow regime did not appear to change. This demonstration verifies that the shear flow forces are capable of moving the fluid in a controlled manner, even against the forces of gravity. In the axial direction (direction of flow), the effective adverse gravity was 0.04g. A second indication that the shear forces in the channel are controlling the flow is shown in Figure 30. In this photo, the vapor is being condensed on the cold side walls and bottom of the channel. The shear forces of the high velocity vapor core are capable of maintaining a liquid layer on the top surface of the channel (the bottom surface of the glass cover). The shear forces maintaining annular flow are greater than the transverse gravity forces on the fluid.

Both of these points verify that for these ground tests, the shear flow forces in the condensing channels dominated the gravitational forces. In a micro-gravity environment, the shear forces would also be dominant.

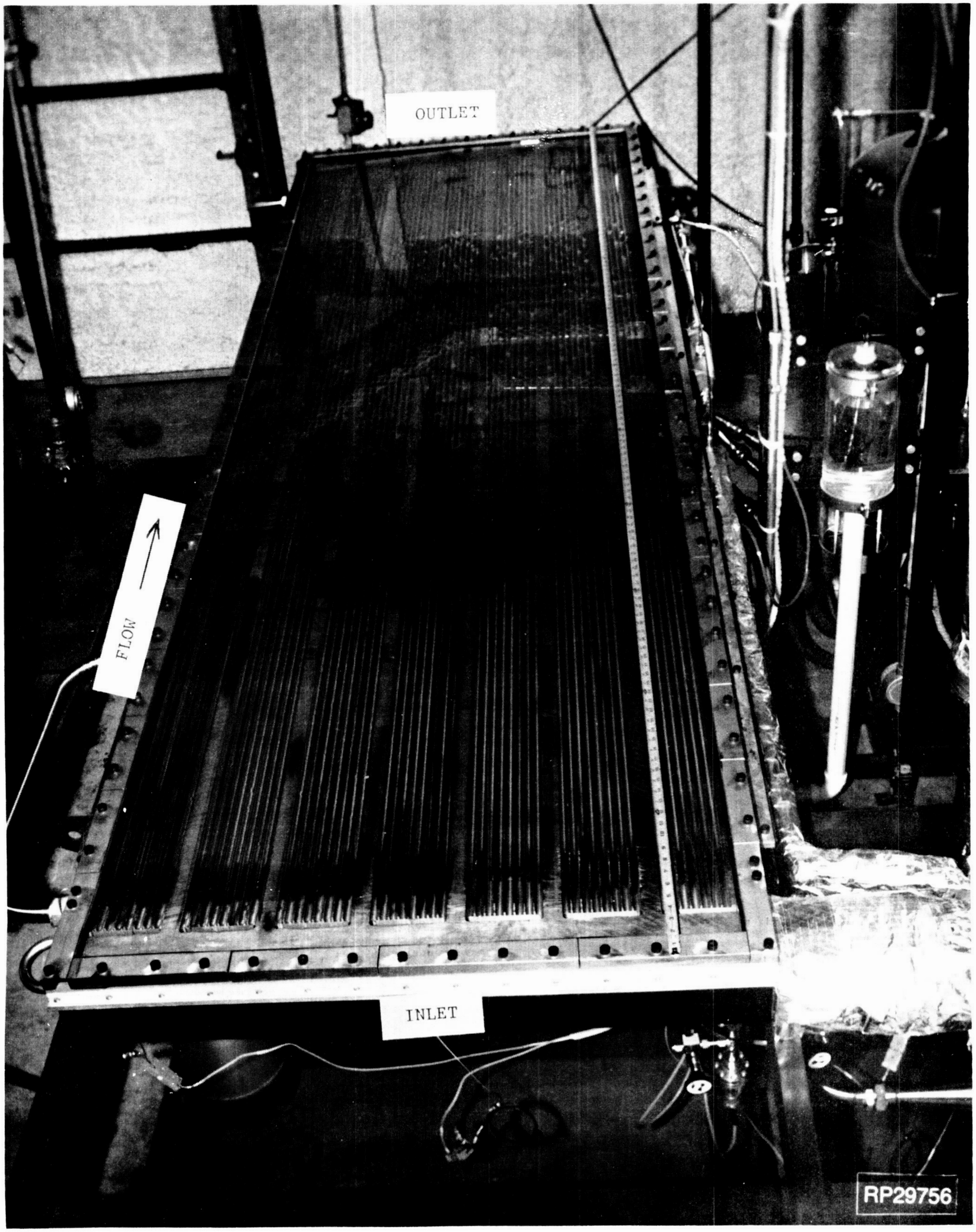


FIGURE 28 - CONDENSER IN OPERATION

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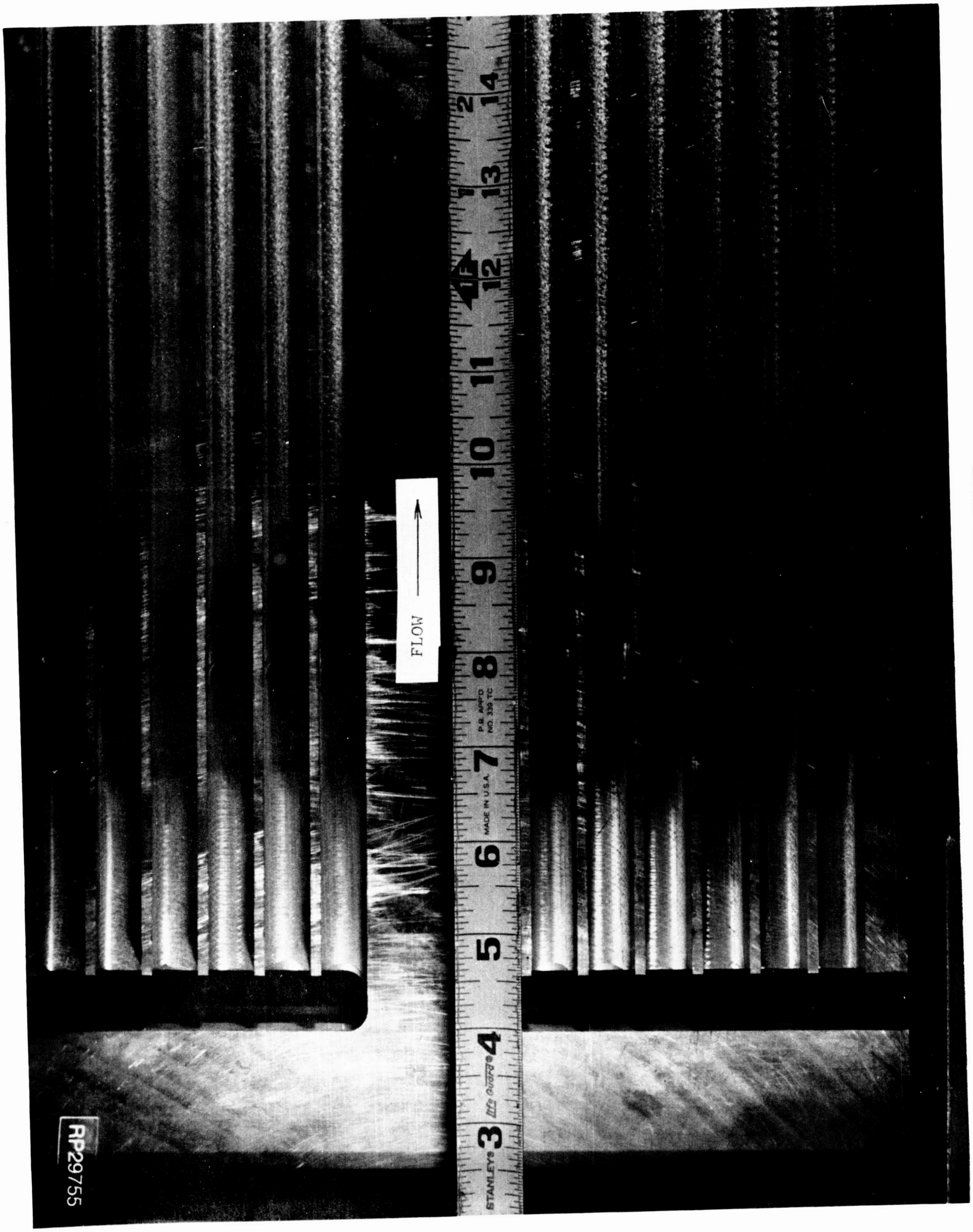


FIGURE 29 - CONDENSER INLET

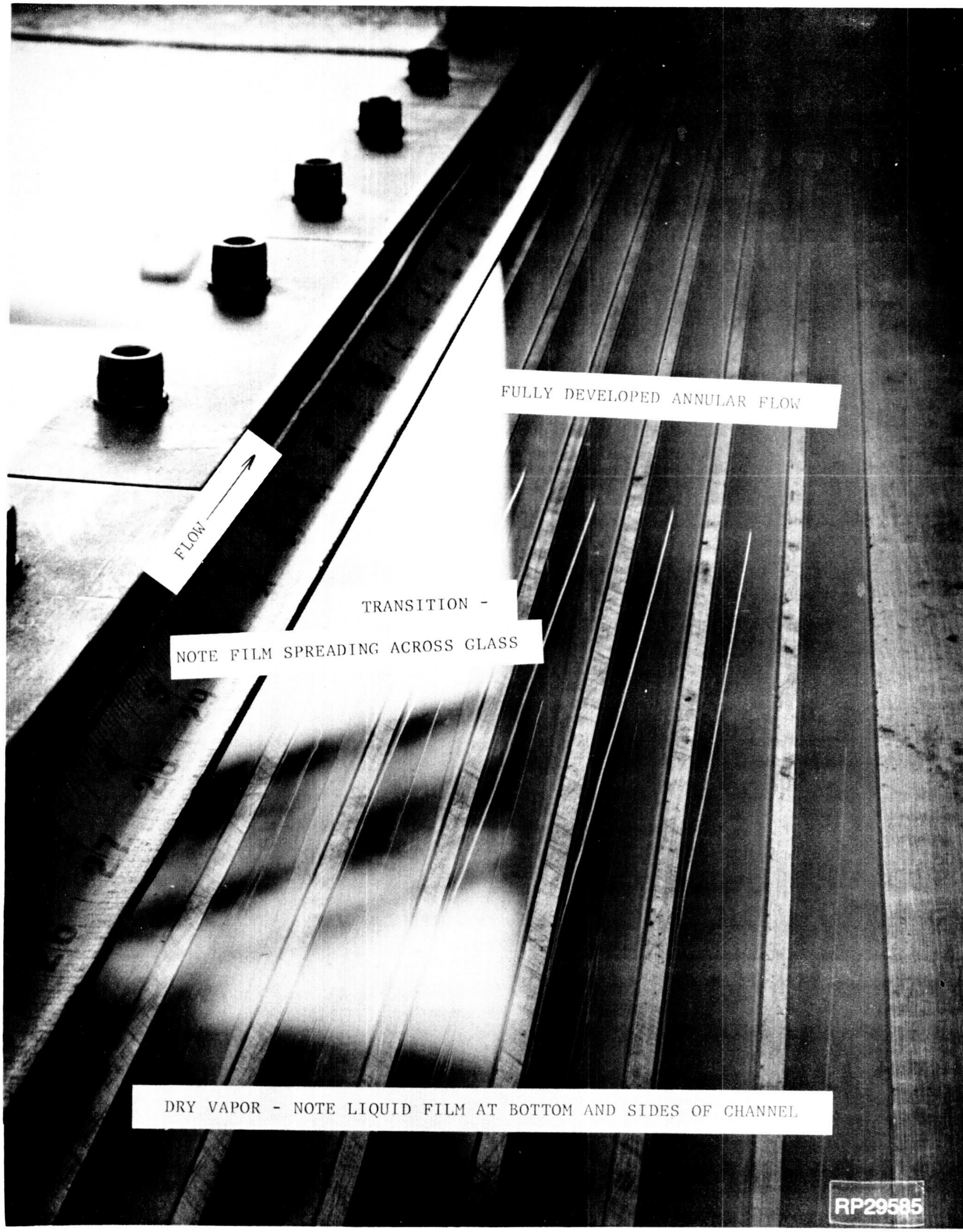


FIGURE 30 - START OF ANNULAR FLOW IN CONDENSER

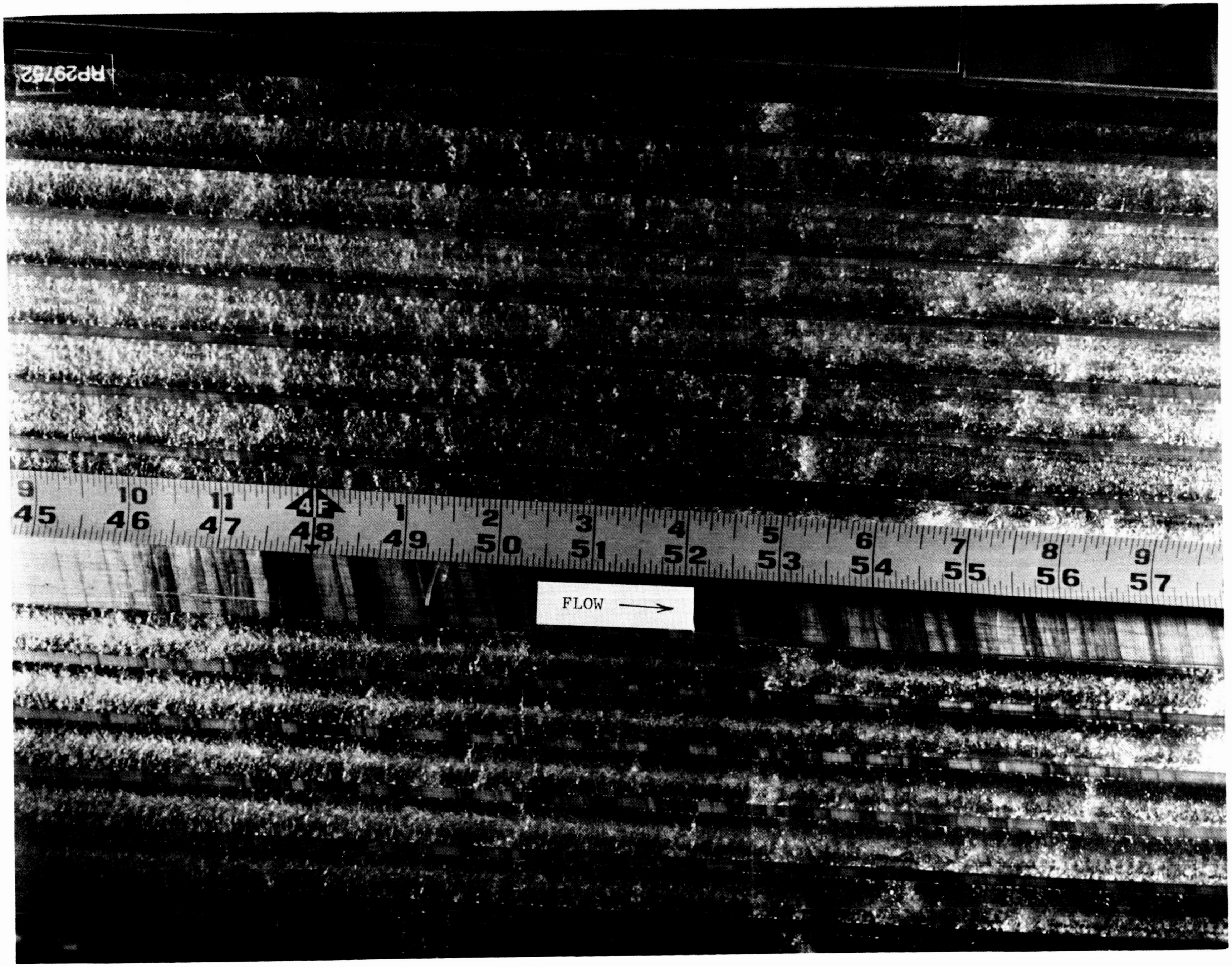


FIGURE 31 - ANNULAR FLOW IN CONDENSER

FIGURE 32 - VAPOR LIQUID INTERFACE IN CONDENSER

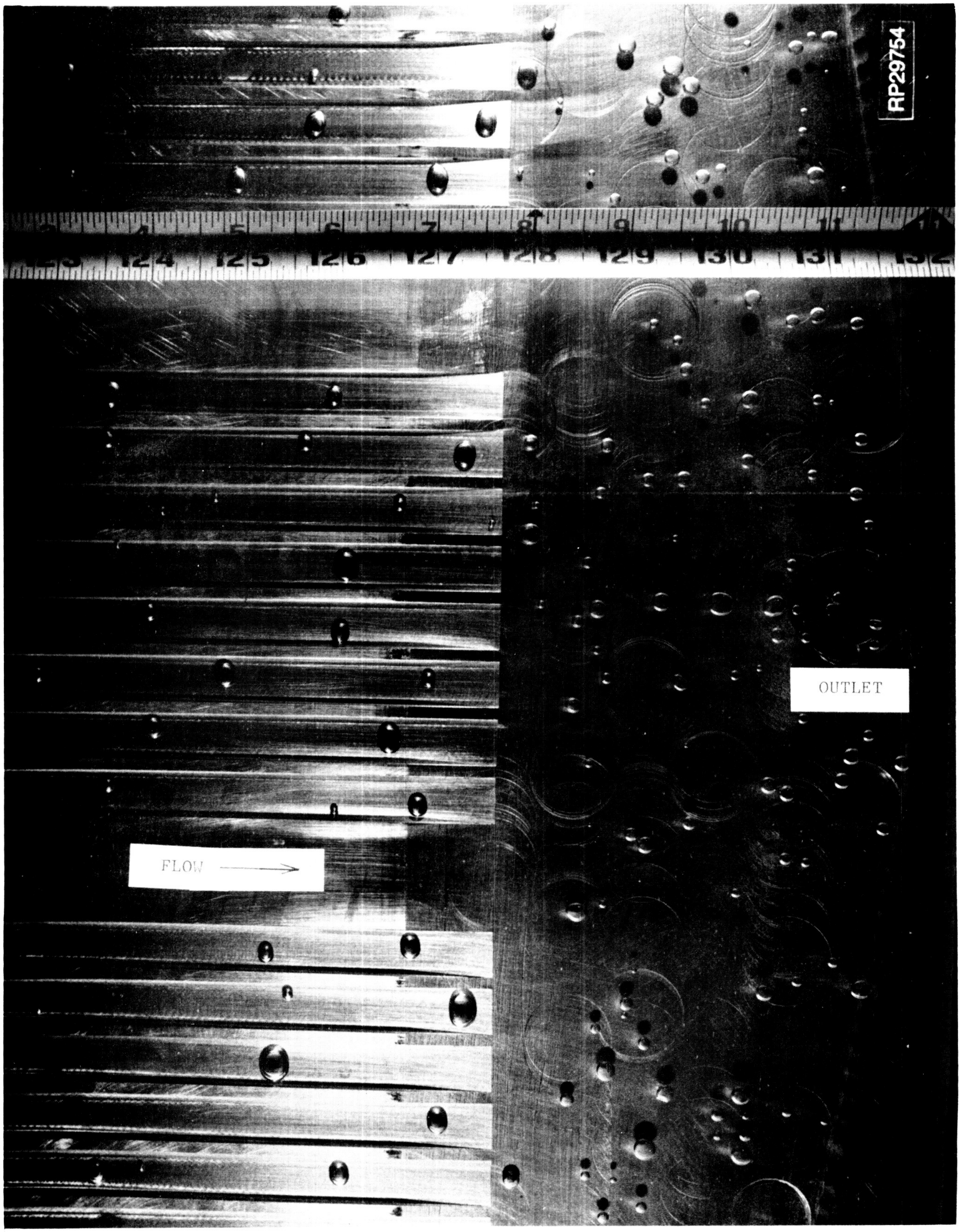


FIGURE 33 - CONDENSER OUTLET

Flow visualization along the length of the condenser channels was made possible by the glass cover on the condenser. Superheated toluene vapor entered the inlet manifold to the condenser and subcooled liquid exited, with the energy from the toluene transferred to the cooling water circuit. Figure 28 is an overview photograph of the condenser during operation. The superheated inlet is at the bottom of the photo and the subcooled liquid outlet is at the top of the photo.

Figures 29 through 33 are close-ups of condenser channels and document samples of the areas of interest. Figure 29 shows the inlet to the condenser channel and contains superheated toluene vapor. The vapor velocity at this point is approximately 200 feet per second and is not detectable by the naked eye. The active condenser (water outlet) begins at "8" inches. At the right edge of the photo, the film developing in the bottom of the channel shows as an apparent surface roughening. Figure 30 shows the 'V' formed on the glass top of the channel by liquid condensate at the perimeter of the channel. This is the start of the fully developed annular flow representative of shear controlled flow. Notice that the shear forces are capable of maintaining a liquid annulus in the channel even on the top surface of the channel. Figure 31 is also fully developed annular flow approximately 30 inches further along the channel. The highly turbulent nature of the film, which promotes heat transfer, is apparent. Figure 32 shows the liquid interface and transition to plug/slug flow. Note the larger "bubbles" close to the interface that decrease in volume as the bubble proceeds through the channel. The decreasing void volume is due to continued toluene condensation from the vapor bubble. The remaining bubble volume stays constant through the balance of the condenser and is noncondensible gas, most probably from air in-leakage as the peak temperatures in this program are well below temperatures that result in the pyrolytic degradation of toluene.[7] Figure 33 is the subcooled liquid outlet of the condenser and contains noncondensible gas bubbles due to a small air in-leaks in the plumbing. Notice that the sweeping action of the toluene flow (1 ft/sec) is capable of forcing the bubbles down the outlet tube and, ultimately, back to the RFMD. Videotapes of condenser operation were made and are available for review; 87336M has 20 minutes of raw footage, 87350-E is a 2.5 minute edited version of the raw footage.

A series of tests were performed that consisted of operating the condenser at a constant heat load and rotating the condenser from a horizontal position to an uphill position. The effect on condenser end-to-end pressure drop was recorded and is shown in Table 5. The first comparison was made between test points 1 and 2 (See Appendix 1). The mass flow rates for these two test points were approximately 10% different, so two additional points of identical mass flow rates were added to the table. For both sets of data points, the end-to-end pressure drop increased when the condenser was tilted uphill. This is as expected due to the static head associated with a change in end-to-end elevation. The measured increase agreed very well with the predictions. As previously stated, the visual appearance and the axial position of each flow regime did not change as the condenser orientation was changed.

The ability to separate and detect an accumulation of NC gases in the RFMD was very effective. The bubbles swept through the condenser were

TABLE 5

CONDENSER PRESSURE DROP
HORIZONTAL VERSUS TILTED UP

	TEST POINTS		ADDITIONAL POINTS	
	TEST POINT 1	TEST POINT 2	7-9-87 12:05	7-9-87 12:14
CONDENSER POSITION	HORIZONTAL	UP 2-1/2^{o}	HORIZONTAL	UP 2-1/2^{o}
MASS FLOW	1149	1084	1200	1218
PCI	2.2	2.2	2.4	2.4
-PCO	-1.4	-1.3	-1.7	-1.5
ΔP CALCULATED	0.8	0.9	0.7	0.9
ΔP TRANSDUCER	0.8	1.0	0.7	0.9

NOTE: ALL PRESSURES HAVE BEEN CORRECTED FOR STATIC HEAD DUE TO LIQUID COLUMNS ON THE PRESSURE TRANSDUCERS.

returned to the cold chamber of the RFMD, where they were trapped at the core of the rotating drum. This is due to the centrifugal action separating the liquid to the outside and leaving the less dense gas at the core. As the gas accumulates, the total chamber pressure rises above saturated conditions and can be detected by an external pressure transducer. Overboard venting was performed with a vacuum pump. Utilizing this technique resulted in a dramatic visual change in the size and quantity of bubbles in the condenser.

At the completion of the subject contract, the RFMD had accumulated a total of 111 starts and 48.4 hours of operation with toluene as the working fluid. The start loads on this style of bearing are the most severe condition. The high number of starts and operational hours verifies that toluene is an acceptable working fluid for this bearing application.

5.2.2 Secondary Technical Objective

The secondary technical objective was to compare predicted versus measured component performance at key operating points. The test points covered the full operating range of the SD-ORC. RFMD and condenser performance were predicted and compared to actual test data for the following conditions:

1. Receiver thermal energy storage status check

   -94% electrical power rating

2. Nominal insolation

   -100% electrical power rating

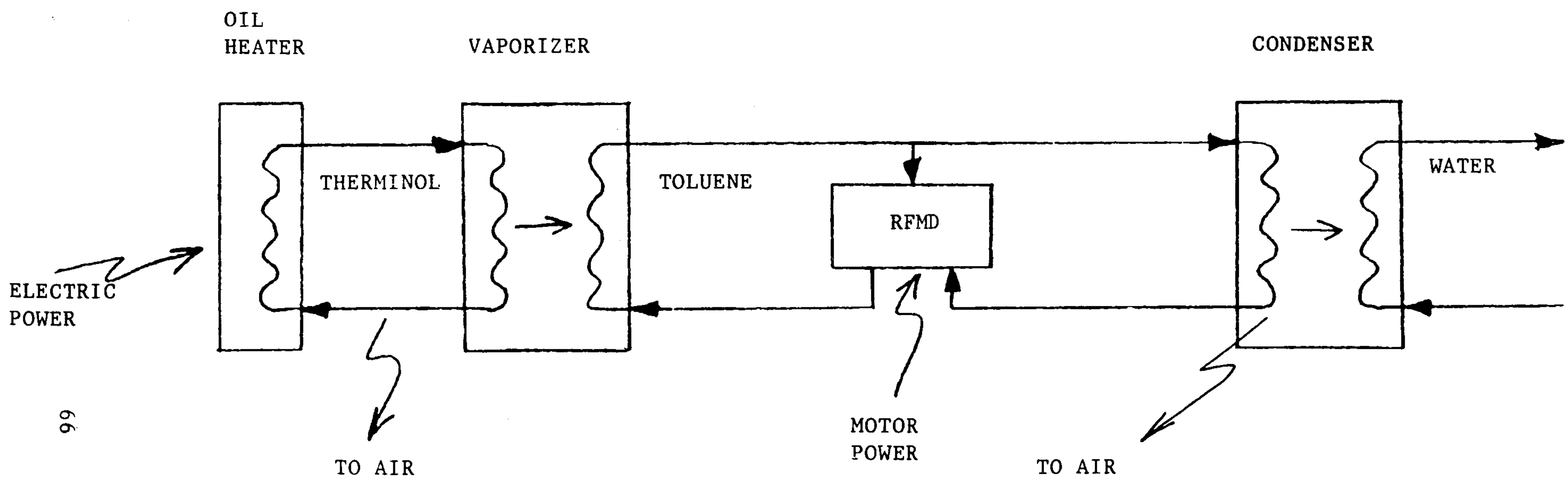
3. Maximum insolation

   -115% electrical power rating

The specific test matrix performed for this program is shown in Table 4. The comparison of measured versus predicted performance was completed for both key components. Samples of the raw data collected for the test points, an instrumentation list, and an instrumentation schematic are included in Appendix 1.

All of the testing parameters were monitored in real time and also recorded for later review. An 8-channel analog strip recorder provided a continuous record of key parameters. A 30-channel digital printout recorded all the instrumentation readings at automatic 10 second intervals. Post-test data reduction was performed on a component basis for both the RFMD and condenser.

A typical heat flow diagram for the test rig is shown in Figure 34. Electrical power was input to the heaters and, ultimately, the heat transfer oil. Toluene gained energy from the oil in the vaporizer and rejected energy to the water cooling loop. The diagram also presents the small amount of power input to the system due to the RFMD motor. Heater inefficiencies and air losses are also indicated.



132.1 Kw ELECTRIC INPUT	12.1 Kw LOSS TO AIR	120.0 Kw * THERMINOL LOSS/ TOLUENE GAIN	0.8 Kwe* MOTOR INPUT	2.6 Kwth LOSS TO AIR	116.6 Kwth*2 TOLUENE LOSS/ WATER GAIN

KEY:

* MEASURED VALUES-OTHERS ARE ASSUMED BASED ON ENERGY BALANCES
1 BASED ON MEASURED FLOW
2 BASED ON FLOW CALCULATED FROM RFMD HEAT BALANCE

FIGURE 34 - POWER FLOW DIAGRAM (115%)

Performance comparisons are presented for RFMD head-flow curve, RFMD motor power, and horizontal condenser end-to-end pressure drop. The RFMD pumping performance can be characterized by a total pitot probe mass flow versus pressure plot, more commonly called a head-flow curve. Figure 35 presents the measured and the predicted RFMD head-flow curve.

The predictions were for a speed of 3300 rpm, 5-inch pitot probe, 1-inch depth of submergence, and 120°F toluene. The measured data points are all at 3300 rpm, 5-inch pitot probe, and 120°F toluene. The flow coordinate for the measured data is the sum of the measured vaporizer flow plus the measured bearing flow plus an estimate for internal bearing flow. This estimate was assumed to be one-third of the measured bearing flow based on the ratio of equivalent sized orifices in each respective bearing. The additional variable for the measured data points was the depth of submergence of the probe. The noted values associated with each data point is the pressure offset on the accumulator. This offset pressure sets the liquid depth of the annulus in the rotating drum, i.e. the higher the pressure, the higher the annulus level. As expected, the pumping performance improved as the annulus depth increased. The general shape of the measured performance curve matches the predicted curve, but is approximately 25% lower than the predicted pressure rise. The significance of lower RFMD performance is that system flow rates would be lower than desired at the normal operating speed. To compensate for the lower flow, the RFMD speed (and as a result, motor power consumption) would increase. Since the head rise of the pitot pump is a function of speed squared, a 12% increase in speed would be required to compensate for a 25% lower head rise.

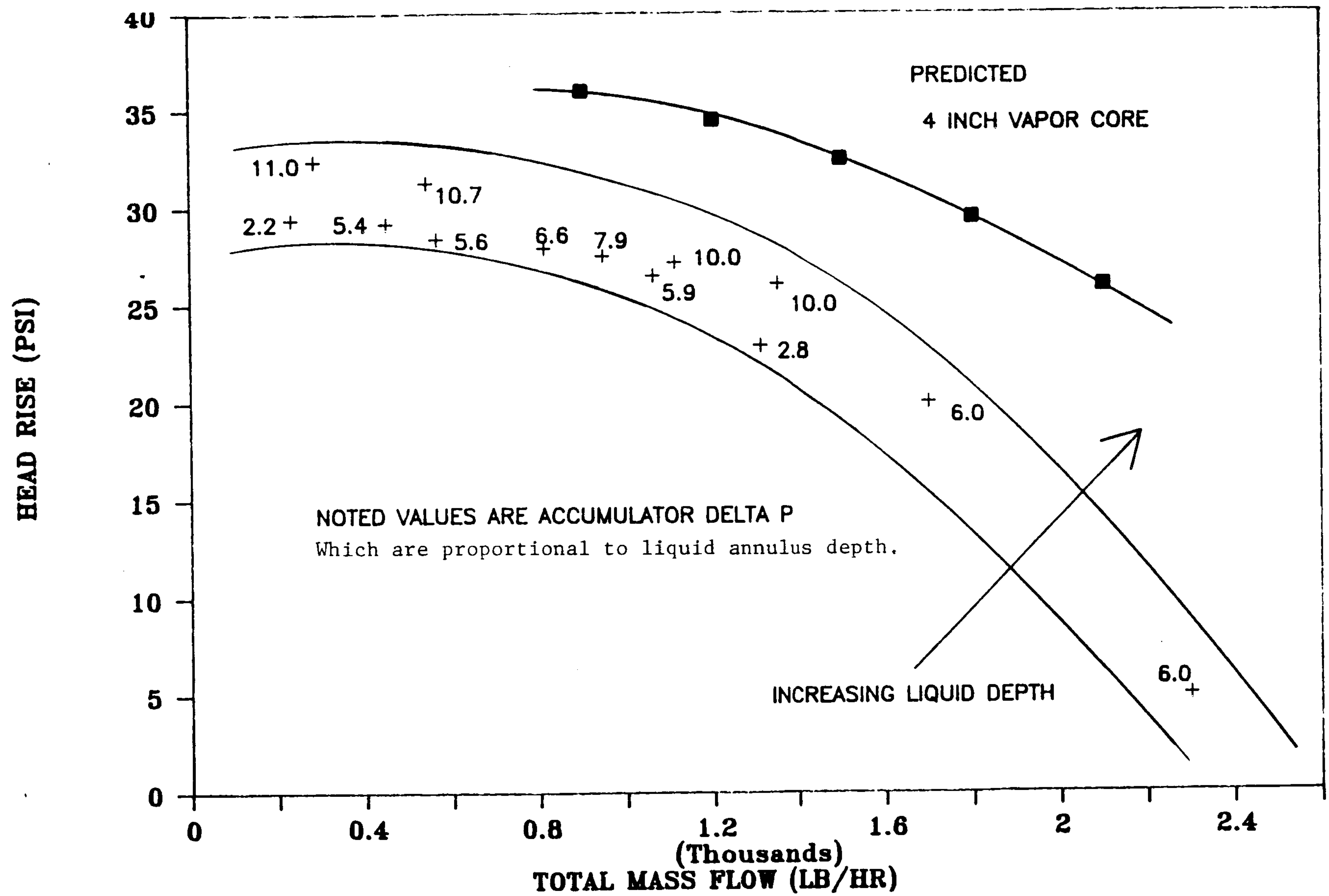


FIGURE 35 - RFMD HEAD-FLOW CURVE

The RFMD pitot pump pressure rise was predicted with the following equation:

$$P = \eta_i \left(\frac{\rho}{2g_c}\right) \left(\frac{\pi N}{60}\right)^2 \left[(Dp^2 - Di^2) + \eta_d \, Dp^2\right]$$

Where:

P = Pressure Rise (lb/ft^2)

η_i = Impeller Efficiency (.85)*

ρ = Density (lb/ft^3)

N = Speed (rpm)

Dp = Probe Diameter (ft)

Di = Liquid Interface Diameter (ft)

η_d = Diffuser Efficiency

(Function of Mass Flow Rate Valves from 0.64 to 1.00)*

* Based on previous empirical results

All of these parameters are straightforward except the impeller efficiency and diffuser efficiency. The assigned values for efficiency were based on previous RFMD experience (TPTMS preprototype unit) with Freon as the working fluid at 2000 rpm. Comparison of measured and predicted pitot pump pressure rise indicates that the assigned values for efficiency were higher than the actual efficiencies. The head-flow curves established by this program are more appropriate as a design basis for the Space Station application than the predicted curves shown. Future designs will be based on data generated from this program. It should be noted that even though the measured performance was below predictions, sufficient margin was provided in the design to successfully fulfill performance requirements of this test program.

A comparison of predicted RFMD motor power versus measured motor power for three test points is shown below:

Power Rating	Predicted (Watts)	Measured (Watts)
94%	706	661
100%	721	692
115%	937	830

Each of the three power ratings was performed at a different RFMD speed. A comparison of motor power predictions versus actuals is most appropriate when based upon speed. This is because the drag forces within the rotating drum consume the majority of the overall power requirements and the drag forces are functions of rotational speed. This chart shows that the measured motor power for each of the conditions was slightly lower than the predicted values for motor power at the same RFMD speed.

Condenser performance measurements primarily focused on end-to-end pressure drops. Comparison of measured and predicted pressure drops are given in the table below.

Mass Flow Rate	Pressure Drop	
	Measured	Predicted
(lb/hr)	(PSI)	(PSI)
1149	0.8	
1321	1.0	
1377		1.04
1495		1.24
1894	1.6	
2090		1.62

Please note that although attempts were made to duplicate pressure drop and mass flow conditions between measured and predicted cases, significant differences existed. It was therefore difficult to adequately evaluate the accuracy of the predictive models used. A more precise comparison in which the actual test inlet conditions are input to the model and predicted results compared to measured results should be performed. This activity is outside the scope of the current contract.

The measured pressures were corrected for static head due to different elevations and depth of liquid columns on the transducer as shown in Table 6.

The condenser pressure drop is important to the system designer because it ultimately determines the turbine back pressure. Higher back pressure results in a lower turbine head, and ultimately lower cycle efficiency. A series of ORC system model runs with varying condenser pressure drops was performed to determine the effect of cycle efficiency. As Figure 36

TABLE 6

CONDENSER PRESSURE DROP
COMPARISON (HORIZONTAL ORIENTATION)

		AS MEASURED (PSIA)	MEASURED CORRECTED FOR STATIC HEAD (PSIA)
TEST POINT 1 94%	CONDENSER INLET	2.4	2.2
	CONDENSER OUTLET	1.9	1.4
	ΔP CALCULATED	0.5	0.8
	ΔP TRANSDUCER	0.6	0.8
TEST POINT 4 100%	CONDENSER INLET	3.0	2.8
	CONDENSER OUTLET	2.4	1.9
	ΔP CALCULATED	0.6	0.9
	ΔP TRANSDUCER	0.8	1.0
TEST POINT 5 115%	CONDENSER INLET	5.2	5.0
	CONDENSER OUTLET	4.0	3.5
	ΔP CALCULATED	1.2	1.5
	ΔP TRANSDUCER	1.4	1.6

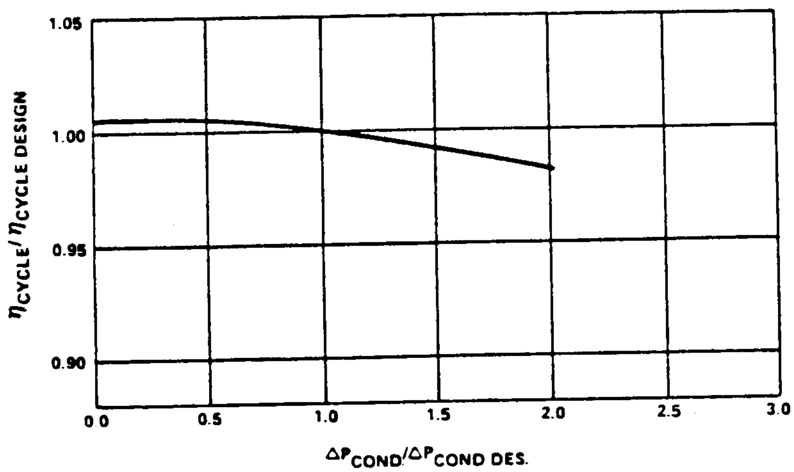


FIGURE 36

EFFECT OF CONDENSER PRESSURE DROP ON SYSTEM PERFORMANCE shows, a lower actual pressure drop than predicted results in an improvement in cycle efficiency. An error in the other direction (i.e. higher pressure drops than predicted) results in lower cycle efficiencies.

A final point of discussion is that the superheated toluene vapor temperatures at the condenser inlet were higher than the goal state point. The temperatures were 30-50°F higher than the goal temperatures used to predict condenser performance. This was due to performance of the commercial heat exchanger used to vaporize the toluene. At the goal state points, the vaporizer was not delivering dry toluene vapor to the condenser inlet. This was most probably due to liquid flowing through one of the parallel flow paths in the commercial heat exchanger used as a vaporizer. This required additional heat input to deliver dry vapor to the condenser. The additional superheat required the condenser to handle approximately 10% more thermal load than designed. Figure 37 shows that the bulk of the thermal load was due to the phase change of toluene and that the excess superheat resulted in a small increase in overall load. The additional load does not significantly change any of the conclusions drawn from the experiment.

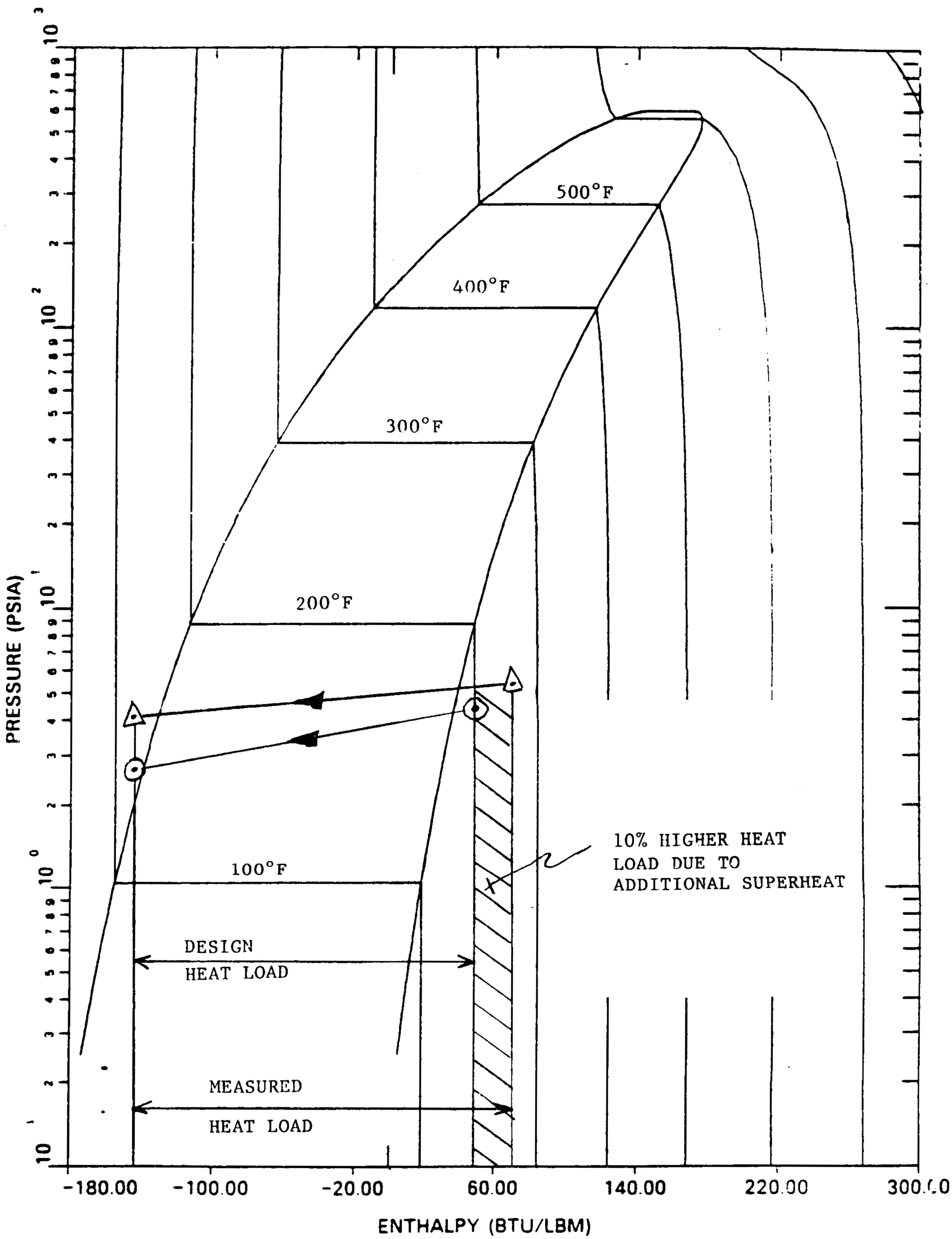


FIGURE 37 - CONDENSER HEAT LOAD (115%)

6.0 Conclusions

The technical objectives of this test program were successfully achieved. The first objective was to verify that the Rotary Fluid Management Device (RFMD) and the shear flow condenser are a viable approach to two-phase fluid management in a micro-gravity environment. The objective was achieved by operating the key components in neutral and adverse gravity orientations. The centrifugal forces of the RFMD and the shear forces of the condenser were dominant and capable of overcoming the adverse gravity forces on the ground. In a micro-gravity environment then, these forces would be very capable of controlling the fluid.

The secondary technical objective of this test program was to compare results of the analytical methods to experimental data. This objective was addressed by operating the test rig at key points covering the range of operation of the Solar Dynamic ORC and comparing subsequently the measured performance to pretest predictions. The predictions resulted from the same analytical tools used to predict the performance of the Space Station SD-ORC flight design.

Specific conclusions about RFMD performance can be made from the test program performed:

1. The RFMD performance and system performance were unaffected by changing the axis of rotation from horizontal to vertical.

2. The measured RFMD head-flow curve was approximately 25% lower

than pretest predictions. This was accommodated in testing by running higher speeds and a deeper annulus. Inherent efficiencies were, therefore, lower than expected.

3. The RFMD endured toluene and it is, therefore, an acceptable lubricant for fluid film bearings. The RFMD endured 111 starts and 48.4 hours of operation for the test duration with no detectable bearing wear.

4. The noncondensible gas separation and removal technique of the RFMD was very effective. The gas removal significantly affected the amount (reduced size and frequency of bubbles) of gas in the subcooled section. The effect on system performance over a fairly large range of noncondensible content was undetectable. No quantification of the noncondensible was attempted but qualitatively the amount appears to be very large relative to expected system levels.

Conclusions about condenser performance are:

1. The shear flow condenser operation and stability were not significantly affected by changing the channel orientations from horizontal to +2.5o uphill (adverse gradient). The only effect on condenser delta P was the increased static head as expected.

2. By visually studying the nature of the flow patterns in the

channels, shear controlled flow was established in all condenser channels, even at one-half the design flow rate.

3. NC gases were swept through the condenser and back to the RFMD. The configuration was capable of "sinking" NC gas bubbles.

4. The comparison between measured and predicted condenser delta P were made based on the first order effects of mass flow rate. Precise comparisons can only be made by using the model to predict exit conditions using measured inlet conditions as inputs. This modeling was beyond the scope of the current contract.

7.0 Recommendations

The test series described in this document successfully achieved all the stated technical objectives of the program. The capability of the test rig to provide additional performance and operational information should be exploited. A follow-on test series could include such items as:

1. Teardown of the RFMD to document the condition of the hardware.

2. Operation of the RFMD and condenser outside the range of the SD-ORC to further explore the margin available in both designs.

3. Quantify the amount of noncondensible gases present in the condenser and compare that to the expected level of NC gas in a SD-ORC due to pyrolytic degradation of toluene.

4. Determine the affect of varying amounts of noncondensible gases on condenser performance.

5. Determine the inclination at which liquid run-back occurs in the condenser by incrementally increasing the amount of inclination. Compare measured data to predictions.

6. Add additional instrumentation to the condenser to map the pressure and temperature distribution along a single toluene channel.

7. Modify existing instrumentation so that no post test adjustment need be made when comparing to predictions.

8. Revise predictions based on measured inlet conditions and perform precise comparison of measured to predicted performance. Verify model predicted sensitivities to various parameters.

8.0 References

1. Hall, C.N., and Mullin, J.P., "Dynamic Power - A Key to the Promise of Space," Twenty-Second Space Congress, April, 1985.

2. Brouns, et al., "Kilowatt Isotope Power System," IECEC 779283.

3. Sorensen, et al., "Status Report of the Dynamic Isotope Power System," IECEC 799299.

4. Prickett, et al., "Applications of the Dynamic Isotope Power System to a Multimission Spacecraft," IECEC 799301.

5. Raab, "Unique Features of Dynamic Isotope Power Systems for High Requirements Spacecraft," Proceedings of the 15th IECEC.

6. Grosskopf, W.J., and Binz, E.F., "Solar Dynamic Power for Space Station," Twenty-Third Space Congress, April, 1986.

7. AER 2741
   "Final Report-Toluene Stability Loop"

8. AER 2733
   "Test Plan for NASA-LeRC Space Station Advanced Development Contract - Task 2: Toluene RFMD/Shear Flow Condenser Test Stand"

9. EP2809-3101

   RFMD Layout Drawing

10. EP2809-3110

   RFMD Assembly Drawing

11. EP2809-3901

   Condenser Layout Drawing

12. EP2809-4501

   Task Loop Layout Drawing

13. EP2809-4511

   Condenser Support Structure

14. EP5531-24 and -27

   RFMD Support Structure

APPENDIX 1

TOLUENE RFMD INSTRUMENTATION

CHANNEL	LABEL	DESCRIPTION
1	TRLO	RFMD LIQUID OUT TEMPERATURE
2	TVI	VAPORIZER INLET TEMPERATURE
3	TTI	THERMINOL IN TEMPERATURE
4	TTO	THERMINOL OUT TEMPERATURE
5	TCI	CONDENSER IN TEMPERATURE
6	TCO	CONDENSER OUT TEMPERATURE
7	TRVI	RFMD VAPOR IN TEMPERATURE
8	TNCV	NONCONDENSIBLE VENT TEMPERATURE
9	TB	BEARING TEMPERATURE
10	TWI	WATER IN TEMPERATURE
11	TWO	WATER OUT TEMPERATURE
12	TA	AMBIENT TEMPERATURE
13	NS	RFMD SPEED
14	PCD	CONDENSER DELTA P
15	PRLO	RFMD LIQUID OUT PRESSURE
16	PVI	VAPORIZER INLET PRESSURE
17	PCI	CONDENSER INLET PRESSURE
18	PCO	CONDENSER OUTLET PRESSURE
19	PNCV	NONCONDENSIBLE VENT PRESSURE
20	PRVI	RFMD VAPOR IN PRESSURE
21	PAL	ACCUMULATOR LIQUID PRESSURE
22	SPARE	
23	WMI	RFMD MOTOR INPUT POWER
24	WHI	HEATER INPUT POWER
25	QV	VAPORIZER FLOW
26	QC	CONDENSER FLOW
27	QB	BEARING FLOW
28	QT	THERMINOL FLOW
29	QW	WATER FLOW
30	Q THERM	HEAT INPUT TO THERMINOL
31	Q WATER	HEAT REJECTED TO WATER

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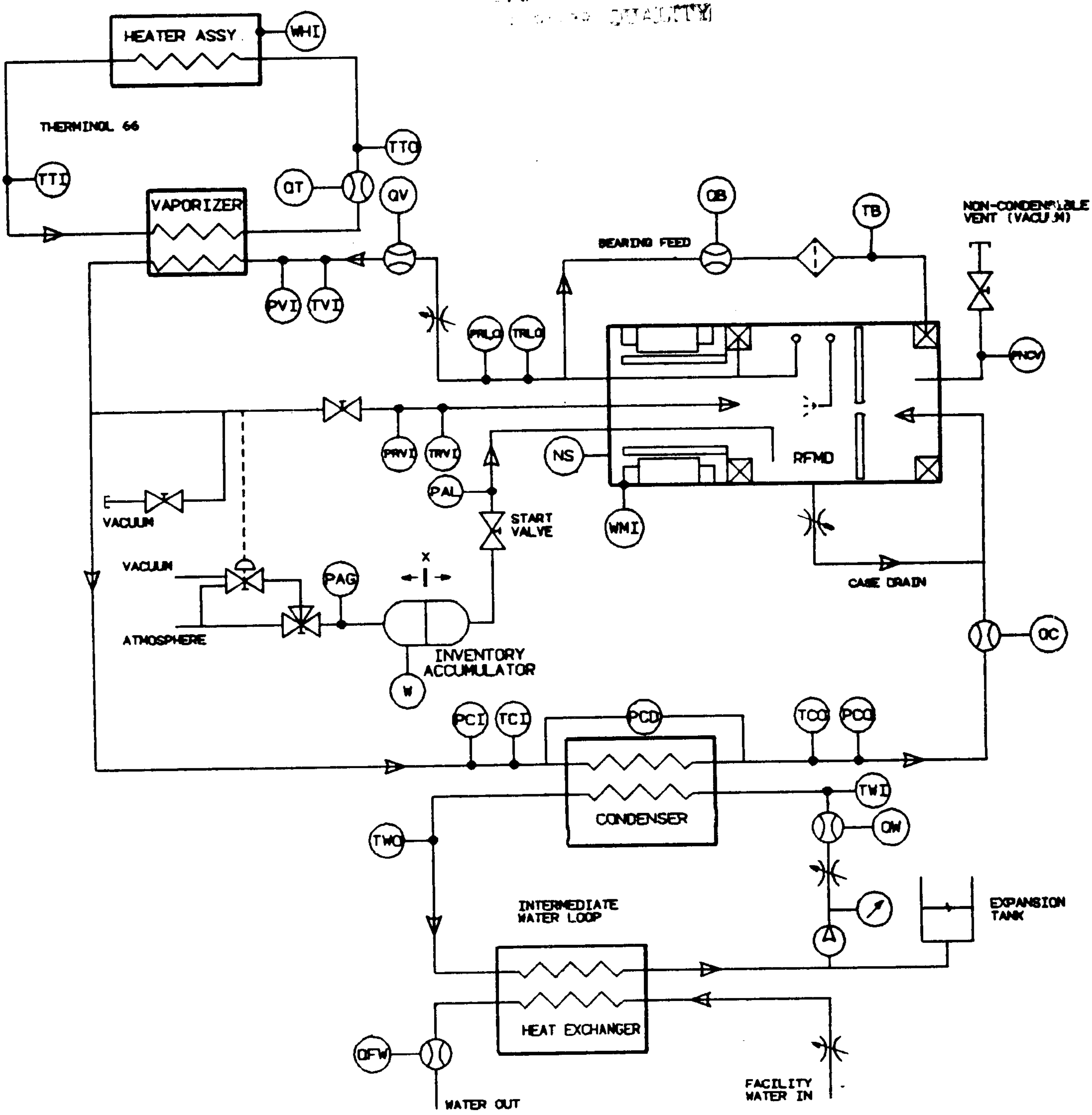


TOLUENE RFMD AND CONDENSER TEST LOOP INSTRUMENTATION

8-06-87

85% ELECTRICAL POWER RATING/TEST POINT #1

INPUT CONDITIONS	GOAL	ACTUAL (1)
RFMD POSITION	HORIZONTAL	HORIZONTAL
CONDENSER POSITION	HORIZONTAL	HORIZONTAL
CONDENSER--TOLUENE SIDE		
MASS FLOW RATE (LBM/HR)	1377	1149 (2)
INLET PRESSURE (PSIA)	2.1	2.4
INLET TEMPERATURE (DEG-F)	170	220
CONDENSER--WATER SIDE		
MASS FLOW RATE (GPM)	30	29.3
INLET TEMPERATURE (DEG-F)	82	85

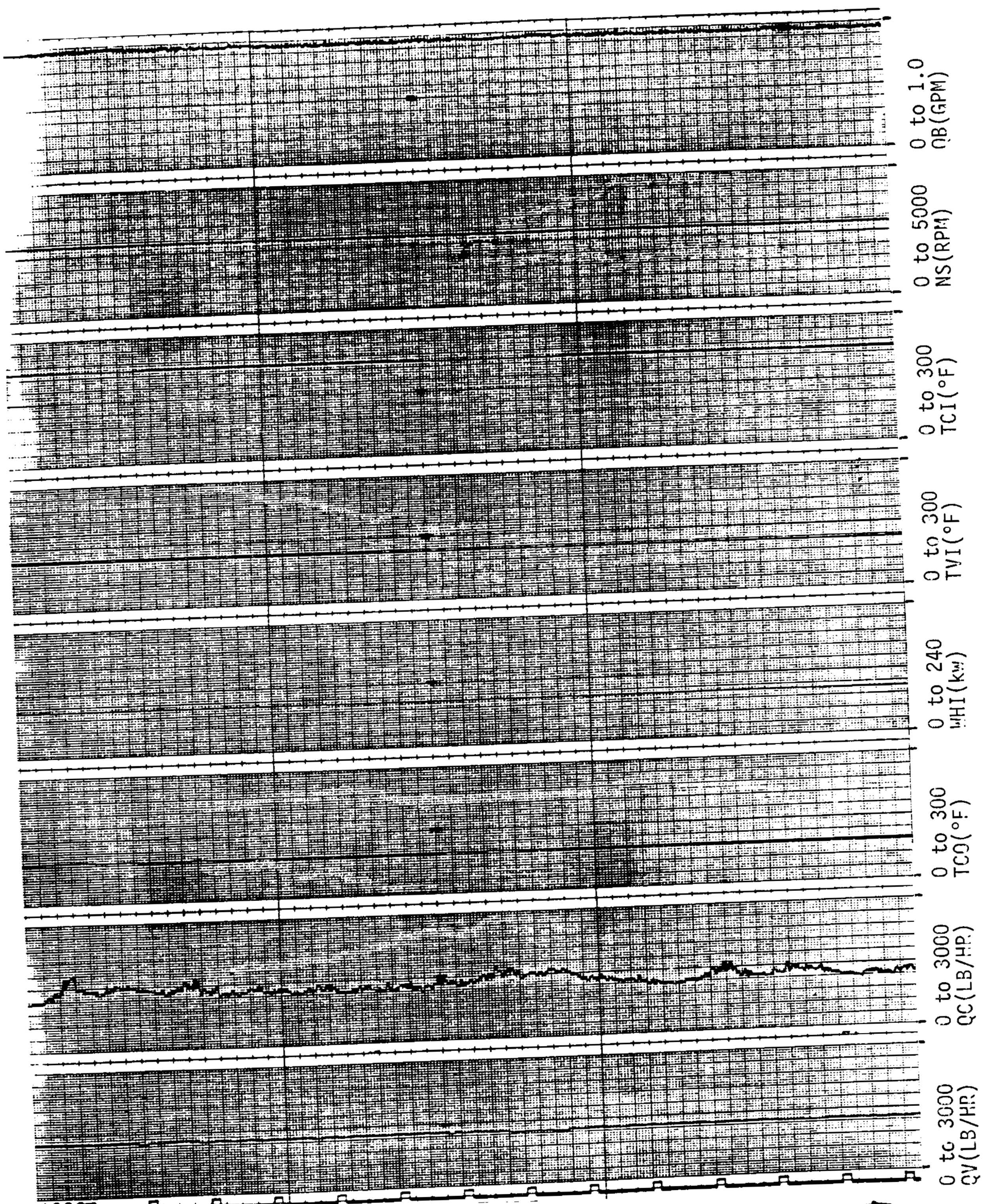
OUTPUT CONDITIONS	GOAL	ACTUAL (1)
ROTARY FLUID MGMT DEVICE (RFMD)		
MASS FLOW RATE (LBM/HR)	1585	1754 (3)
MOTOR INPUT POWER (WATTS)	706	661
MOTOR SPEED (RPM)	2900	2811
CONDENSER--TOLUENE SIDE		
OUTLET PRESSURE (PSIA)	1.06	1.9
OUTLET TEMPERATURE (DEG-F)	94	92
HEAT REJECTED (BTU/HR)	273,101	266,044
CONDENSER--WATER SIDE		
OUTLET TEMPERATURE (DEG-F)	100.5	103

COMMENTS OR NOTES

(1) UNLESS OTHERWISE NOTED, ACTUAL VALUES ARE THE AVERAGE OF THE (6) DIGITAL DATA POINTS SHOWN.

(2) QC FLOWMETER WAS NOT INDICATING THE CORRECT MASS FLOW DUE TO ENTRAINED AIR. NOTED VALUE WAS CALCULATED FROM A HEAT BALANCE OF THE RFMD USING THE MEASURED QV.

(3) TOTAL PITOT FLOW = (QV) + (4/3)*(QB)



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7/6/87

85 % POWER
RFMD & CONDENSER HORIZONTAL

DIR. OF PAPER TRAVEL

TOLUENE RFMD / CELL 62

DATA POINT:		1381	1382	1383	1384	1385	1386		
DATE:		7/6/87	7/6/87	7/6/87	7/6/87	7/6/87	7/6/87		
TIME:		13:58:6	13:58:16	13:58:26	13:58:36	13:58:46	13:58:57	UNITS	LABEL
CHNL	1	108.7	107.6	109.4	108.5	108.4	108.4	DEG F	TRLO
CHNL	2	107.3	107.6	107.5	107.2	107.6	107.5	DEG F	TVI
CHNL	3	227.4	232.5	229.7	227.7	228.0	227.7	DEG F	TTI
CHNL	4	214.9	215.0	215.0	215.0	215.3	213.0	DEG F	TTO
CHNL	5	220.0	220.7	220.6	220.0	218.7	219.7	DEG F	TCI
CHNL	6	92.1	92.0	92.2	91.9	92.7	92.4	DEG F	TCO
CHNL	7	158.3	158.5	158.2	157.9	158.3	157.7	DEG F	TRVI
CHNL	8	507.3	506.1	507.3	507.3	507.3	507.3	DEG F	TNCV
CHNL	9	108.5	108.6	101.4	108.6	107.9	108.2	DEG F	TB
CHNL	10	85.6	86.0	85.4	85.7	84.9	85.7	DEG F	TWI
CHNL	11	104.2	104.2	105.4	101.2	105.1	103.1	DEG F	TWO
CHNL	12	88.3	88.3	87.9	88.2	88.2	88.3	DEG F	TA
CHNL	13	2813.7	2816.3	2814.6	2812.5	2803.3	2813.6	RPM	NS
CHNL	14	0.6	0.7	0.6	0.6	0.6	0.6	PSID	PCD
CHNL	15	17.0	16.9	17.1	17.3	17.5	17.2	PSIA	PRLO
CHNL	16	3.8	4.0	3.8	4.4	4.1	3.7	PSIA	PVI
CHNL	17	2.4	2.5	2.4	2.6	2.4	2.4	PSIA	PCI
CHNL	18	1.9	1.9	1.9	2.0	1.9	1.9	PSIA	PCO
CHNL	19	0.9	0.9	0.9	0.9	0.9	0.9	PSIA	PNCV
CHNL	20	2.2	2.3	2.2	2.4	2.3	2.2	PSIA	PRVI
CHNL	21	8.5	8.4	8.4	8.4	8.6	8.4	PSIA	PAL
CHNL	22	0.0	0.0	0.0	0.0	0.0	0.0		SPARE
CHNL	23	661.0	628.4	647.5	666.5	716.3	652.2	WATTS	WMI
CHNL	24	83.7	84.2	84.1	84.1	83.9	83.8	KW	WHI
CHNL	25	1178.8	1173.9	1164.2	1168.4	1167.9	1177.4	LB/HR	QV
CHNL	26	1235.4	1343.6	1222.4	1349.3	1319.8	1159.2	LB/HR	QC
CHNL	27	1.0	1.0	1.0	1.0	1.0	1.0	GPM	QB
CHNL	28	106.4	106.4	106.4	106.3	106.3	106.3	GPM	QT
CHNL	29	29.3	29.2	29.3	29.3	29.3	29.2	GPM	QW
CHNL	30	81.3	114.0	95.7	83.0	82.4	95.7	KW	Q THERM
CHNL	31	79.2	77.3	85.0	66.0	85.8	74.0	KW	Q WATER

85% ELECTRICAL POWER RATING/TEST POINT #2

INPUT CONDITIONS	GOAL	ACTUAL (1)
RFMD POSITION	HORIZONTAL	HORIZONTAL
CONDENSER POSITION	UP	UP (+2.5 DEG)
CONDENSER--TOLUENE SIDE		
MASS FLOW RATE (LBM/HR)	1377	1084 (2)
INLET PRESSURE (PSIA)	2.1	2.4
INLET TEMPERATURE (DEG-F)	170	218
CONDENSER--WATER SIDE		
MASS FLOW RATE (GPM)	30	29.4
INLET TEMPERATURE (DEG-F)	82	85

OUTPUT CONDITIONS	GOAL	ACTUAL (1)
ROTARY FLUID MGMT DEVICE (RFMD)		
MASS FLOW RATE (LBM/HR)	1585	1635 (3)
MOTOR INPUT POWER (WATTS)	706	692
MOTOR SPEED (RPM)	2900	2803
CONDENSER--TOLUENE SIDE		
OUTLET PRESSURE (PSIA)	1.06	1.8
OUTLET TEMPERATURE (DEG-F)	94	92
HEAT REJECTED (BTU/HR)	273,101	275,606
CONDENSER--WATER SIDE		
OUTLET TEMPERATURE (DEG-F)	100.5	104

COMMENTS OR NOTES

(1) UNLESS OTHERWISE NOTED, ACTUAL VALUES ARE THE AVERAGE OF THE (6) DIGITAL DATA POINTS SHOWN.

(2) QC FLOWMETER WAS NOT INDICATING THE CORRECT MASS FLOW DUE TO ENTRAINED AIR. NOTED VALUE WAS CALCULATED FROM A HEAT BALANCE OF THE RFMD USING THE MEASURED QV.

(3) TOTAL PITOT FLOW = (QV) + (4/3)*(QB)

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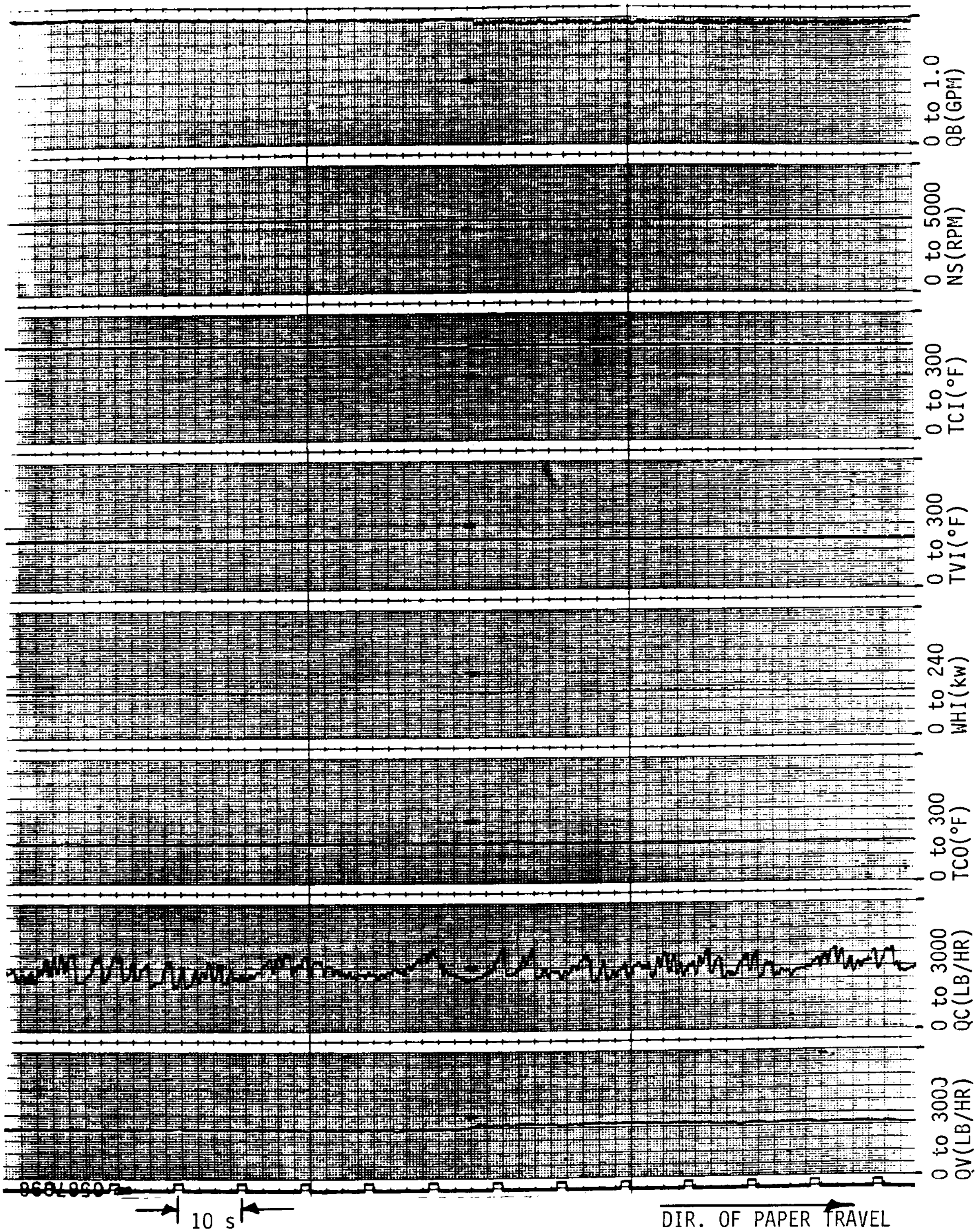

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85 % POWER
RFMD HORIZONTAL, CONDENSER UP 2½ °

TOLUENE RFMD / CELL 62

DATA POINT:		1405	1406	1407	1408	1409	1410		
DATE:		7/6/87	7/6/87	7/6/87	7/6/87	7/6/87	7/6/87		
TIME:		14:2:12	14:2:23	14:2:33	14:2:43	14:2:53	14:3:4	UNITS	LABEL
CHNL	1	108.5	108.3	108.0	108.7	108.5	107.9	DEG F	TRLO
CHNL	2	108.1	107.8	107.4	107.6	108.0	107.6	DEG F	TVI
CHNL	3	227.3	227.5	228.0	228.0	227.8	227.2	DEG F	TTI
CHNL	4	215.1	215.7	215.5	214.8	214.8	215.3	DEG F	TTO
CHNL	5	219.4	220.4	220.2	213.2	218.9	219.7	DEG F	TCI
CHNL	6	92.3	92.6	92.7	93.1	93.3	92.0	DEG F	TCO
CHNL	7	156.8	156.8	157.3	157.7	155.4	156.1	DEG F	TRVI
CHNL	8	507.3	507.3	506.5	508.1	506.9	507.3	DEG F	TNCV
CHNL	9	108.5	108.1	108.6	108.4	108.2	108.3	DEG F	TB
CHNL	10	82.7	85.6	85.6	85.1	85.7	85.3	DEG F	TWI
CHNL	11	104.3	105.5	104.5	104.8	104.7	104.4	DEG F	TWO
CHNL	12	88.0	87.9	88.0	87.7	88.1	88.1	DEG F	TA
CHNL	13	2815.4	2816.0	2809.3	2804.3	2787.5	2796.0	RPM	NS
CHNL	14	0.5	0.6	0.5	0.6	0.5	0.6	PSID	PCD
CHNL	15	16.9	17.3	17.1	17.9	17.7	18.0	PSIA	PRLO
CHNL	16	3.9	3.9	3.9	3.9	3.8	3.9	PSIA	PVI
CHNL	17	2.4	2.4	2.4	2.5	2.4	2.4	PSIA	PCI
CHNL	18	1.8	1.8	1.8	1.8	1.8	1.7	PSIA	PCO
CHNL	19	0.9	0.9	0.9	0.9	0.9	0.9	PSIA	PNCV
CHNL	20	2.3	2.3	2.3	2.3	2.2	2.2	PSIA	PRVI
CHNL	21	8.4	8.4	8.4	8.6	8.7	8.6	PSIA	PAL
CHNL	22	0.0	0.0	0.0	0.0	0.0	0.0		SPARE
CHNL	23	662.8	629.9	663.2	698.0	733.5	725.8	WATTS	WMI
CHNL	24	83.9	83.8	83.7	82.9	83.3	83.6	KW	WHI
CHNL	25	1161.6	1186.0	1169.0	1071.7	1051.8	1068.1	LB/HR	QV
CHNL	26	1746.1	1586.4	1789.7	1344.9	1381.4	1628.7	LB/HR	QC
CHNL	27	1.0	0.9	1.0	1.0	1.0	1.0	GPM	QB
CHNL	28	106.1	106.1	106.1	106.1	106.2	106.1	GPM	QT
CHNL	29	29.3	29.4	29.2	29.3	29.2	29.2	GPM	QW
CHNL	30	79.3	76.6	81.0	85.8	84.4	77.1	KW	Q THERM
CHNL	31	92.0	84.6	80.1	83.6	80.4	81.1	KW	Q WATER

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85% ELECTRICAL POWER RATING/TEST POINT #3

INPUT CONDITIONS	GOAL	ACTUAL (1)
RFMD POSITION	VERTICAL	VERTICAL
CONDENSER POSITION	UP (+2.5 DEG)	UP (+2.5 DEG)
CONDENSER--TOLUENE SIDE		
MASS FLOW RATE (LBM/HR)	1377	1033 (2)
INLET PRESSURE (PSIA)	2.1	2.3
INLET TEMPERATURE (DEG-F)	170	224
CONDENSER--WATER SIDE		
MASS FLOW RATE (GPM)	30	29.4
INLET TEMPERATURE (DEG-F)	82	85
OUTPUT CONDITIONS		
ROTARY FLUID MGMT DEVICE (RFMD)		
MASS FLOW RATE (LBM/HR)	1585	1700 (3)
MOTOR INPUT POWER (WATTS)	706	685
MOTOR SPEED (RPM)	2900	2805
CONDENSER--TOLUENE SIDE		
OUTLET PRESSURE (PSIA)	1.06	1.6
OUTLET TEMPERATURE (DEG-F)	94	92
HEAT REJECTED (BTU/HR)	273,101	255,286
CONDENSER--WATER SIDE		
OUTLET TEMPERATURE (DEG-F)	100.5	104

COMMENTS OR NOTES

(1) UNLESS OTHERWISE NOTED, ACTUAL VALUES ARE THE AVERAGE OF THE (6) DIGITAL DATA POINTS SHOWN.

(2) QC FLOWMETER WAS NOT INDICATING THE CORRECT MASS FLOW DUE TO ENTRAINED AIR. NOTED VALUE WAS CALCULATED FROM A HEAT BALANCE OF THE RFMD USING THE MEASURED QV.

(3) TOTAL PITOT FLOW = (QV) + (4/3)*(QB)

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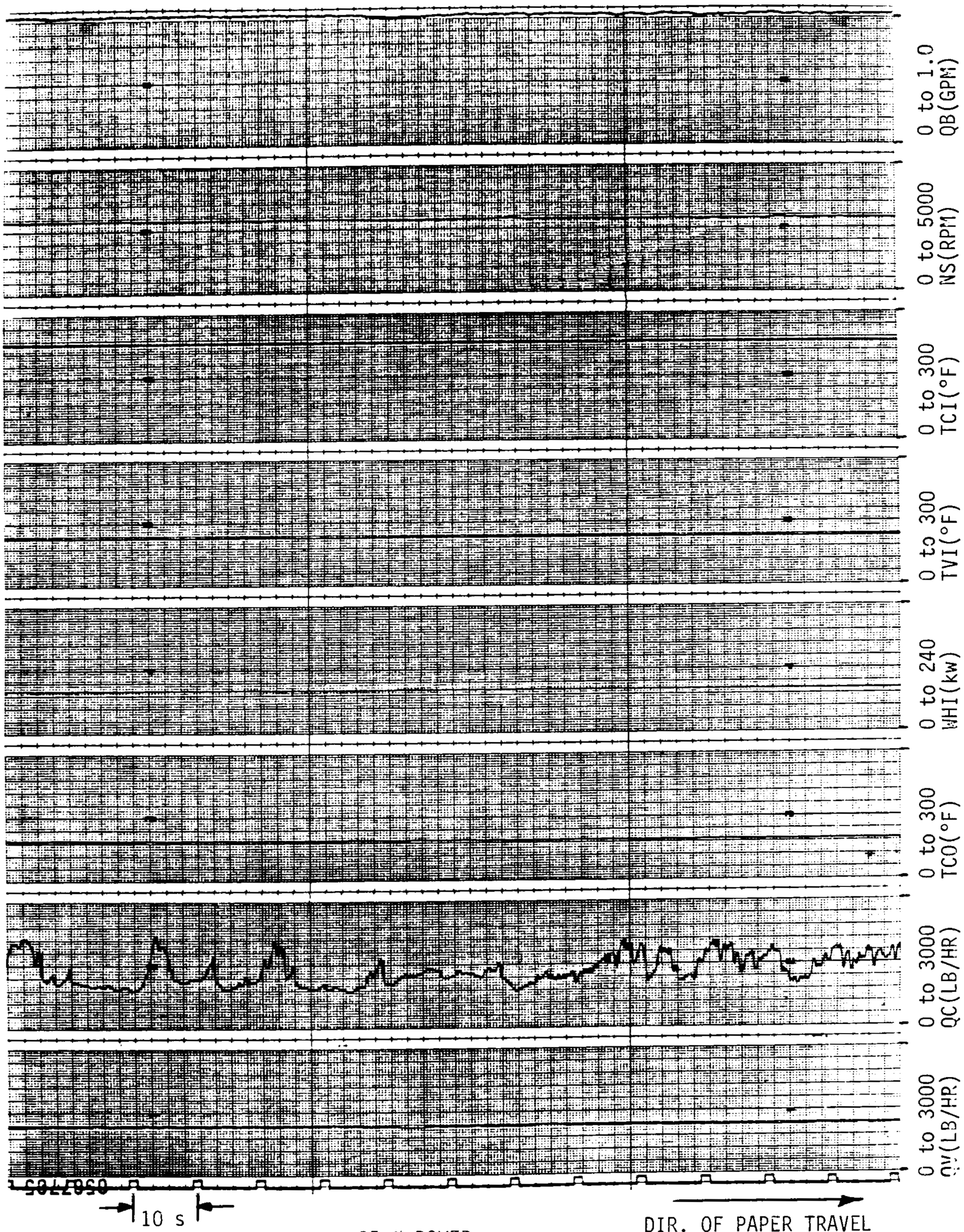

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7/6/87

85 % POWER
RFMD VERTICAL, CONDENSER OUTLET UP 2½°

TOLUENE RFMD / CELL 62

DATA POINT:		1525	1526	1527	1528	1529	1530		
DATE:		7/6/87	7/6/87	7/6/87	7/6/87	7/6/87	7/6/87		
TIME:		14:22:49	14:22:59	14:23:10	14:23:20	14:23:31	14:26:42	UNITS	LABEL
CHNL	1	106.7	107.7	106.7	106.6	106.5	107.2	DEG F	TRLO
CHNL	2	106.4	106.8	107.2	106.3	106.8	106.8	DEG F	TVI
CHNL	3	230.7	229.5	230.9	230.9	230.9	231.3	DEG F	TTI
CHNL	4	218.7	219.4	218.7	220.2	219.5	218.7	DEG F	TTO
CHNL	5	224.4	224.6	224.3	225.0	224.1	224.5	DEG F	TCI
CHNL	6	91.6	91.0	92.3	91.5	91.9	91.5	DEG F	TCO
CHNL	7	155.3	153.1	155.2	154.7	155.8	155.8	DEG F	TRVI
CHNL	8	508.1	513.4	507.7	513.0	507.7	507.7	DEG F	TNCV
CHNL	9	106.7	107.7	105.7	106.7	106.9	107.2	DEG F	TB
CHNL	10	85.5	85.5	85.5	84.4	85.3	85.1	DEG F	TWI
CHNL	11	103.1	102.8	102.1	102.5	102.9	103.0	DEG F	TWO
CHNL	12	89.2	89.1	89.2	89.1	89.4	89.3	DEG F	TA
CHNL	13	2804.7	2818.0	2798.9	2818.8	2790.1	2790.1	RPM	NS
CHNL	14	0.6	0.5	0.5	0.5	0.6	0.8	PSID	PCD
CHNL	15	17.0	16.9	16.6	17.0	16.8	16.7	PSIA	PRLO
CHNL	16	3.6	3.7	3.6	3.4	3.7	3.6	PSIA	PVI
CHNL	17	2.3	2.3	2.2	2.3	2.3	2.4	PSIA	PCI
CHNL	18	1.6	1.7	1.6	1.6	1.6	1.5	PSIA	PCO
CHNL	19	1.3	1.3	1.3	1.3	1.3	1.3	PSIA	PNCV
CHNL	20	2.3	2.3	2.2	2.3	2.3	2.4	PSIA	PRVI
CHNL	21	8.6	8.4	8.6	8.4	8.7	8.6	PSIA	PAL
CHNL	22	0.0	0.0	0.0	0.0	0.0	0.0		SPARE
CHNL	23	677.1	627.3	699.5	667.2	747.8	736.5	WATTS	WMI
CHNL	24	77.8	79.0	78.1	78.3	78.4	77.8	KW	WHI
CHNL	25	1064.0	1049.4	1063.8	1051.8	1034.2	1050.5	LB/HR	QV
CHNL	26	1794.3	1508.8	1448.6	1117.9	1891.9	1188.2	LB/HR	QC
CHNL	27	1.0	1.0	1.0	1.0	1.0	1.0	GPM	QB
CHNL	28	105.0	104.9	105.0	105.0	105.0	105.0	GPM	QT
CHNL	29	29.3	29.3	29.4	29.5	29.4	29.4	GPM	QW
CHNL	30	77.5	65.0	78.2	68.8	73.4	81.1	KW	Q THERM
CHNL	31	75.0	73.5	70.8	77.5	75.2	76.5	KW	Q WATER

100% ELECTRICAL POWER RATING/TEST POINT #4

INPUT CONDITIONS	GOAL	ACTUAL (1)
RFMD POSITION	HORIZONTAL	HORIZONTAL
CONDENSER POSITION	HORIZONTAL	HORIZONTAL
CONDENSER--TOLUENE SIDE		
MASS FLOW RATE (LBM/HR)	1495	1321 (2)
INLET PRESSURE (PSIA)	2.44	3.0
INLET TEMPERATURE (DEG-F)	176	203
CONDENSER--WATER SIDE		
MASS FLOW RATE (GPM)	30	29.4
INLET TEMPERATURE (DEG-F)	87	89

OUTPUT CONDITIONS		
ROTARY FLUID MGMT DEVICE (RFMD)		
MASS FLOW RATE (LBM/HR)	1702	1749 (3)
MOTOR INPUT POWER (WATTS)	721	692
MOTOR SPEED (RPM)	2950	2930
CONDENSER--TOLUENE SIDE		
OUTLET PRESSURE (PSIA)	1.2	2.4
OUTLET TEMPERATURE (DEG-F)	100	97
HEAT REJECTED (BTU/HR)	296,667	308,449
CONDENSER--WATER SIDE		
OUTLET TEMPERATURE (DEG-F)	107	110

COMMENTS OR NOTES

(1) UNLESS OTHERWISE NOTED, ACTUAL VALUES ARE THE AVERAGE OF THE (6) DIGITAL DATA POINTS SHOWN.

(2) QC FLOWMETER WAS NOT INDICATING THE CORRECT MASS FLOW DUE TO ENTRAINED AIR. NOTED VALUE WAS CALCULATED FROM A HEAT BALANCE OF THE RFMD USING THE MEASURED QV.

(3) TOTAL PITOT FLOW = (QV) + (4/3)*(QB)

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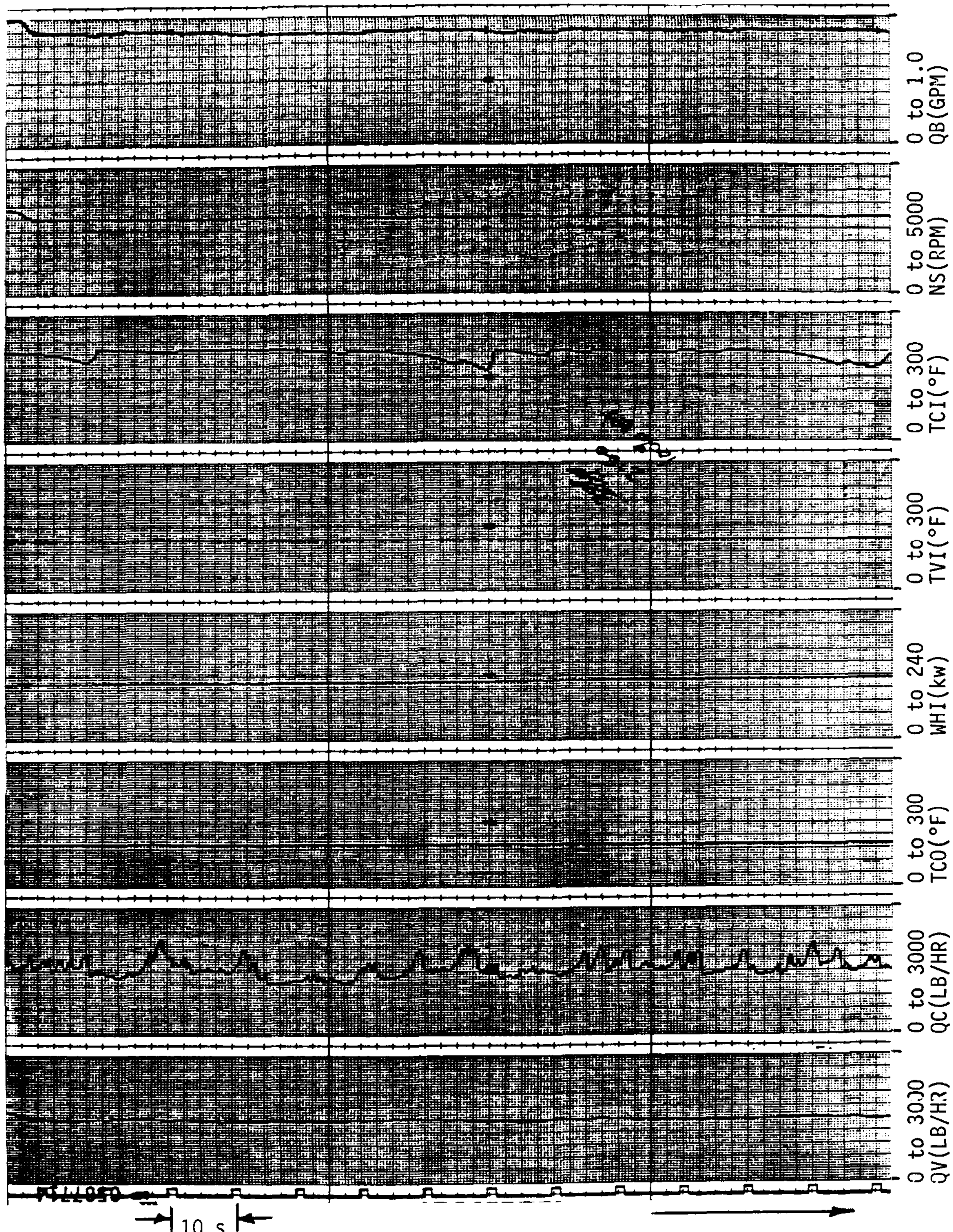

DIR. OF PAPER TRAVEL

14:48
7/6/87

100 % POWER
RFMD & CONDENSER HORIZONTAL

C-2

TOLUENE RFMD / CELL 62

DATA POINT:		1657	1658	1659	1660	1661	1662		
DATE:		7/6/87	7/6/87	7/6/87	7/6/87	7/6/87	7/6/87		
TIME:		14:48:28	14:48:38	14:48:49	14:48:59	14:49:9	14:49:19	UNITS	LABEL
CHNL	1	112.2	112.4	112.5	113.6	112.9	113.7	DEG F	TRLO
CHNL	2	111.5	111.7	111.5	111.7	112.0	112.0	DEG F	TVI
CHNL	3	230.2	230.8	231.9	232.6	232.4	232.8	DEG F	TTI
CHNL	4	215.7	216.1	216.1	216.7	216.8	217.7	DEG F	TTO
CHNL	5	202.8	202.5	183.2	206.0	209.1	215.1	DEG F	TCI
CHNL	6	97.5	97.8	96.5	97.0	97.4	97.3	DEG F	TCO
CHNL	7	157.0	157.5	157.7	157.8	157.5	158.5	DEG F	TRVI
CHNL	8	507.7	507.3	507.3	507.7	508.1	509.7	DEG F	TNCV
CHNL	9	112.1	112.2	112.5	112.9	112.2	112.6	DEG F	TB
CHNL	10	87.1	89.0	89.2	89.0	89.2	88.7	DEG F	TWI
CHNL	11	109.0	110.6	111.3	109.0	109.5	109.7	DEG F	TWO
CHNL	12	88.7	88.3	88.2	89.4	88.2	88.4	DEG F	TA
CHNL	13	2922.7	2936.6	2921.6	2923.0	2939.5	2938.8	RPM	NS
CHNL	14	0.8	0.9	0.8	0.8	0.7	0.8	PSID	PCD
CHNL	15	16.4	16.1	16.4	16.1	15.7	15.8	PSIA	PRLO
CHNL	16	4.7	4.7	4.5	4.4	4.4	4.8	PSIA	PVI
CHNL	17	3.0	3.1	3.0	2.9	2.9	3.1	PSIA	PCI
CHNL	18	2.4	2.4	2.4	2.3	2.3	2.4	PSIA	PCO
CHNL	19	1.2	1.2	1.2	1.2	1.3	1.3	PSIA	PNCV
CHNL	20	2.8	2.8	2.8	2.7	2.7	2.9	PSIA	PRVI
CHNL	21	8.9	8.9	8.9	8.9	8.6	8.8	PSIA	PAL
CHNL	22	0.0	0.0	0.0	0.0	0.0	0.0		SPARE
CHNL	23	709.0	680.8	729.5	705.0	654.4	678.2	WATTS	WMI
CHNL	24	107.6	107.3	106.2	105.0	104.9	104.2	KW	WHI
CHNL	25	1365.6	1336.3	1379.0	1375.3	1349.7	1330.4	LB/HR	QV
CHNL	26	1841.9	1353.0	1537.9	1640.6	1295.1	1476.2	LB/HR	QC
CHNL	27	0.9	0.9	0.9	0.9	0.9	0.9	GPM	QB
CHNL	28	106.0	105.9	105.8	105.4	105.6	105.4	GPM	QT
CHNL	29	29.4	29.3	29.3	29.4	29.5	29.5	GPM	QW
CHNL	30	94.2	94.8	102.4	102.3	100.6	97.2	KW	Q THERM
CHNL	31	93.4	92.2	94.1	85.3	86.9	90.0	KW	Q WATER

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115% ELECTRICAL POWER RATING/TEST POINT #5

INPUT CONDITIONS	GOAL	ACTUAL (1)
RFMD POSITION	HORIZONTAL	HORIZONTAL
CONDENSER POSITION	HORIZONTAL	HORIZONTAL
CONDENSER--TOLUENE SIDE		
MASS FLOW RATE (LBM/HR)	2090	1894 (2)
INLET PRESSURE (PSIA)	4.3	5.2
INLET TEMPERATURE (DEG-F)	198	261
CONDENSER--WATER SIDE		
MASS FLOW RATE (GPM)	30	29.8
INLET TEMPERATURE (DEG-F)	107	110

OUTPUT CONDITIONS	GOAL	ACTUAL (1)
ROTARY FLUID MGMT DEVICE (RFMD)		
MASS FLOW RATE (LBM/HR)	2293	2212 (3)
MOTOR INPUT POWER (WATTS)	937	830
MOTOR SPEED (RPM)	3300	3275
CONDENSER--TOLUENE SIDE		
OUTLET PRESSURE (PSIA)	2.7	4.0
OUTLET TEMPERATURE (DEG-F)	123	125
HEAT REJECTED (BTU/HR)	409,896	413,011
CONDENSER--WATER SIDE		
OUTLET TEMPERATURE (DEG-F)	134.5	138

COMMENTS OR NOTES

(1) UNLESS OTHERWISE NOTED, ACTUAL VALUES ARE THE AVERAGE OF THE (6) DIGITAL DATA POINTS SHOWN.

(2) QC FLOWMETER WAS NOT INDICATING THE CORRECT MASS FLOW DUE TO ENTRAINED AIR. NOTED VALUE WAS CALCULATED FROM A HEAT BALANCE OF THE RFMD USING THE MEASURED QV.

(3) TOTAL PITOT FLOW = (QV) + (4/3)*(QB)

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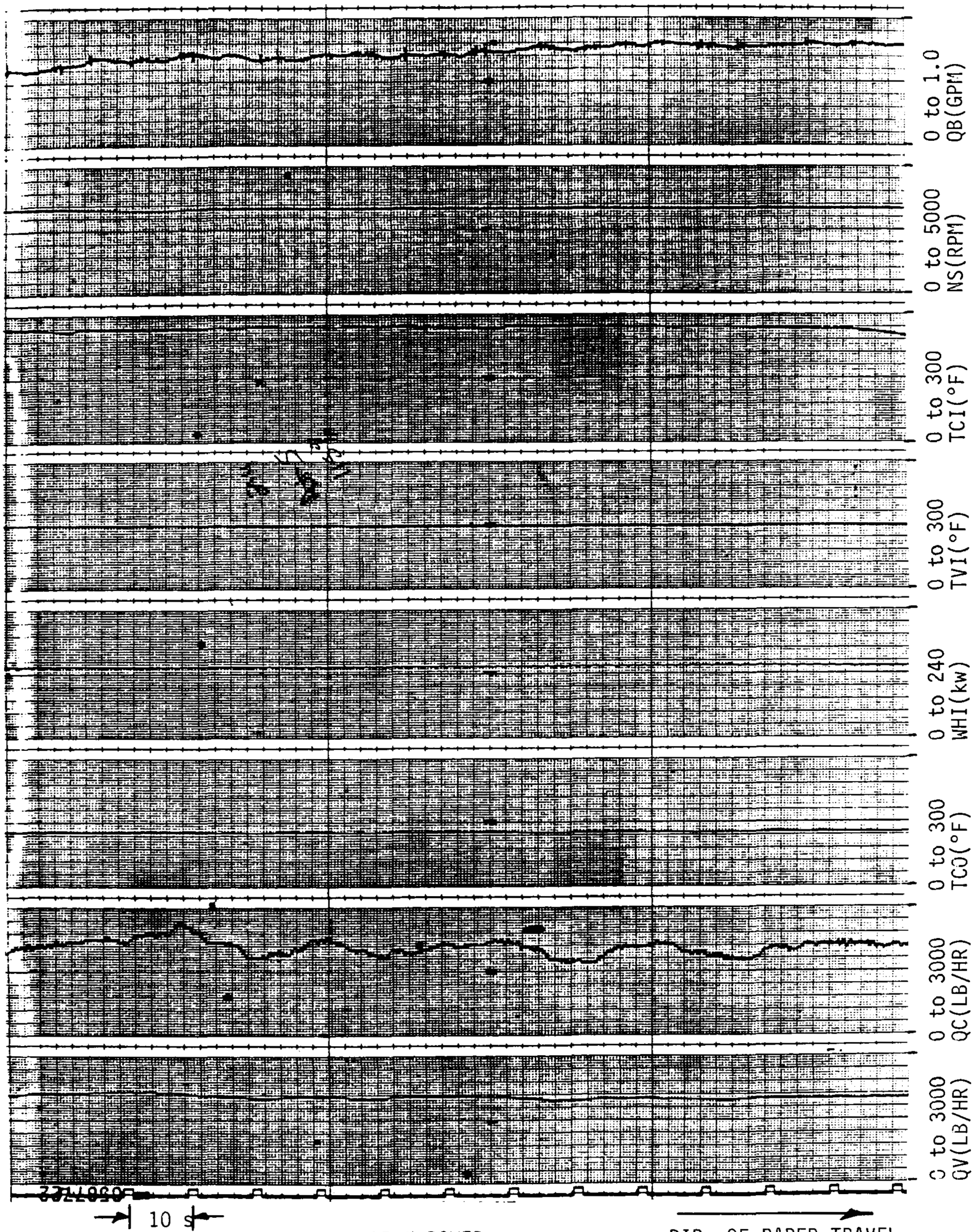

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115 % POWER
REMD & CONDENSER HORIZONTAL

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TOLUENE RFMD / CELL 62

DATA POINT:		1789	1790	1791	1792	1793	1794		
DATE:		7/6/87	7/6/87	7/6/87	7/6/87	7/6/87	7/6/87		
TIME:		15:11:8	15:11:19	15:11:29	15:11:39	15:11:49	15:12:0	UNITS	LABEL
CHNL	1	140.5	140.0	140.5	141.0	140.8	141.3	DEG F	TRLD
CHNL	2	141.1	141.4	140.9	141.1	141.5	141.2	DEG F	TVI
CHNL	3	281.1	279.6	280.8	280.6	280.6	280.3	DEG F	TTI
CHNL	4	252.4	253.5	253.5	253.6	253.3	252.4	DEG F	TTO
CHNL	5	266.7	248.6	265.2	264.8	260.6	264.5	DEG F	TCI
CHNL	6	124.2	126.5	125.4	126.3	126.2	125.5	DEG F	TCO
CHNL	7	205.3	204.5	206.2	206.8	206.9	206.7	DEG F	TRVI
CHNL	8	508.1	508.5	507.7	508.1	508.5	508.5	DEG F	TNCV
CHNL	9	140.6	140.6	139.8	139.8	140.3	140.2	DEG F	TB
CHNL	10	110.8	110.6	111.0	110.6	110.3	110.2	DEG F	TWI
CHNL	11	138.4	138.8	138.1	138.7	139.2	138.2	DEG F	TWO
CHNL	12	81.2	82.2	82.8	83.0	83.1	83.2	DEG F	TA
CHNL	13	3281.6	3275.8	3276.4	3268.0	3271.0	3282.6	RPM	NS
CHNL	14	1.5	1.2	1.5	1.6	1.4	1.2	PSID	PCD
CHNL	15	10.5	9.7	10.5	10.8	10.2	10.0	PSIA	PRLD
CHNL	16	7.4	6.4	7.5	7.4	6.6	6.9	PSIA	PVI
CHNL	17	5.4	4.9	5.4	5.4	5.2	5.0	PSIA	PCI
CHNL	18	4.1	3.9	4.1	4.1	4.1	4.0	PSIA	PCO
CHNL	19	2.4	2.3	2.4	2.3	2.3	2.4	PSIA	PNCV
CHNL	20	5.1	4.7	5.1	5.1	4.9	4.8	PSIA	PRVI
CHNL	21	10.7	10.6	10.4	10.9	10.9	10.7	PSIA	PAL
CHNL	22	0.0	0.0	0.0	0.0	0.0	0.0		SPARE
CHNL	23	815.2	827.3	830.2	855.1	841.2	811.9	WATTS	WMI
CHNL	24	133.0	132.6	132.6	132.0	131.3	131.3	KW	WHI
CHNL	25	1926.5	1966.7	1931.2	1958.0	1978.3	1944.4	LB/HR	QV
CHNL	26	2152.0	2098.9	2160.5	2307.1	2219.6	1986.9	LB/HR	QC
CHNL	27	0.6	0.6	0.6	0.6	0.6	0.6	GPM	QB
CHNL	28	93.5	92.8	92.8	92.5	92.6	93.0	GPM	QT
CHNL	29	29.7	29.8	29.8	29.7	29.8	29.7	GPM	QW
CHNL	30	164.5	147.8	154.9	153.0	154.8	158.5	KW	Q THERM
CHNL	31	119.2	122.3	116.9	121.5	125.0	120.7	KW	Q WATER

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15. Supplementary Notes

Project Manager, Myron E. Hill, Solar Dynamic Power Module Division, NASA Lewis Research Center.

16. Abstract

Management of two-phase fluid and control of the heat transfer process in micro-gravity is a technical challenge that must be addressed for an orbital Organic Rankine Cycle (ORC) application. A test program was performed in 1-g that satisfactorily demonstrated the two-phase management capability of the rotating fluid management device (RFMD) and shear-flow condenser. Operational tests of the RFMD and shear flow condenser in adverse gravity orientations, confirmed that the centrifugal forces in the RFMD and the shear forces in the condenser, were capable of overcoming gravity forces. In a micro-gravity environment, these same forces would not have to compete against gravity and would therefore be dominant. The specific test program covered the required operating range of the Space Station Solar Dynamic Organic Rankine Cycle power system. Review of the test data verified that: fluid was pumped from the RFMD in all attitudes; subcooled states in the condenser were achieved; condensate was pushed uphill against gravity; and noncondensible gases were swept through the condenser.

17. Key Words (Suggested by Author(s)) Power system; Organic Rankine cycle; Two phase; Toluene; Shear flow condenser	18. Distribution Statement Unclassified - Unlimited Subject Category 20

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