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# WATER SEPARATION BY SELECTIVE PERMEATION THROUGH MICROPOROUS MATERIALS

J. R. Katz

## FOREWORD

This report was prepared by the Missile and Space Division, General Electric Company, Valley Forge, Pennsylvania, under USAF Contract No. AF 33(615)-1475. The contract was initiated under Project No. 6146 "Atmosphere and Thermal Control", Task No. 614611 "Carbon Dioxide and Water Vapor Control Techniques." The work was administered under the direction of the Air Force Flight Dynamics Laboratory, Research and Technology Division, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio. Mr. Jack J. Fedderke was the Project Engineer.

This report covers work conducted from May 1964 to March 1965. The manuscript was released by the author in March 1965 for publication as an RTD Technical Report.

Publication of this report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

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ABSTRACT

The objective of this program was to investigate the feasibility of separating water vapor and/or droplet water from air by selective permeation through microporous materials. Two concepts were studied with the goal of removing 0.1 pound per hour of water or water vapor from approximately ten pounds of air per hour.

In the first concept, a 15 micron average pore size barrier of sintered Kel-F was positioned perpendicularly across a droplet laden air stream. Positive removal of droplet water was observed. However, droplet accumulation in and on the barrier surface contributed to a 12 fold increase in pressure drop across the barrier.

A second concept studied utilized a continuous sheet membrane of cellulose acetate which transmitted water vapor but blocked the passage of permanent gases. Two methods of condensing the water vapor extracted from the air were tested. In one method, a modified dry-vane type commercial compressor was used to produce a high suction, resulting in sufficient pressure differential to induce permeation of water vapor across the membrane. The vapor was then compressed and transported to a conical condenser for condensation at room temperature. In the second method, a vacuum was created in the conical condenser causing the permeated water vapor to diffuse into the condenser, where it was frozen. While positive evidence of vapor transfer and water condensation were observed, problems of complete edgewise sealing of the cellulose acetate membrane and the cooling and suction limitations of the compressor precluded operation of the ideal cycle and achievement of design water removal rates.

Further work to perfect water droplet separation from air through porous media should be preceeded by pressure drop determination of droplet laden air through unit areas of porous media. The contract droplet water barrier design was based upon pressure drop data for dry air flow through the porous barrier material. Further work to perfect the membrane permeation concepts for the separation of water vapor from air would have to include a thorough search for, or development of, a high suction, oil-free compressor for compression of water vapor and more effective means for edgewise sealing of sheet membrane materials.

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## SECTION I

## INTRODUCTION

The objectives of this program were to design, fabricate, test and deliver prototypes representative of two concepts (for application in zero-gravity environment) for the continuous separation of water from air by selective permeation through microporous and permeable materials. One concept was for the removal of water droplets from air; the second was for the removal of water vapor. Two methods for transporting and condensing vapor to water were tested.

In the first concept, water droplets were to be removed by constraining the flow of moisture laden air through a porous membrane. Figure 1 illustrates the water droplet removal concept. Droplet-laden air enters the collection cone. The membrane permits air to pass through, but not water droplets, which is retained in the cone and collected as a liquid. For this contract the membrane selected was porous Kel-F. Water was removed during tests, but at a higher membrane pressure drop than expected.

The concept to remove water vapor from the air stream was based on the theory that a membrane could be chosen which would be highly selective to the passage of water vapor; thus, a high concentration of water vapor could be obtained on one side of the membrane, while air and other gases remained on the other side of the membrane. Then the vapor could be collected and condensed to water. Cellulose acetate was the membrane chosen for this concept.

Two variations of this latter concept were tested. Figure 2 shows the schematic for the vapor permeation with subsequent vapor compression concept. A compressor is located between the membrane and the condenser cone. The compressor created a vacuum such that the absolute pressure at the compressor inlet was lower than the absolute partial pressure of the water vapor in the duct. Vapor then permeated through the membrane to the compressor, where it was compressed and pumped to the cone. Ideally, the water vapor in the cone would be condensed at room temperature.

Tests were run, but only after many fabrication difficulties were resolved. Even then, construction of the compressor prevented the collection of water at room temperature. By freezing the vapor, however, water was collected.

The alternate concept for recovering water vapor from an air stream, consisted of a continuous freezing diffusion process. The schematic is shown in Figure 3.

Theoretically, an initial vacuum could be drawn at the cone and the vacuum pump shut off. Because of the vapor pressure gradient between the duct and cone, water vapor permeates through the membrane to the cone. By rapidly freezing-out the

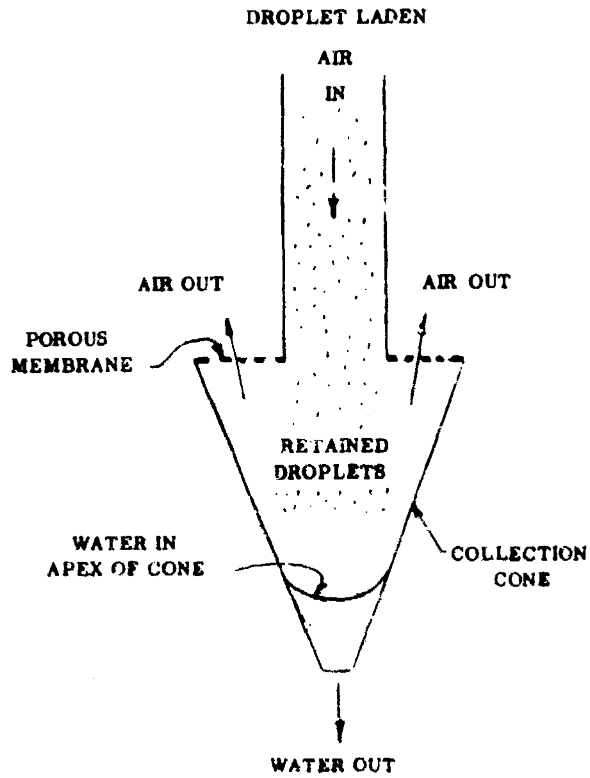



**Figure 1. Water Droplet Removal Schematic**

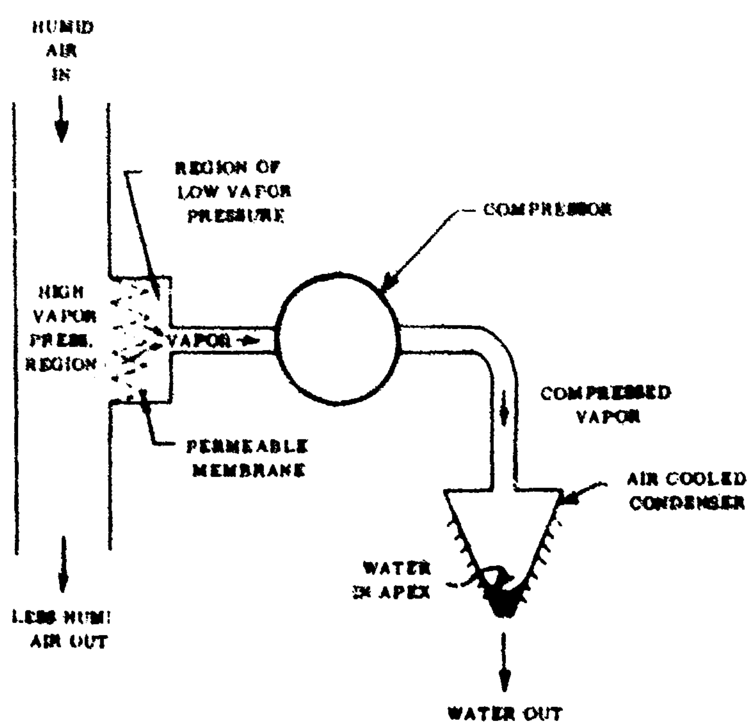



**Figure 2. Vapor Permeation with Subsequent Vapor--Compression Concept**

vapor, vacuum is maintained at the cone, and vapor flow continues through the membrane. The vacuum pump is used to remove the noncondensable gases. This concept was satisfactorily tested, but the removal of noncondensables required continuous operation of the vacuum pump.

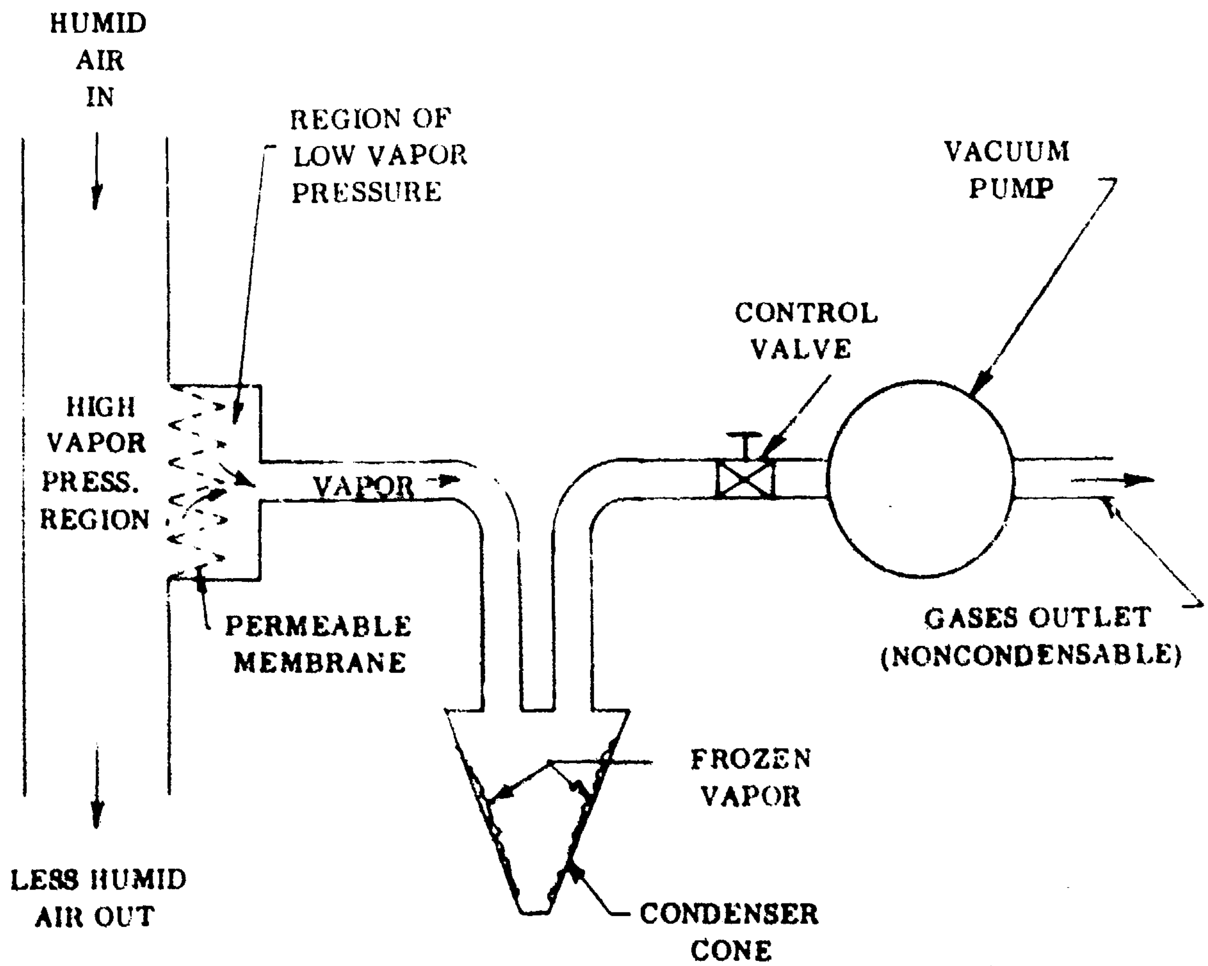



Figure 3. Vapor Permeation with Subsequent Vapor--Freeze-Out Concept

## SECTION II

## SUMMARY OF RESULTS

1. REMOVAL REQUIREMENT

Remove 2.5 lbs/day of water from an air stream of 250 lbs/day.

a. Water Droplet Removal Concept, Kel-F Membrane

Removal rate - 1.75 lbs/day, based upon one 5 3/4-hour test which showed 70% water removal.

b. Vapor Permeation Concepts

Cellulose acetate membrane 0.002 inch thick
Area of 7560 $cm^2$ (8.15 $ft^2$)
Water vapor compression and condensation concept removed 1.18 lbs/day
System efficiency of 51%, based on membrane permeability calculated from data obtained during one 35-minute test.
Water vapor permeation with direct freeze-out concept removed 0.82 lb/day
System efficiency 23%, based on membrane permeability calculated from data obtained during one 2 1/4-hour test.

NOTES: Actual vapor flow rates were @ 13 lbs/day in an air stream of 224 lbs/day. Several tests were run for the water droplet removal concept and both vapor permeation concepts.

2. DESIRED LIFE

30 days minimum

a. Water Droplet Removal Concept

Life of 30 days is feasible, dependent on amount of dirt or bacteria in air.

b. Water Vapor Permeation Concepts

There is a possibility of leaks developing in the membrane within 30 days. More effort is needed in this phase of design.

## 3. METHOD OF "SCALING UP" FOR GREATER FLOW

### a. Water Droplet Removal Concept

The items to scale up in this method are the size of membrane and collector cone. The Materials Section (Section VI) shows the velocity-pressure drop curves from which the membrane can be sized. The volume of the cone should be small enough to cause "wicking to the apex," and large enough to limit the frequency of drainage.

### b. Water Vapor Permeation Concept

The key equations and a typical calculation are given in Section IV. These may be used to scale up the acetate membrane assemblies and select a suitable compressor. The flow of vapor by membrane permeation is directly proportional to material permeability, material area and pressure gradient, and inversely proportional to the thickness. Appendix IV shows sample calculations for determining the pressure gradient for given temperature-humidity conditions and a given compressor characteristic. The Materials Section (Section VI) shows material properties.

## SECTION III

## WATER DROPLET REMOVAL CONCEPT

### 1. EQUIPMENT DESIGN

The approach to water separation by the use of porous membranes rests on two assumptions. First, there must be liquid water in droplet form mixed with air. Second, the membrane must not "wet", or at least "wet" very little. Samples of several non-wetting materials were obtained, and tested for flow-pressure drop characteristics. (See the Materials Section for tests and curves.) The material selected as having the best combination of low pressure drop, light weight, and structural rigidity was porous Kel-F.

Reference 1 shows that in zero gravity, a liquid will tend to flow to the apex of a cone if the sides are wettable; hence, the water collector was designed in a conical shape. A simple, inexpensive way of making a porous Kel-F barrier is to cut a disk from a flat sheet. Figure 4 shows the assembly of the membrane and cone. Airborne water droplets enter the assembly by the center hole through a connecting tube. The air escapes through the Kel-F, but the droplets do not. The droplets collect on the walls of the cone and in zero-g flow to the apex. To prevent water evaporation in the cone, a cooling tube was tack-welded to the cone, but was found to be unnecessary during tests.

### 2. TEST DESCRIPTION - WATER DROPLET REMOVAL CONCEPT

The contract requirement was to show that 2.5 pounds of water could be collected from 250 pounds of air per day. This reduces to an air flow of 2.35 cfm, and a water droplet rate of 0.00174 pound per minute (equal to 0.788 gram per minute).

Figure 5 is a drawing of the test setup. Figures 6 and 7 are photographs of the equipment. Two nozzles of the atomizing type were used to produce airborne droplets; that is, water was siphoned into the nozzles and atomized by gas pressure. Dry compressed nitrogen was utilized because it was readily available. By regulating the nitrogen pressure to 9.3 $\pm$ 0.2 psig, a gas flow of 2.35 cfm was obtained as indicated by a venturi differential pressure of one inch of water. Flexible tubes from the nozzles were then inserted into a jar of water. The water was siphoned by the nitrogen, atomized by the nozzles and injected into the mixing chamber. Some of the atomized water evaporated, but most remained as droplets. The large droplets fell into the mixing chamber and were later recovered and measured as excess water.

There was sufficient pressure at the nozzles to carry the small droplets and vapor into the straightener duct without need for an additional fan or use of the damper cont[illegible]. Additional condensation occurred in the duct, the water ran down the duct walls [illegible]nd was collected as excess water from the lower duct. The center stream flow of nitrogen, vapor, and entrained droplets entered the cone after passing the temperature sensor, humidity sensor, and the venturi meter. The nitrogen and water vapor freely passed through the membrane into the plastic shroud, then past the

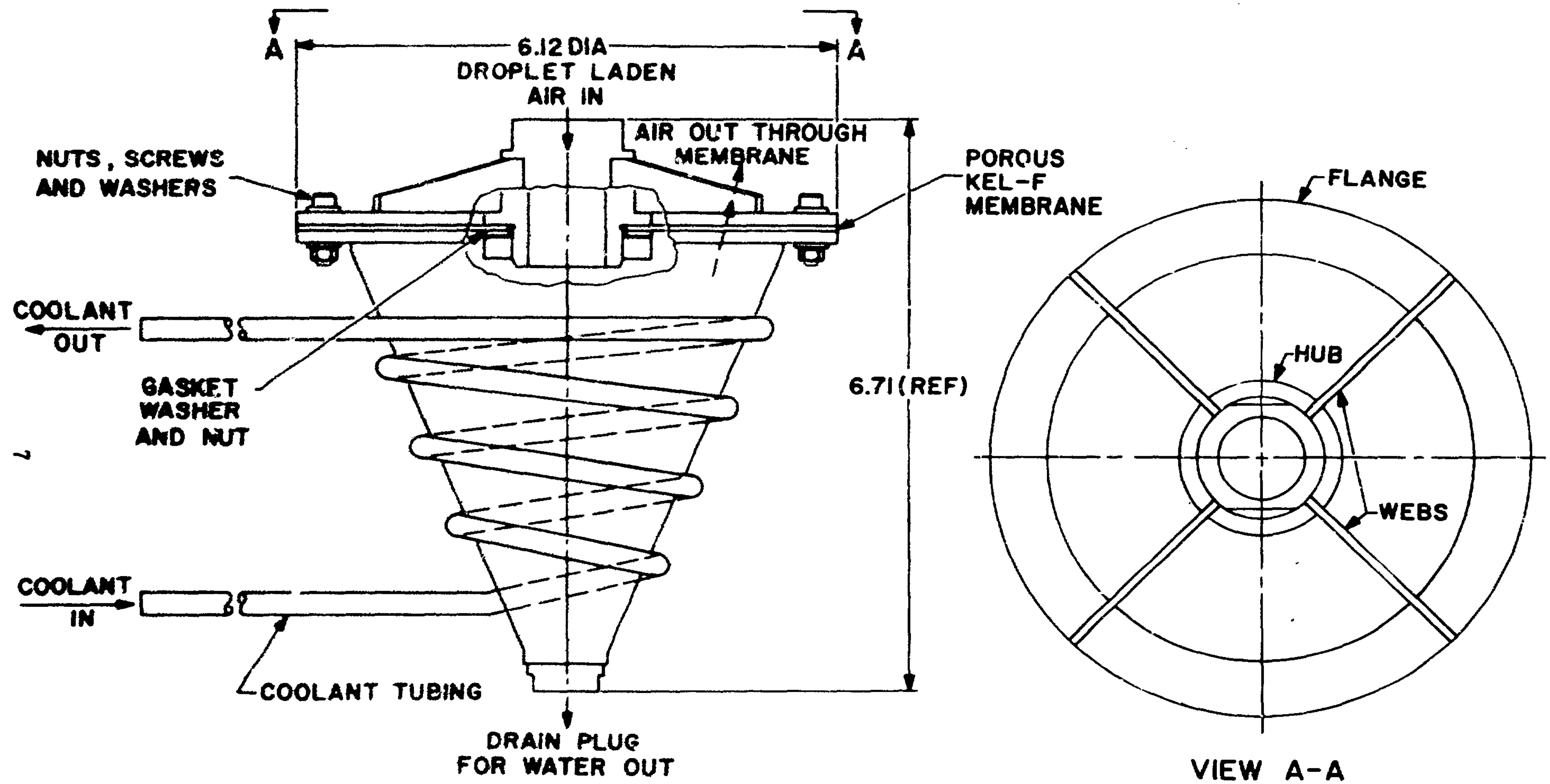



Figure 4. Collector for Water Droplet Removal

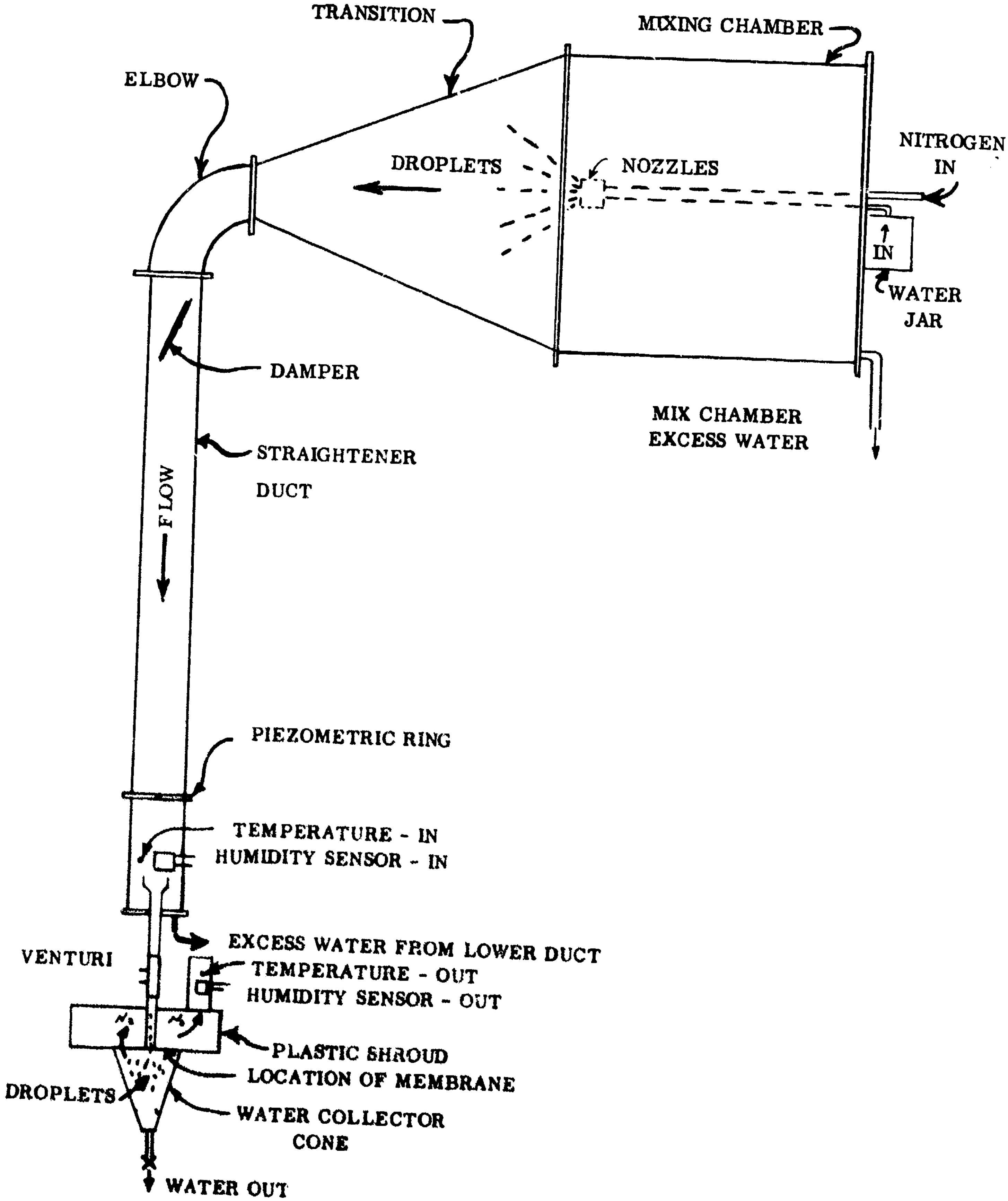



Figure 5. Test Setup for the Water Droplet Removal Concept

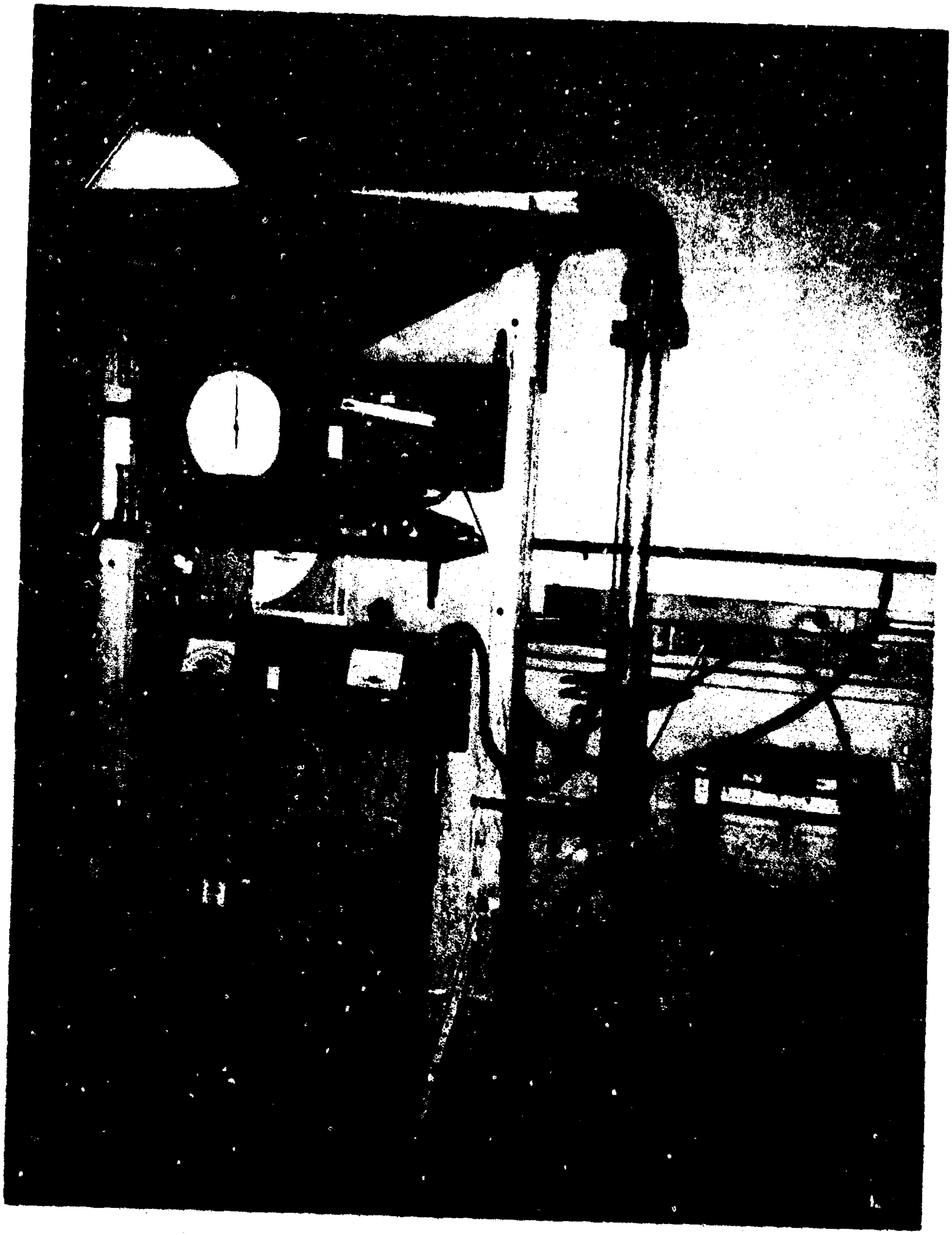


Figure 6. Water Droplet Test Equipment

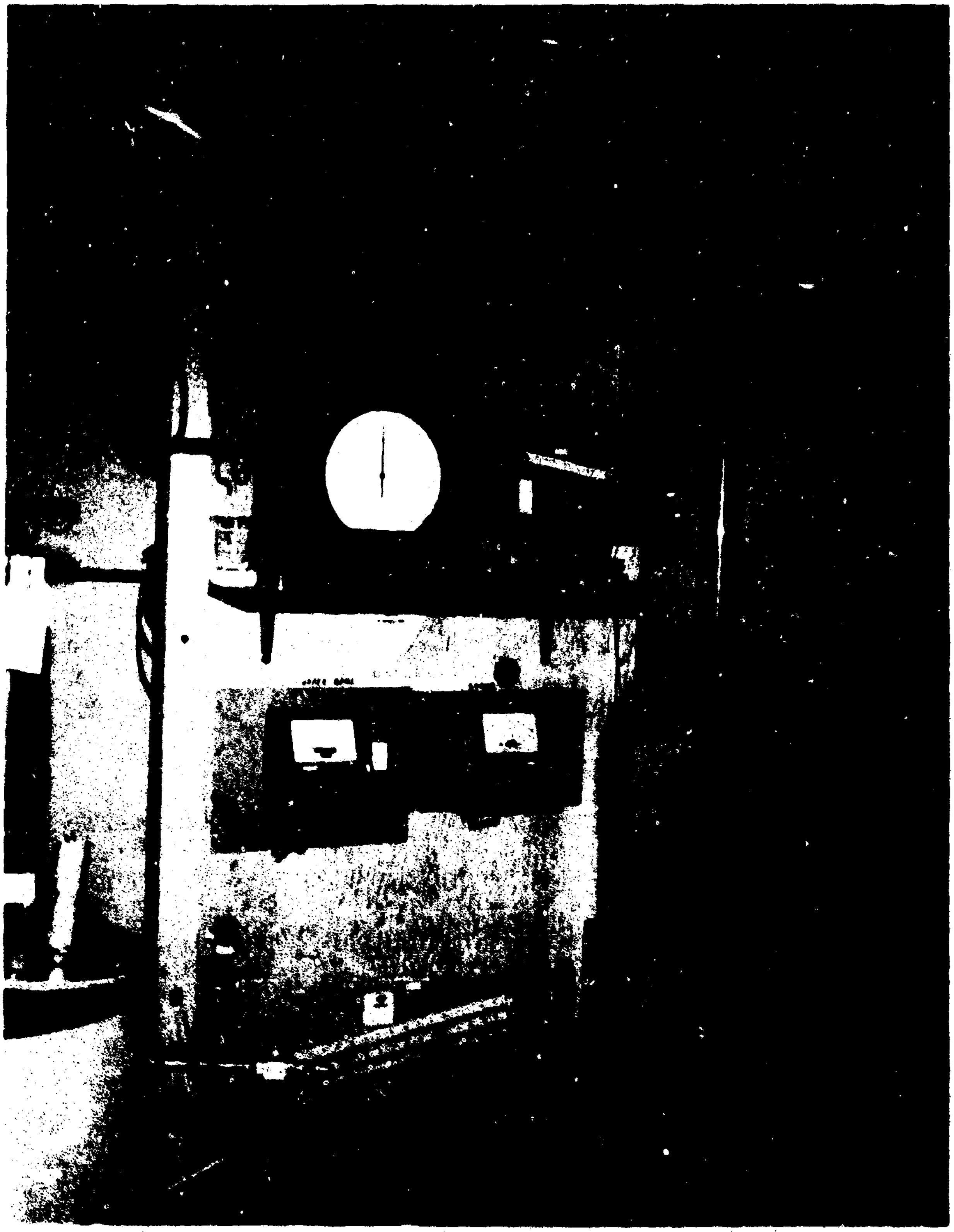


Figure 7. Water Droplet Test Equipment, Rear of Mix Chamber

outlet temperature and humidity sensors, and out into room ambient. The piezometric ring measured the total pressure drop from the cone inlet to the room outlet for the flowing gases. The droplets, however, were retained in the cone and were later removed, weighed, and recorded as "water collected in the cone."

### 3. DISCUSSION OF RESULTS - WATER DROPLET REMOVAL CONCEPT

Tables I, II, and III give the results for various test conditions. In every test, the jar of supply water was weighed at the beginning and the end of the test; the difference in weight being the total water injected. The excess water that fell or condensed in the ducts was drawn off and measured. Having set the flow rate and measured temperature and humidity, the flow of water that became vapor was computed for both the inlet and outlet humidity conditions. Subtracting the sum of "Excess Water In" and "Water-to-Vapor In" from the "Total Water Injected" gives the computed "Water-to-Entrained Droplets." Some of this water was collected in the cone and was measured.

With the exception of Test No. 8 (Table III), the calculated quantity of water vapor at the outlet was less than the calculated water vapor at the inlet. There was apparent condensation of vapor in the cone; hence, the water quantity represented by vapor differential was subtracted from the water collected by the cone. The net collected water was credited to the porous membrane. (In Test No. 8, the vapor difference was added to the membrane collection because of apparent evaporative loss from the cone.) The percent of water removal by the membrane was computed by dividing membrane collection rate by entrained droplet rate (see equation below).

$$\text{Percent water removal} = \frac{\text{Membrane Collection Rate}}{\text{Entrained Droplet Rate}} \times 100 \qquad \text{(III-1)}$$

For every test run, a water balance was made to see if the water inlet equalled the sum of water discharged plus water collected. A perfect balance was not achieved. One possible reason for the unbalance is that an unpredictable immeasurable amount of water would stick to the duct walls throughout an entire test. On subsequent runs, some of this water would be dislodged and collected resulting in a water gain (Tests 10 and 11, Table II). By combining the results of all tests of a given day, it is believed that the water loss or gain of the ductwork did average out; hence, results from Tests 2, 3 and 4 (Table I) were combined, as were Tests 10, 11 and 12 (Table II).

The ductwork seams were coated with silicone rubber during manufacture, and gaskets were used throughout; hence, water loss by duct leakage was unlikely. The membrane itself could have been passing very fine droplets, too small for the eye to see, since visible droplets were forced through the membrane during some of the test. Although it is not certain that all the loss was due to the membrane, the nature of the calculations and measurements charges all the unaccounted water loss to the membrane.

A comparison of Tests 3 + 4 + 5 and 10 + 11 +12 shows that much higher droplet entrainment rates occur when there is no membrane; hence, a comparison of efficiencies should be made, not just total water collected. The Kel-F appears to be

TABLE I. WATER DROPLET REMOVAL - KEL-F MEMBRANE

| TEST NO. | 2 | 3 | 4 | 5 | COMBINED RESULTS 3 + 4 + 5 |
|---|---|---|---|---|---|
| Test Time, min. | 44.0 | 45.0 | 45.0 | 45.0 | 135.0 |
| Temperature Mix In, °F | 71.5 | 71.0 | 71.0 | 71.5 | 71.2 |
| Relative Humidity In, % | 84.0 | 84.5 | 84.0 | 84.0 | 84.2 |
| Temperature Mix Out, °F | 71.5 | 71.5 | 71.0 | 71.0 | 71.2 |
| Relative Humidity Out, % | 81.0 | 81.0 | 80.5 | 81.0 | 80.7 |
| Nitrogen-Vapor Flow, cfm | 2.4 | 2.4 | 2.4 | 2.4 | 2.4 |
| Pressure Drop, in $H_2O$ | 10.6 | 11.5 | 11.5 | 11.9 | 11.6 |
| Total Water Injected, g | 795.0 | 815.0 | 849.0 | 852.0 | 2516.0 |
| Excess Water in Mix Chamber, g | 702.5 | 698.5 | 749.4 | 752.5 | 2200.4 |
| Excess Water in Lower Duct, g | 23.0 | 22.0 | 18.0 | 21.0 | 61.0 |
| Water to Vapor In, g* | 48.7 | 43.7 | 49.1 | 49.7 | 142.5 |
| Water to Entrained Droplets, g* | 20.8 | 50.8 | 32.5 | 28.8 | 112.1 |
| Water Collected in Cone, g | 20.0 | 16.0 | 20.0 | 25.0 | 61.0 |
| Water to Vapor Out, g* | 47.0 | 42.6 | 47.1 | 47.5 | 137.2 |
| Vapor In-Vapor Out, g* | 1.7 | 1.1 | 2.0 | 2.2 | 5.3 |
| Membrane Collection, g* | 18.3 | 14.9 | 18.0 | 22.8 | 55.7 |
| Membrane Collection Rate g/min.* | 0.415 | 0.331 | 0.400 | 0.506 | 0.412 |
| Droplet Rate, g/min* | 0.472 | 1.13 | 0.723 | 0.640 | 0.830 |
| Percent Water Removal * | 88.0 | 29.3 | 55.4 | 79.1 | 49.5 |
| Water Balance Loss (-), gain (+), g* | -0.8 | -34.8 | -12.5 | -3.8 | -51.1 |

* Computed quantities. All other items are measured.

TABLE II. WATER DROPLET REMOVAL - NO MEMBRANE

| TEST NO. | 10 | 11 | 12 | COMBINED RESULTS 10 + 11 + 12 |
|---|---|---|---|---|
| Test time, min. | 30.0 | 30.0 | 60.0 | 120.0 |
| Temperature Mix In, $^oF$ | 71.0 | 70.0 | 70.0 | 70.3 |
| Relative Humidity In, % | 84.3 | 83.3 | 83.3 | 83.6 |
| Temperature Mix Out, $^oF$ | 71.0 | 69.5 | 69.5 | 70.0 |
| Relative Humidity Out, % | 83.8 | 83.9 | 84.9 | 84.2 |
| Nitrogen-Vapor Flow, cfm | 2.4 | 2.4 | 2.4 | 2.4 |
| Pressure Drop, in $H_20$ | 0.3 | 0.3 | 0.3 | 0.3 |
| Total Water Injected, g | 883.0 | 899.0 | 1827.0 | 3609.0 |
| Excess Water in Mix Chamber, g | 815.5 | 822.5 | 1552.0 | 3190.0 |
| Excess Water in Lower Duct, g | 33.0 | 28.0 | 59.0 | 120.0 |
| Water to Vapor, in g* | 32.5 | 31.6 | 63.0 | 127.1 |
| Water to Entrained Droplets, g* | 2.0 | 16.9 | 153.0 | 171.9 |
| Water Collected in Cone, g | 16.0 | 21.0 | 45.0 | 82.0 |
| Water to Vapor out, g* | 32.3 | 30.5 | 62.0 | 124.8 |
| Vapor In-Vapor Out, g* | 0.2 | 1.6 | 1.0 | 2.3 |
| Net Collection, g* | 15.8 | 19.9 | 44 | 79.7 |
| Net Collection Rate g/min. * | 0.526 | 0.652 | 0.733 | 0.665 |
| Droplet Rate g/min. * | 0.067 | 0.563 | 2.55 | 1.43 |
| Percent Water Removal * | 788.0** | 116.0** | 28.7 | 46.5 |
| Water Balance Loss (-), gain (+), g * | +14.0 | +4.1 | -109.0 | -89.9 |

* Computed quantities. All other items were measured.

**Cone was gaining water from system.

TABLE III. WATER DROPLET REMOVAL - KEL-F MEMBRANES

| TEST NUMBER | 7 | 8 | 14 | 15 |
|---|---|---|---|---|
| Test Time, min. | 190.0 | 180.0 | 45.0 | 345.0 |
| Temperature Mix in, g | 70.0 | 68.0 | 73.0 | 72.5 |
| Relative Humidity in, % | 84.3 | 84.6 | 83.2 | 81.8 |
| Temperature Mix Out, g | 69.5 | 69.5 | 69 | 68 |
| Relative Humidity out, % | 80.4 | 81.4 | 80.4 | 81.6 |
| Nitrogen-Vapor Flow, cfm | 2.4 | 2.4 | 2.4 | 2.4 |
| Pressure Drop in $H_2O$ | 11.9** | 11.9 | 11.9 | 11.8 |
| Total Water Injected, g | 3694.0 | 3855.5 | 817.0 | 6668.0 |
| Excess Water in Mix Chamber, g | 3175.0 | 3455.5 | 717.0 | 5972.0 |
| Excess Water in Lower Duct, g | 68.0 | 94.0 | 23.0 | 186.0 |
| Water to Vapor in, g* | 201.0 | 179.0 | 51.6 | 382.0 |
| Water to Entrained Droplets, g* | 250.0 | 127.0 | 25.4 | 148.0 |
| Water Collected Cone, g | 87.0 | 92.5 | 19.0 | 155.5 |
| Water to Vapor Out, g* | 188.0 | 180.0 | 44.0 | 330.5 |
| Vapor In-Vapor Out, g* | 13.0 | -1.0 | 7.6 | 52.5 |
| Membrane Collection, g* | 74.0 | 93.5 | 11.4 | 104.0 |
| Membrane Collection Rate, g/min.* | 0.39 | 0.520 | 0.253 | 0.301 |
| Droplet Rate, g/min.* | 1.32 | 0.705 | 0.565 | 0.430 |
| Percent Water Removal * | 29.5 | 73.5 | 44.8 | 70.0 |
| Water Balance Loss (-) Gain (+) * | -176.0 | -33.5 | -14.0 | -44.0 |

* Computed quantities. All other items were measured.

** Changing pressure during first 20 minutes of test.

only slightly more efficient than "no membrane," since the effects of gravity could not be eliminated in these tests. But with the membrane removed, a cloud of droplets was observed leaving the exhaust in a direction opposing gravitational forces; however, with the Kel-F in place, no such cloud was visible. This is visual proof that the membrane will remove droplets.

Test Number 6 (not included in the Tables) was run to determine the piezometric pressure drop from the cone inlet to the room outlet. At 2.35 cfm of dry nitrogen, the pressure drop was 1.2 inches of water with the Kel-F membrane in place. Without the membrane, the pressure drop was 0.3 inch of water; thus by difference, the pressure drop across the dry membrane is 0.9 inch of water. As soon as water was added, the pressure drop in the system increased until a peak of nearly 12 inches of water was obtained, indicating that fine droplets were clogging the membrane pores. At this point, some droplets were forced through the membrane. Discussion with the Kel-F manufacturer indicated that the "bubble point" had been reached. (By applying air to a porous filter immersed in alcohol the pressure at which a "bubble" first appears determines the micron size of the filter. This is the "bubble point" test. If a filter is immersed in water instead of alcohol, the bubble point pressure would double, approximately.) To reduce the loss of water, a material of higher bubble point could be used; hence, a test with sintered teflon was attempted. Having sized the cone for Kel-F, the back pressure created by installing the same area of available sintered teflon was so great that water could not be injected by the nozzles into the system. (Nevertheless, a test was run with dry nitrogen to obtain teflon properties. This is reported in the Materials Section, Figure 26.)

After the pressure drop test, a run of over three hours was made on a Kel-F membrane. This was test Number 7. Examination of the data showed that the efficiency was very low, but that the piezometric pressure drop steadily increased for the first 20 minutes; hence, test Number 8 was run, essentially repeating Test 7 with one major exception. Before beginning the test, nitrogen and water were injected into the system for one hour. This saturated the ductwork with droplets and condensate, and brought the piezometric pressure drop to a stable value of 11.9 inches of water. The excess water from the mixing chamber, duct and cone were removed and discarded. This was termed a "saturation run."

After the saturation run, Test 8 was run. Comparing the results of 7 and 8 showed that Test 8 had virtually stable vapor flow, a better total water balance, a lower droplet rate than Test 7, but a higher collection rate; hence, the 73.5% water removal of Test 8 was 2-1/2 times the water removal of Test 7. It is concluded, therefore, that the saturation run tends to stabilize the system resulting in better water balance and more valid test data than tests without a preceding saturation run. In addition, once the membrane has reached a stable pressure drop, the highest efficiency will be obtained. The pressure drop was recorded at 11.9 inches of water, but subtracting the "no membrane" drop of 0.3 inch, gives the net Kel-F pressure drop of 11.6 inches of water.

The last two tests, Nos. 14 and 15, were run on a new Kel-F membrane on successive days. A saturation run preceded each test. Again, the higher efficiency was obtained for the longer test.

## 4. CONCLUSION-WATER DROPLET REMOVAL CONCEPT

The Kel-F membrane will remove water droplets from the airstream at a rate of approximately 70% after a saturation run; however, the net pressure drop of 0.9 inch of water for the membrane, when dry, will rise to 11.6 inches of water when the membrane is saturated. At 70% recovery, 1.75 pounds of water per day is removed from the required flow of 250 pounds of air containing 2.5 pounds of water droplets. Water vapor passes freely through the membrane, and the indications are that sub-visual droplets (ten microns or less) also pass freely.

To increase collection performance, two approaches are suggested. The first is to use a thicker membrane. This would result in more pores with a greater droplet holding capacity than a thin membrane resulting in a slower rise in pressure drop. With a thicker membrane, there would also be a more tortuous path for the fine droplets, resulting in a greater retention of these droplets. The penalty for increasing the Kel-F thickness from 1/16 in. to 1/8 in. would be an increased pressure drop from 0.9 to 1.8 inches of water when dry.

The second suggestion is an increased area. A test with a larger area membrane would ascertain whether a Kel-F membrane would always have a buildup to the "bubble point" when screening out droplets, or would reach a lower equilibrium pressure. If the pressure drop across the membrane is less than the "bubble point" pressure of 11.6 inches of water, droplets would not be forced through the membrane.

## SECTION IV

## WATER VAPOR PERMEATION WITH SUBSEQUENT VAPOR COMPRESSION

### 1. THEORY AND DESIGN

Various solid materials have the property that permits passage of gases, vapors, and liquids. The transmission of a gas or vapor through a film is normally of the active diffusion type; that is, the gas or vapor dissolves into the surface of the film where the gas concentration is high, diffuses through the film, and evaporates at the low-gas concentration side. Under steady-state conditions, the rate of transmission follows the relationship

$$q = -D \frac{dc}{dx} \qquad \text{(IV-1)}$$

where $q$ is the rate of diffusion per unit area
$D$ is the diffusion constant
$dc/dx$ is the concentration gradient for a thickness $x$.

If D is independent of the concentration, c, equation IV-1, can be integrated over the thickness "t" to give

$$q = \frac{D}{t} (c_1 - c_2) \qquad \text{(IV-2)}$$

where $c_1$ and $c_2$ are the gas concentrations at the two surfaces.

But c can be expressed by the equation

$$c = pS \qquad \text{(IV-3)}$$

where $S$ is the solubility coefficient of the gas in the film and
$p$ is the pressure of the gas in equilibrium with the film.

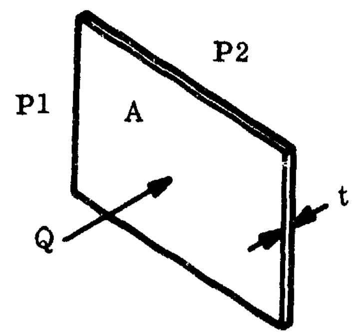
Then $$q = \frac{DS}{t} (p_1 - p_2) \qquad \text{(IV-4)}$$

Let P be defined as the permeability constant such that $P = D\,S$

Then $$q = \frac{P}{t} (p_1 - p_2) \qquad \text{(IV-5)}$$

or $$Q = \frac{PA}{t} (\Delta p) \qquad \text{(IV-6)}$$

where $Q$ is the total flow through the surface
$A$ is the total surface area
$\Delta p$ is the pressure gradient across the barrier
$t$ is barrier thickness



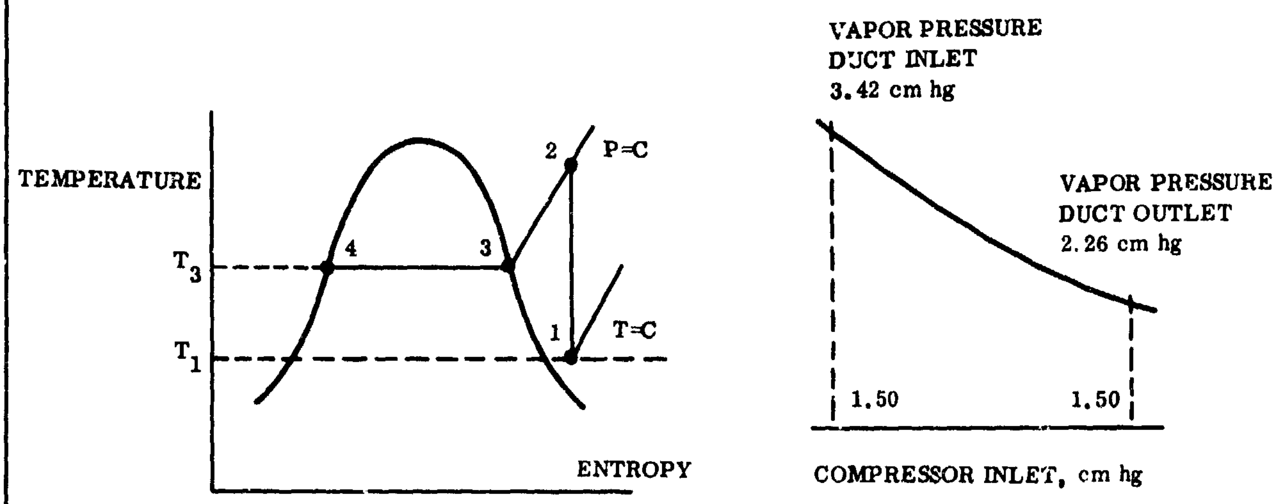

The object of the Water Vapor Permeation Concept is to apply the principles depicted by equation (IV-6) to continuously remove water vapor from a stream of air, and then collect and condense the vapor with equipment suitable for use in a zero gravity environment. Two collection methods were tested. The first method of recovery wherein a compressor was used as an active part of the system is described in this section. The second method is described in Section V under the heading Water Vapor Permeation with Subsequent Vapor Freeze Out.

The process of using a compressor and membrane to recover water vapor is as follows. A membrane is located so that one side is open to a duct of humid air with the other side of the membrane leading to the compressor inlet. The compressor, when operating, creates a vacuum behind the membrane causing water vapor to permeate through the membrane into the low-pressure region. The vapor is then compressed, until the saturation pressure of water is achieved, and discharged into a condenser. If the saturation pressure is increased, the corresponding saturation temperature will be increased to a value above room temperature; thus, it will be possible to condense the vapor to water at room temperature eliminating the need for a low-temperature coolant supply.

The cycle is shown in Figure 8 with calculations to determine the requirements to remove 2.5 lbs/day of water from 250 lbs/day of air. Assuming an initial humid air condition of $90^{o}$F, 95% R.H., and removing 70 grains per pound of air, the vapor pressure on one side of the membrane would vary from 3.42 cm Hg to 2.26 cm Hg. To create an effective vapor flow, the compressor inlet must be below 2.26 cm Hg., and 1.5 cm Hg was assumed. The resulting logarithmic mean pressure difference is 1.30 cm Hg.

To obtain a small area, and thus a small volume for space vehicle application, a thin membrane of high permeability is desired. The material chosen was 0.001-inch cellulose acetate. This film has a very high permeability to water vapor with 1500 x $10^{-9}$ (Std cc) (cm thick)/(sec) $(cm)^2$(cm Hg) as a typical conservative average. The required area for two units in parallel is 10,600 $cm^2$ each.

The temperature-entropy chart (Figure 8) shows the compression and cooling cycle. For an inlet vapor pressure of 1.50 cm Hg (point 1), the saturation temperature is $63^{o}$F. If the vapor is compressed (5:1 compression ratio) to 7.50 cm Hg (point 2), the saturation temperature is $115^{o}$F. Thus, there is the possibility of using room temperature air (nominally $77^{o}$F) to condense the water vapor along constant pressure line.




Related Calculations

Pressure at inlet to duct

For $90^{\circ}F$ - 95 R.H. 207 gr/lb
Water Removed 70 gr/lb
137 gr/lb

p = 3.42 cm Hg

Note: 70 grains/lb x 250 $\frac{\text{lb}}{\text{day}}$ of air x $\frac{1}{7000}$ $\frac{\text{lb}}{\text{gr}}$ = 2.5 lb/day

At 137 gr/lb and $90^{\circ}F$, the relative humidity is 63 percent and the vapor pressure is 2.26 cm Hg at the duct exit.

For an estimated compressor inlet of 1.50 cm Hg (Point 1)

$$\text{Mean log } \Delta P = \frac{(3.42 - 1.5) - (2.26 - 1.5)}{\ln \frac{(3.42 - 1.5)}{(2.26 - 1.5)}} = 1.30 \text{ cm Hg}$$

Desired water recovery of 2.5 lb/day = .0131 g/sec.

At STP, specific volume of water vapor is 1245 $\frac{\text{cc}}{\text{g}}$.

Therefore flow rate v = (.0131) (1245) = 16.3 Std $\frac{\text{cc}}{\text{sec}}$.

For a membrane of 1 mil thick and P = $1500 \times 10^{-9}$ $\frac{(\text{Std cc})\ (\text{cm thick})}{(\text{sec})\ (\text{cm})^2\ (\text{cm Hg})}$

Substituting in equation (4-6) $A = \frac{Qt}{P\ (\Delta P)}$

$$A = \frac{(16.3)\ (.00254)}{(1500 \times 10^{-9})\ (1.3)} = 21,200 \text{ cm}^2 = 22.8 \text{ ft.}^2$$

For two units in parallel, A = 10,600 $\text{cm}^2$ each

Figure 8. Compressor and Membrane Cycle with Related Calculations

To condense vapor at room temperature, the surface area of a cone shaped condenser was greatly increased on the air side by the use of cooling pins. The cone-condenser design is described in Section V. To get a large area of membrane in a small volume, a folded membrane technique was used as shown in Figures 9 through 12.

Originally a membrane assembly was made with 36 active sides (37 corrugations) with the ends "sealed" in silicone adhesive. Each corrugation was 10.5 x 5 inches. Therefore $A = (36)\ (10.5)\ (5) = 1890\ in.^2$ or $12,200\ cm^2$ compared to the $10,600\ cm^2$ calculated requirement (Figure 8). When tested, the membrane leaked, especially at the ends. Therefore, the ends were repotted in epoxy adhesive, with reduction in width and active area. Also, to protect the membrane from tearing, and to prevent direct contact between the membrane and the corrugations (with consequent reduction in effective area), a qualitative grade of filter paper was added to cover each corrugation in subsequent assemblies. This reduced the number of corrugations that could fit into the housing from 37 to 33, hence, the two assemblies tested had active areas of $8000\ cm^2$ and $7560\ cm^2$, respectively. Figure 11 shows an assembly with filter paper, membrane, and 33 corrugations "sealed" in silicon adhesive. To stop leakage, it was necessary to also "pot" the ends of this assembly with epoxy.

The compressor was chosen after sixteen companies were contacted (see Appendix I). The problem was to find a "dry" compressor that would handle the expected flow of 16.3 Std cc/sec at the desired inlet pressure of 1.5 cm of mercury absolute. To get good vacuum, most manufacturers use an oil sealed pump. Oil would become contaminated with water vapor, and would cause loss of water; hence, the compressor selected was a Leiman Brothers (East Rutherford, N. J.) two-stage carbon-vane pump. When on test, the best pressure it provided was about 2.5 cm mercury absolute. To get a good pressure differential across the membrane, the temperature of the moist air inlet was increased from 90°F (from Figure 8) to 125°F.

## 2. TEST DESCRIPTION

### a. Preliminary Tests

The schematic diagram for the water vapor permeation-compressor concept is shown in Figure 13. Moisture is added to air in a laboratory humidity chamber. The fan draws the humid air out of the chamber into the vertical duct and through the venturi. From the transition, the humid air enters the "steam duct," passes the sensors and membranes, and exits beyond point B. The compressor produces a vacuum behind the membrane assembly as measured by the manometer. Vapor permeates from the duct through the membrane to the compressor. (Notice that the valves can be used to seal off either membrane.) The compressor discharges to the condenser. Ideally, the vacuum pump would remove the air from the cone, creating an initial vacuum, and then would be shut off. As the vapor subsequently enters the cone, it would be condensed, maintaining the vacuum. Because of the high selectivity of the cellulose acetate membrane, the amount of air that would permeate the membrane and enter the cone should be about 1 cc/minute. This air flow is so low, that the vacuum pump should be needed only periodically to purge the cone. Figure 14 is a photograph of the steam duct, membrane assemblies and compressor.

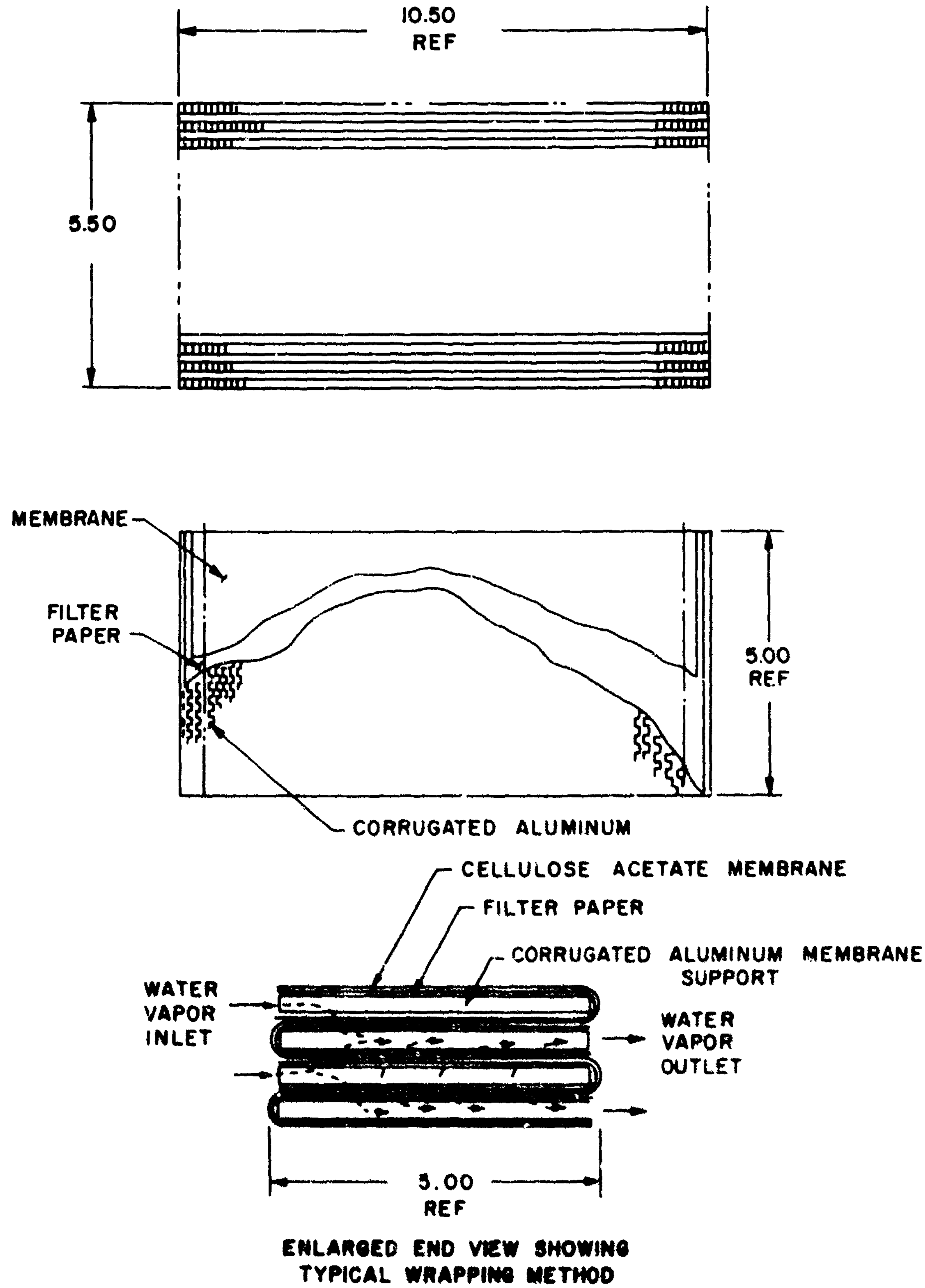



Figure 9. Membrane Assembly

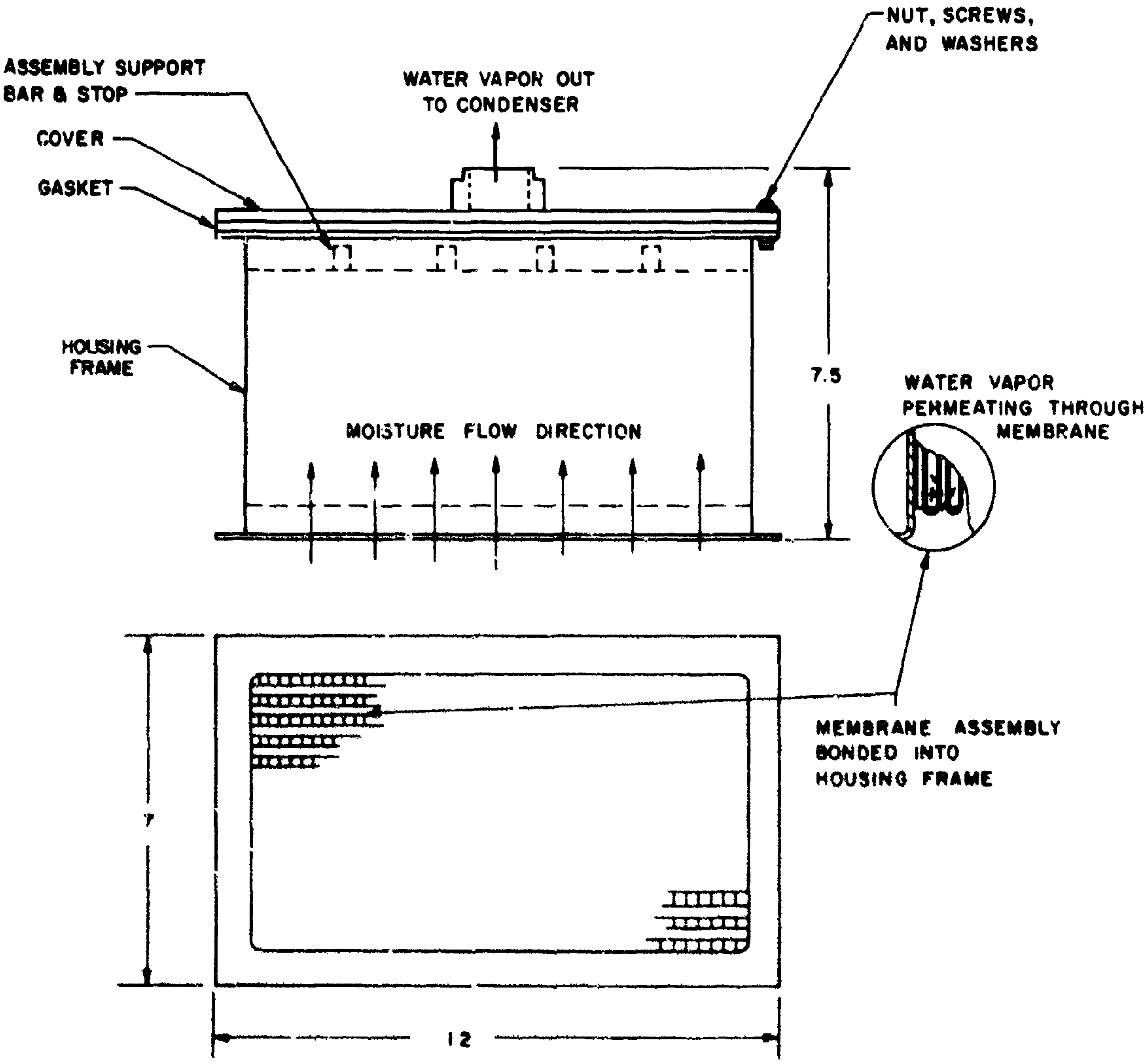



Figure 10. Membrane Assembly in Support Housing

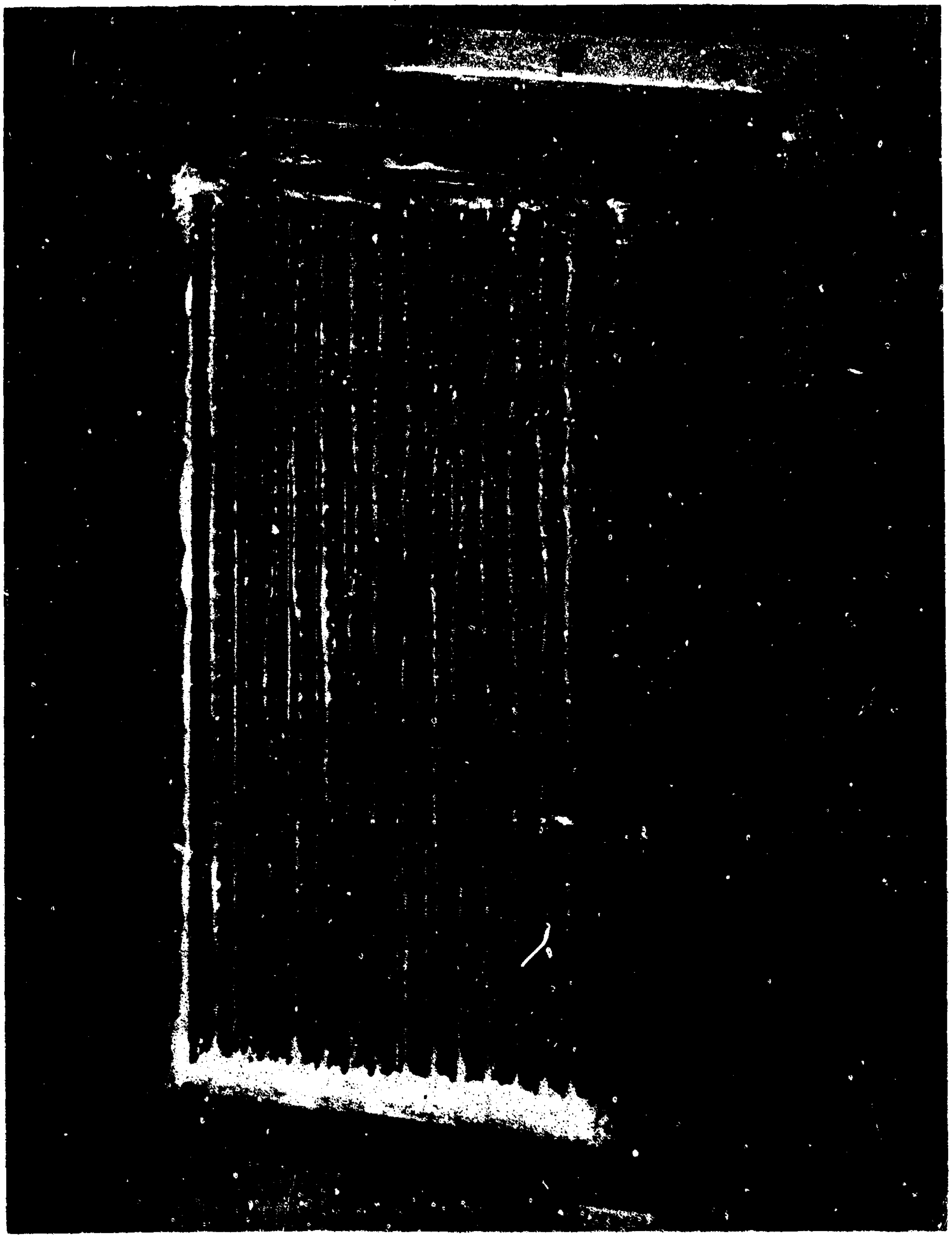
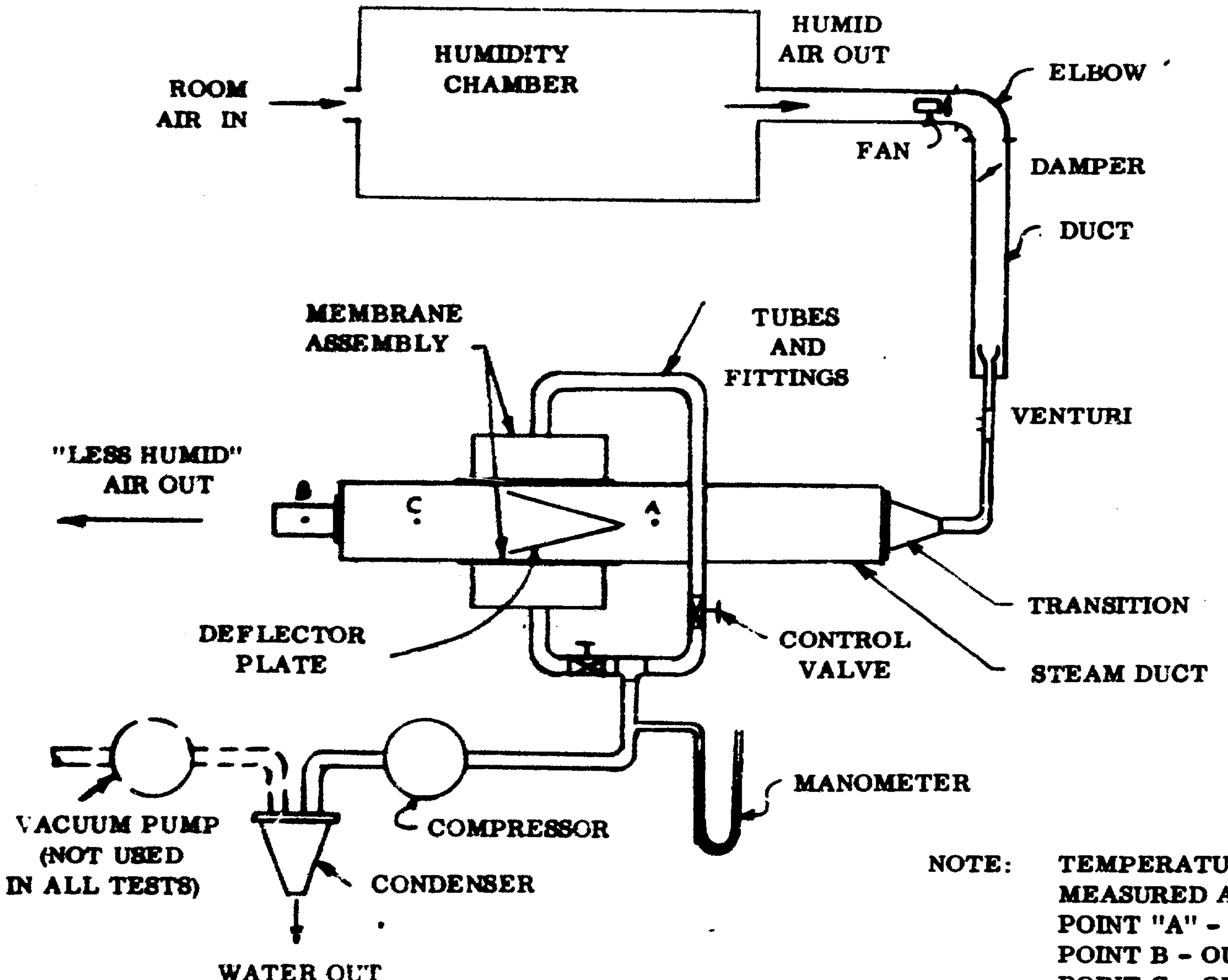


**Figure 11. Membrane Assembly - Front View**

**Figure 12. Membrane Assembly - Rear View**




NOTE: TEMPERATURE & HUMIDITY MEASURED AT:
POINT "A" - INLET
POINT B - OUTLET
POINT C - OUTLET FOR TEST OF TWO MIL MEMBRANE ONLY (TABLES X AND XIV)

Figure 13. Test Schematic for Vapor Permeation - Compressor Concept

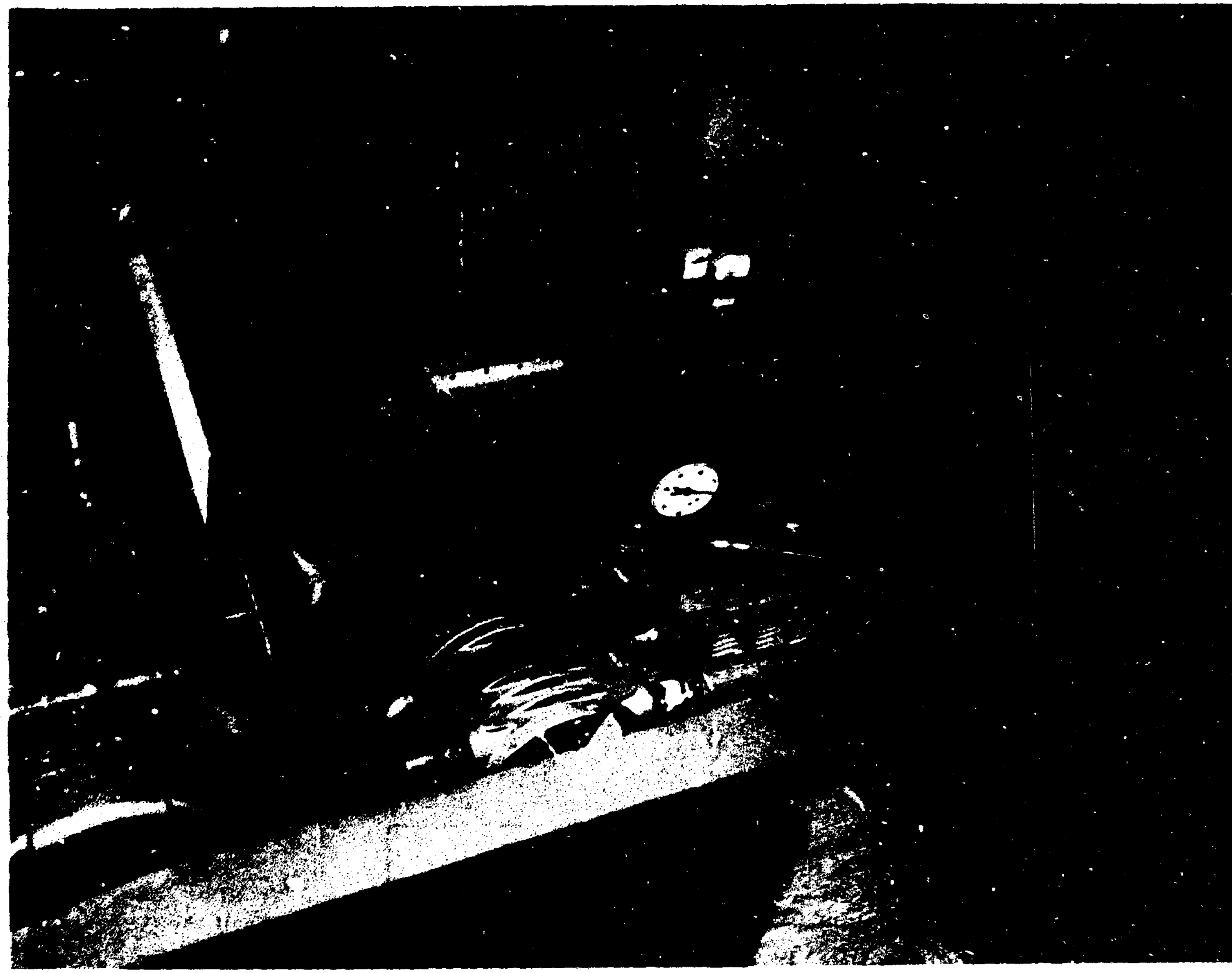


Figure 14. Steam Duct, Membrane Assembly and Compressor

In the preliminary tests, the vacuum pump could not purge the cone. The Leiman compressor is constructed with small holes near the bearings that allow room air to enter the compressor to keep it cool. These are called "weep holes." When the vacuum pump was turned on, it pulled room air through the compressor weep holes, which are in series with the compressor and cone. The vacuum pump, therefore, was removed and a plug was inserted to seal the cone; hence, in test, the compressor discharged into a closed cone. This apparently caused the compressor to overheat and the vanes temporarily siezed, causing the motor to stall, and the circuit breaker to cut off power. Several attempts were made to cool the pump externally, but to no avail. When the compressor pulled a vacuum at the inlet and pressurized the closed cone at the outlet, there was no way for air to enter the compressor through the weep holes. It was, therefore, concluded that the pump must not be operated in such a condition.

A test was made with the compressor discharging to a cone with the outlet open, but again no water was collected. The cone temperature was much too high to condense any water. Cooling with ambient air was not practical with the quantities of hot weep-hole air being discharged; hence, the cone-collector with integral cooling coils used for the water droplet tests was put into this test. (The cones were designed as interchangeable units.) Coolant was circulated through the tube holding the temperature to 76°F during a one-hour test with the drain open. A trace of water was collected but the pressure at the compressor side of the membrane was so poor (5.64 in. Hg absolute) that a leak was suspected. The membrane assembly for these first runs was one-mil cellulose acetate with no filter paper backing. This was rebuilt with a one-mil film with filter paper backing for later tests.

b. Compressor Weep-Hole Air Flow

The amount of air coming through the compressor weep holes was of great concern. From preliminary tests, the weep holes could not be blocked or rendered inactive because compressor overheating would occur. To evaluate the amount of weep-hole air, a Tissot Spirometer (also called a gasometer) was set up as shown in Figures 15, 16 and 17. The spirometer consists of a collection bell floating in water. By use of the three-way valve, air can be blocked, diverted away from the gasometer, or directed into the bell. If air enters the bell, the bell rises and the meter stick (attached to a counter weight) lowers, moving the chain from one side of the pulley to the other. The links of the chain are of such a weight that the movement continuously keeps the bell and the counter-weight balanced for every position. The volume of air entering the bell is directly proportional to the distance the meter stick moves as determined by the difference in scale readings. The conversion factor is 1 cm of linear difference = 1.332 liters displaced.

To determine the compressor weep-hole air flow the valve to the membrane was shut off and then the compressor was started; therefore, all the outlet air from the compressor was coming from the weep holes. The three-way valve was used to control the start and finish of the test. Table IV lists the results for a series of 1.5-minute tests. Volumes are adjusted to STP and also to the compressor temperature. Tables

V and VI show the results of variation in the tests. If the membrane had a large leak, then it should show up with the membrane open and dry as recorded in Table V. In Table VI, the humidity chamber was "on" in an attempt to measure the moisture passing through the membrane and compressor. Examination of the tables shows that most of the volumetric variation occurred because of temperature—the higher flows occurring at the colder pump temperatures. Figure 18 shows a plot of volumes reduced to STP versus compressor temperature, greatly expanded to clarify the points above 200°F. In the region of overlap, 200 to 208°F, the valve-open curve indicates a flow approximately 0.95 liter/minute greater than the valve-closed curve. This indicates an air leak in the membrane. With the humidity chamber on, the flow rate drops an average of 1 liter/minute below the valve-open curve, and condensation was observed in plastic connecting tubes. This must mean that water vapor was displacing some of the air coming through the membrane because in the calculation for Table VI, the flow rate of the bell temperature was reduced to STP dry air. From the test data, the average pressure gradient across the membrane was 1.82 cm Hg. The permeability was computed as $2900 \times 10^{-9}$ (cc/sec) (cm thick)/(cm$^2$) (cm Hg) for the 1 liter/minute flow (16.7 cc/sec).

Another observation of the tables should be made. Weep hole air flow volumes were 30 liters/minute and more at the compressor operating temperature range from 210°F to 230°F. The desired vapor flow to obtain the required 2.5 lb/day of water is 0.978 liter/minute; thus, approximately 97% of the gas leaving the compressor is air. The purpose of using a membrane before the compressor was to selectively pass water vapor, and ideally no air would be compressed at all. The weep holes, therefore, not only prevented the testing of the ideal cycle, but also prevented testing a cycle approaching the ideal concentration of water vapor.

c. Membrane and Cone Leakage

Another way that air could have entered the system other than through weep holes was through leaks in the membrane, or various gaskets. One test merely pumped out the cone with valves to the membrane shut, and then the cone was sealed off. After 45 minutes, there was no change in vacuum indicating that the cone was leak free. In one of the tests described in the vapor permeation freeze-out concept (see paragraph V.2.c), the cone was connected directly to the membrane. The system was pumped down to 1.8 inches of mercury (4.56 cm) absolute, and the valve to the pump closed. In 2-1/2 hours, only a trace of water was collected, but a record of cone pressure was made. The results are shown in Table VII. Taking an average cone temperature of -10°F, the absolute cone pressure reading was increased to the standard temperature of 32°F. The cone volume was computed to be 632 cc. Then taking the ratio of measured pressure divided by standard pressure (760 torr) and multiplying by 632 cc, the gas volume was computed. From this column, the leak rate was computed. The average leak rate for the one-mil, 8000-cm$^2$ membrane was computed as 3.06 cc/min.

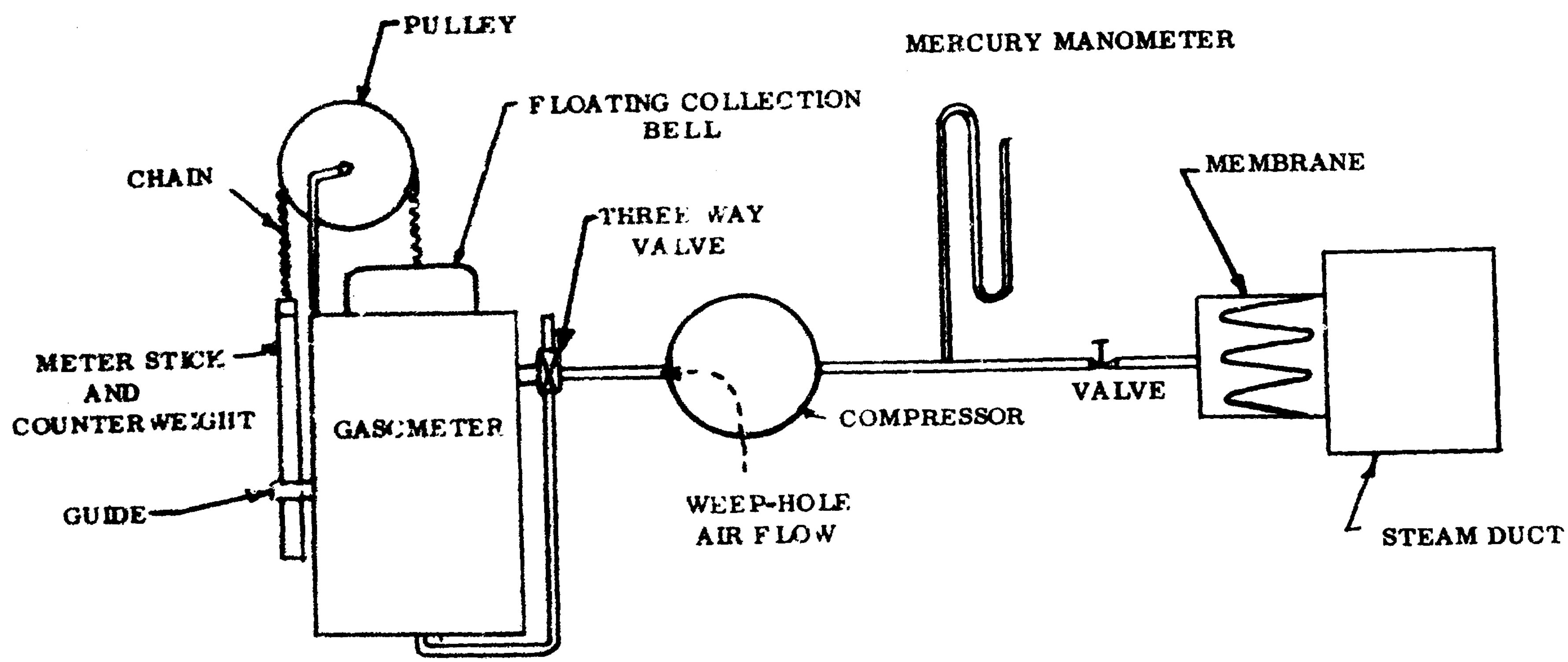



Figure 15. Volumetric Measurement of Weep-Hole Air

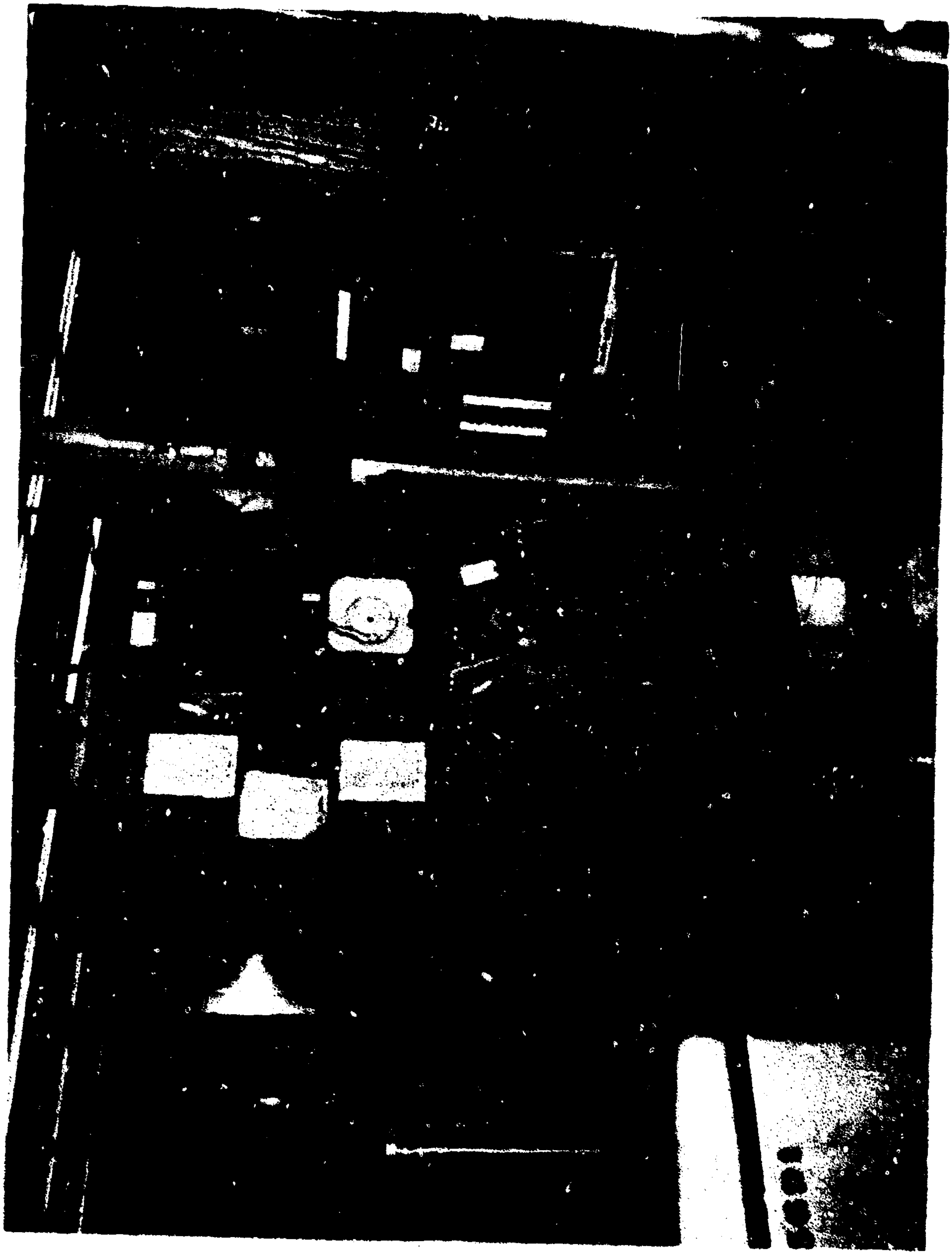


Figure 16. Compressor Discharging to Spirometer

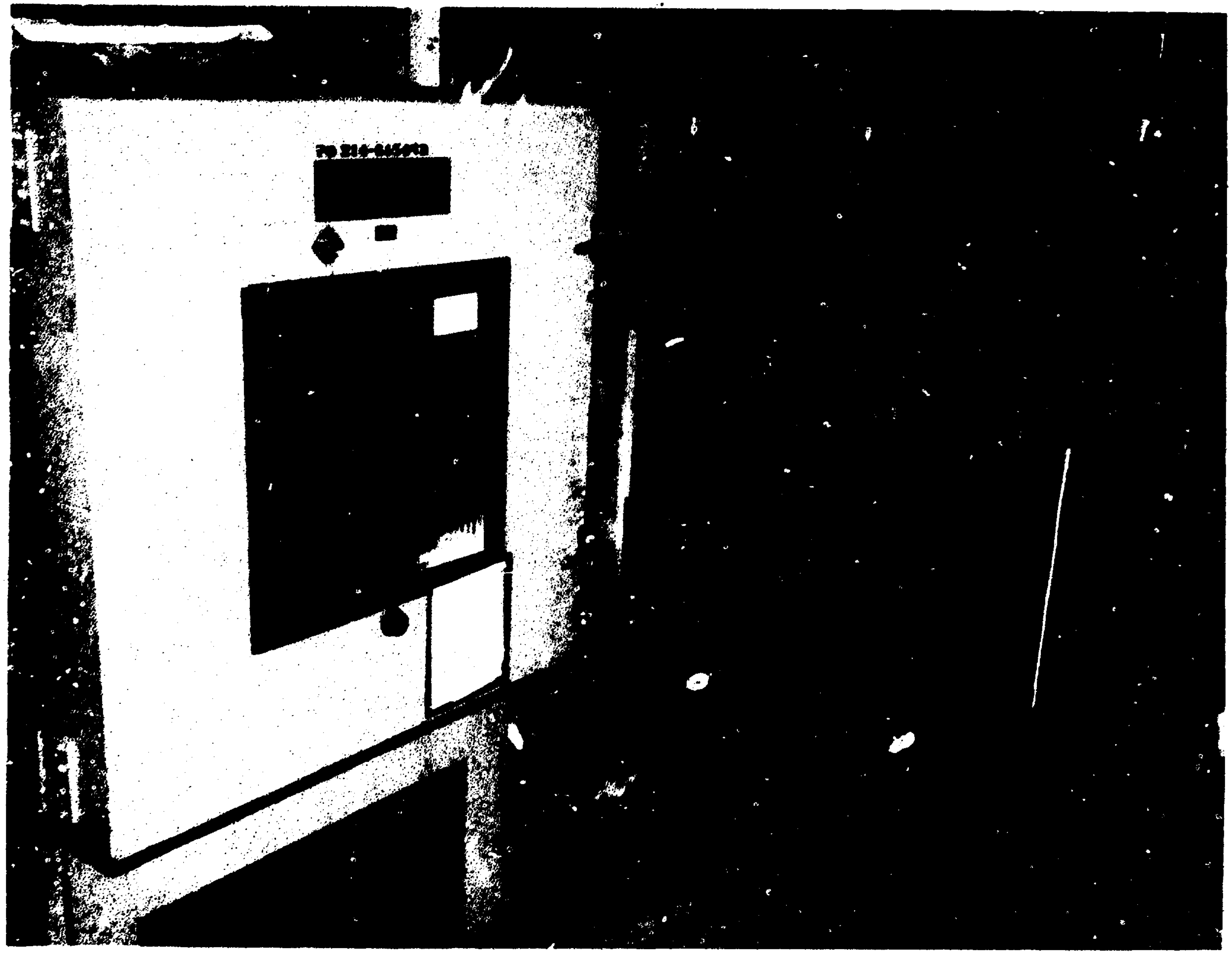


Figure 17. Humidity Chamber - Motor Compressor

TABLE IV. COMPRESSOR WEEP-HOLE AIR, VALVE TO MEMBRANE CLOSED

| TEST NO. | 1 | 2 | 3 | 4 |
|---|---|---|---|---|
| Scale Reading Start, cm | 90.0 | 90.0 | 90.0 | 90.0 |
| Scale Reading End, cm | 22.7 | 38.7 | 51.5 | 52.3 |
| Reading Diff., cm | 67.3 | 51.3 | 38.5 | 37.7 |
| Flow Volume, liters* | 90.0 | 68.3 | 51.4 | 50.4 |
| Flow Time, Min. | 1.5 | 1.5 | 1.5 | 1.5 |
| Flow Rate at Bell, liters/min. | 60.0 | 45.6 | 34.3 | 33.6 |
| Temp at Bell, $^{o}F$ | 75.2 | 75.2 | 75.2 | 75.2 |
| Vol. Reduced to STP, liters/min. | 55.2 | 42.0 | 31.6 | 31.0 |
| Temp at Compressor $^{o}F$ | 75.2 | 150.0 | 210.0 | 212.0 |
| Vol. Through Compressor liters/min. | 55.2 | 52.0 | 43.0 | 42.3 |

*Conversion Factor 1.332 liters/cm of scale reading difference.

TABLE V. COMPRESSOR WEEP-HOLE AIR MEMBRANE* OPEN - HUMIDITY CHAMBER OFF

| TEST NO. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Scale Reading Start, cm | 90.0 | 90.0 | 90.0 | 90.0 | 90.0 |
| Scale Reading End, cm | 49.0 | 50.3 | 46.5 | 48.0 | 50.0 |
| Reading Diff, cm | 41.0 | 39.7 | 43.5 | 42.0 | 40.0 |
| Flow Volume, liters | 54.6 | 53.0 | 58.0 | 56.0 | 53.4 |
| Flow Time, min | 1.5 | 1.5 | 1.5 | 1.5 | 1.5 |
| Flow Rate at Bell, liters/min | 36.4 | 35.3 | 38.6 | 37.3 | 35.5 |
| Temp at Bell, $^{o}F$ | 75.2 | 75.2 | 75.2 | 75.2 | 75.2 |
| Vol Reduced to STP, liters/min | 33.4 | 32.4 | 35.5 | 34.3 | 32.6 |
| Temp at Compressor, $^{o}F$ | 204.0 | 208.0 | 204.0 | 200.0 | 205.0 |
| Vol Through Comp., liters/min | 41.4 | 40.5 | 44.0 | 42.3 | 40.6 |

* One-Mil membrane, 8000 $cm^2$ area

TABLE VI. COMPRESSOR WEEP-HOLE AIR, MEMBRANE OPEN - HUMIDITY CHAMBER ON

| TEST NO. | 1 | 2 | 3 | 4 | 5 |
|---|---|---|---|---|---|
| Scale Reading Start, cm | 90.0 | 90.0 | 90.0 | 90.0 | 90.0 |
| Scale Reading End, cm | 52.7 | 49.0 | 50.8 | 52.8 | 53.6 |
| Reading Diff., cm | 37.3 | 51.0 | 39.2 | 37.2 | 36.4 |
| Flow Volume, liters | 49.6 | 54.6 | 52.3 | 49.6 | 48.5 |
| Time, minutes | 1.5 | 1.5 | 1.5 | 1.5 | 1.5 |
| Flow Rate at Bell, liters/min | 33.1 | 36.4 | 34.1 | 33.1 | 32.3 |
| Temp at Bell, $^{o}$F | 76.0 | 76.0 | 76.0 | 76.0 | 76.0 |
| Flow of Dry Air STP, liters/min* | 29.4 | 32.4 | 30.3 | 29.4 | 28.7 |
| Temp at Compressor, $^{o}$F | 216.0 | 204.0 | 212.0 | 220.0 | 228.0 |
| Flow of Dry Air at Comp. Temp. liters/min | 49.4 | 44.4 | 41.5 | 40.6 | 40.0 |

| | |
|---|---|
| Average inlet vapor pressure @ 127.6$^{o}$F/70.4% RH | 7.53 cm Hg |
| Average outlet vapor pressure @ 113.2$^{o}$F/69.5% RH | 5.00 cm Hg |
| Average compressor inlet pressure | 4.16 cm Hg |
| Mean log differential pressure | 1.82 cm Hg |
| Measured vapor flow, (STP) | 16.7 cc/sec |
| Area of one-mil cellulose acetate membrane | 8000 cm$^2$ |
| Apparent computed permeability (std cc/sec) (cm thick) / (cm$^2$) (cm Hg) | 2900 x $10^{-9}$ |

*NOTE: Water condensed in tube to spirometer; hence, the table assumes 100% relative humidity at the spirometer bell temperature.

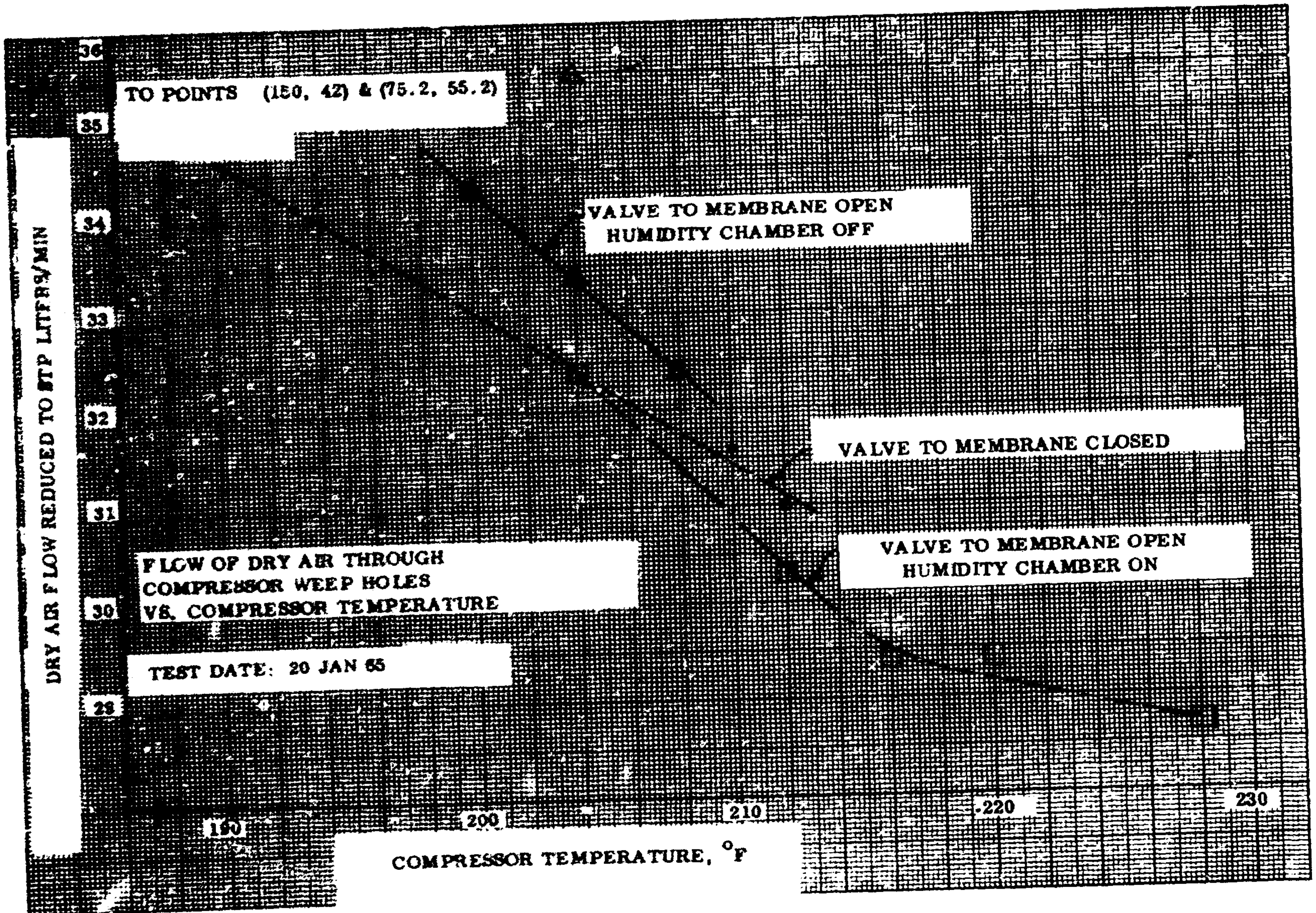



Figure 18. Air Flow vs Compressor Temperature

TABLE VII. LEAK RATE TEST

| ELAPSED TIME MIN | ABSOLUTE* PRESSURE AT CONE TORR | ABSOLUTE PRESSURE AT 32°F TORR | VOLUME GAS STP, CC | VOLUME CHANGE STP CC | LEAK RATE CC/MIN. |
|---|---|---|---|---|---|
| Start | 45.6 | 50.0 | 41.5 | -- | -- |
| 15 | 73.6 | 80.5 | 67.0 | 25.5 | 1.70 |
| 15 | 104.0 | 114.0 | 95.0 | 28.0 | 1.87 |
| 15 | 142.0 | 155.0 | 129.0 | 34.0 | 2.27 |
| 15 | 168.0 | 183.0 | 152.0 | 23.0 | 1.53 |
| 15 | 219.0 | 239.0 | 199.0 | 47.0 | 3.13 |
| 15 | 271.0 | 303.0 | 252.0 | 53.0 | 3.54 |
| 15 | 351.0 | 383.0 | 319.0 | 67.0 | 4.46 |
| 15 | 416.0 | 455.0 | 378.0 | 59.0 | 3.94 |
| 15 | 495.0 | 540.0 | 449.0 | 71.0 | 4.74 |
| 15 | 549.0 | 600.0 | 500.0 | 51.0 | 3.40 |
| 150 Total | | | | 458.5 Total | |

NOTES: Membrane was one-mil thick, 8000 $cm^2$ cellulose acetate.

$$\text{Average leak rate} = \frac{458.5}{150} = 3.06 \ \frac{cc}{min}$$

*Cone Temperature was - 10°F average.

d. Power Measurement

In the original proposal, the vapor-compressor concept indicated that a very low power would be required; yet the compressor manufacturer, Leiman Brothers, insisted that a 1.5-horsepower motor would be needed to power the model 295-2x3 compressor. With the compressor and cone set up as shown in Figure 13, the power drawn by the motor was measured. At first, measurements were made with an amp probe and a voltmeter.

Table VIII gives the results of the amp-probe and voltmeter readings. In all three tests the valves to the membranes were closed; hence, the compressor merely pulled a vacuum against the mercury manometer. The discharge in test no. 1 was the condenser-cone with the drain plug removed. In test no. 2, the drain plug was replaced and was closed. In test no. 3, the vacuum pump was running at the same time as the compressor, reducing the absolute cone pressure to 11.6 psia. The test indicated that discharging to the cone under a slight vacuum made no measurable change in power from discharging to the cone at atmospheric pressure. Test 2, however, indicated an increase in power from 2100 to 2750 watts by increasing to a compression ratio of 26:1 from the 17.5 compression ratio of Test 1.

The results shown in Table VIII indicate that power factor readings were important; hence, polyphase power measurements were taken using a GE Watt-VAR meter. This instrument is used with a recording wattmeter on three-phase circuits to record both watts and vars on a single chart. The results read from the chart are given in Table IX. The number and type of test was increased over the amp-volt tests, resulting in sufficient data to plot the curve of Figure 19.

The tests labeled E and Er (which are vacuum inlet-vacuum cone outlet tests) have compression ratios of 10.7 and 14.1, respectively. The power for these tests is almost the same as tests A and B where the compression ratio is unity. This confirms the prediction of low power for a vacuum-to-vacuum condition; however, the absolute power is still far above the theoretical. From the weep-hole measurements, a flow of 33 liters/min is typical. The theoretical power to compress this air-flow from 0.61 psia to 8.58 psia (as per test Er) is 220 watts. This is still well below the actual measured power of 1660 watts. Since the motor itself required 1120 watts, the apparent additional power for compression is 540 watts, which is relatively close to the theoretical 220 watts. Pump inefficiency and friction losses apparently make up the remaining power. See Appendix I for calculation of compressor power.

e. Water Recovery by Vapor Permeation with Subsequent Vapor Compression

After running preliminary tests, measuring weep-hole air, leakage rates, and power, a test was run wherein water was collected by vapor-compression and condensation. The set up of Figure 13 was used with one exception—the cone was chilled by immersing in a mixture of alcohol and dry ice. The membrane assembly utilized the two-mil membrane.

TABLE VIII. POWER MEASUREMENTS - AMP PROBE AND VOLTMETER

| ITEM | TEST NO. 1 | TEST NO. 2 | TEST NO. 3 |
|---|---|---|---|
| Pump pressure inlet, psia | 0.835 | 1.18 | 0.738 |
| Pump Pressure Discharge, psia | 14.6 | 30.6 | 11.6 |
| Pressure Ratio Outlet/Inlet | 17.5 | 26.0 | 15.7 |
| Line to ground, Volts | 280.0 | 278.0 | 280.0 |
| Line to Ground Current, Amps | 2.5 | 3.3 | 2.5 |
| Apparent Power, Volt Amp | 700.0 | 916.0 | 700.0 |
| Power Total (Three lines) Volt Amp | 2100.0 | 2750.0 | 2100.0 |

NOTES: TEST NO. 1 - Vacuum Inlet, open cone outlet

TEST NO. 2 - Vacuum Inlet, closed cone outlet

TEST NO. 3 - Vacuum Inlet, vacuum outlet

TABLE IX. POWER MEASUREMENTS BY WATT-VAR METER

| TEST LETTER | COMP. INLET PSIA | COMP. OUTLET PSIA | COMP. RATIO OUTLET INLET | POWER WATTS | REACTIVE LOAD VARS | APPARENT POWER VOLT-AMP | POWER FACTOR |
|---|---|---|---|---|---|---|---|
| A | 14.85 | 14.85 | 1.0 | 1620 | 1520 | 2020 | 0.729 |
| B | 14.85 | 14.85 | 1.0 | 1660 | 1500 | 2240 | 0.742 |
| C | 0.905 | 14.85 | 16.5 | 2160 | 1360 | 2550 | 0.846 |
| Cr | 0.909 | 14.85 | 16.4 | 2080 | 1400 | 2510 | 0.830 |
| D | 1.392 | 30.8 | 22.1 | 2880 | 1180 | 3120 | 0.925 |
| E | 0.610 | 6.54 | 10.7 | 1640 | 1520 | 2230 | 0.734 |
| Er | 0.610 | 8.58 | 14.1 | 1660 | 1500 | 2200 | 0.742 |
| F | -- | -- | -- | 1120 | 1680 | 2020 | 0.554 |
| Fr | -- | -- | -- | 1120 | 1660 | 2000 | 0.560 |

NOTES:

| | | |
|---|---|---|
| NOTES: | Test A | Room inlet, Room Outlet |
| | Test B | Room inlet, open cone outlet |
| | Test C, Cr | Vacuum inlet, open cone outlet |
| | Test D | Vacuum inlet, closed cone outlet |
| | Test E, Er | Vacuum inlet, vacuum cone outlet |
| | Test F, Fr | Belts to pump removed |

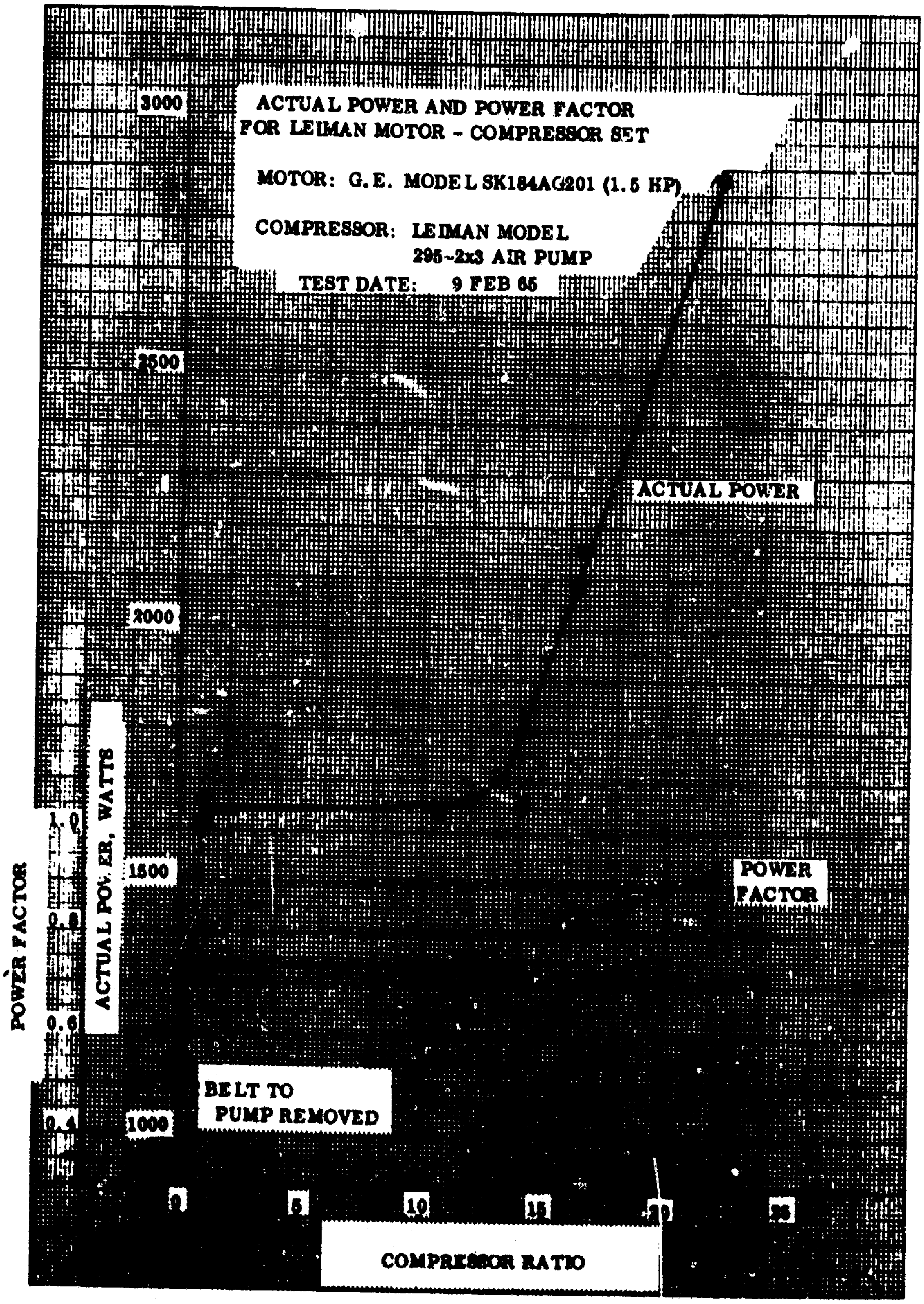



Figure 19. Actual Power and Power Factor vs. Compression Ratio

In the first few minutes of operation of a previous test where dry ice alone was the coolant, the vapor was frozen to ice on the cone. As the compressor continued pumping water vapor and weep-hole air, the heat of compression of the air caused the ice to melt to water, with subsequent loss of water from the cone. In a matter of minutes, all condensation stopped.

With the above observation in mind, the equipment was set up with the condenser cone set in a container filled with alcohol and dry ice to overcome the heat of compression of the weep-hole air. Measurements on the cone surface showed temperatures of -105$^{\circ}$F; nevertheless, the air-vapor temperature in the cone which was measured twice, was recorded once at +65$^{\circ}$F and once +50$^{\circ}$F. The vapor condensed into snow over most of the cone surface. The exhaust from the compressor dislodged some of the snow and blew it out of the open cone. (The cone was open because of the compressor need for weep-hole air.) The test was stopped before all the condensate was lost. Most of the water collected (13 cc) came from the upper portion of the cone (which was not immersed in the alcohol-dry ice mixture), where the vapor had condensed as ice; thus, it was concluded that there had been little actual loss of water, and the test was meaningful. Table X discussed below, shows the resulting data.

In previous test work with the same two-mil membrane, a pressure of 2.28 cm Hg absolute was obtained at the compressor inlet. In this test, the average absolute pressure reached was 3.50 cm Hg; hence, the membrane assembly was checked for leaks by recording the change in pressure in a closed evacuated system for a time of one hour. By computation, 1.1 cc/min was leaking, but 464 cc/min were permeating through the membrane; thus, leakage was insignificant.

## 3. DISCUSSION OF RESULTS - VAPOR PERMEATION WITH SUBSEQUENT VAPOR COMPRESSION

In Table X, the inlet and outlet temperatures and relative humidities refer to measured values of the humid air in the "steam" duct, points A and C, respectively, of Figure 13. Knowing the temperature, the saturation vapor pressure can be found in any standard steam table. Multiplying the saturation pressure by the relative humidity results in the vapor pressure at "inlet" and "outlet." The compressor inlet pressure was taken as the manometer reading. The logarithmic mean pressure difference was then computed by the equation

$$\ln \Delta P = \frac{(pv_1 - pv_3) - (pv_2 - pv_3)}{\ln \dfrac{(pv_1 - pv_3)}{(pv_2 - pv_3)}} \qquad \text{(IV-7)}$$

where $pv_1$ is the vapor pressure inlet - humid air

$pv_2$ is the vapor pressure outlet - humid air

$pv_3$ is the vapor pressure inlet - compressor

TABLE X. VAPOR PERMEATION WITH SUBSEQUENT VAPOR COMPRESSION

MEMBRANE TESTED: Two-Mil Cellulose Acetate

Computed Area 7560 $Cm^2$

| | |
|---|---|
| Test Time, minutes | 35.0 |
| Inlet Temperature, $^oF$ | 124.3 |
| Inlet Relative Humidity % | 63.5 |
| Inlet Vapor Pressure, cm Hg* | 6.71 |
| Outlet Temperature, $^oF$ | 111.3 |
| Outlet Relative Humidity % | 68.4 |
| Outlet Vapor Pressure, cm Hg* | 4.64 |
| Compressor Inlet Pressure, cm Hg | 3.50 |
| Log Mean Pressure Difference, cm Hg* | 1.99 |
| Water Recovered, g | 13.0 |
| Water Vapor at STP, cc/sec* | 7.7 |
| Computed Apparent Permeability* | |
| $\frac{\text{(Std cc/sec) (cm Thick)}}{(\text{cm}^2)\ \text{(cm Hg)}}$ | $2570 \times 10^{-9}$ |

* Computed values

In the next calculation, the recovered 13 grams of water per test time of 35 minutes, was converted to a vapor rate of 7.7 cc/sec. Then by substitution into equation (IV-6), the permeability of $2570 \times 10^{-9}$ (std cc/sec) (cm)/(cm$^2$) (cm Hg), was computed.

A permeability of $2570 \times 10^{-9}$ is lower than the manufacturer claim of $5000 \times 10^{-9}$, but higher than the $1500 \times 10^{-9}$ value assumed for design purposes (see Figure 8); thus, the results of the test were satisfactory.

4. CONCLUSION - VAPOR PERMEATION WITH SUBSEQUENT VAPOR COMPRESSION

The compressor construction requiring the weep holes for bleed air prevented proper checkout of the theory. The weep-hole leakage was over 30 times the expected vapor flow, preventing condensation of vapor at room temperature. The key factor in the choice of cellulose acetate as the membrane is its high selectivity for water vapor instead of air. For the two-mil, 7560-cm$^2$ area, the computed permeability of oxygen is 0.094 std cc/min, and for nitrogen 0.131 std cc/min (see Appendix IV). At a nominal 30 liters per minute weep-hole flow, there is 136,000 times more air passing through the compressor by weep-hole leakage than by membrane permeation. Even if two one-mil membranes were used in parallel with four times the air permeation there would still be nearly 34,000 times more weep-hole air to handle than the theoretical total. Consequently, the cone could not be kept at room temperature by air cooling and the power consumption was high.

The theoretical work of a compressor is directly proportional to the initial flow volume. If the 30 liters/min of weep-hole air are removed, leaving only one liter/min of water vapor, the work should be reduced by 1/30; hence, the peak power of 2880 watts measured when the compressor discharged to a closed cone would be reduced to 96 watts. (This does not consider losses such as friction, etc., and the differences of properties between air and steam. See Appendix I.)

The computed air-leakage rates through the membrane are insignificant in comparison to the vapor permeability rate. For the two-mil membrane, a rate of 7.7 cc/sec or 462 cc/min permeated, with a computed 1.1 cc/minute of leakage. Thus, the vapor collected as ice can be said to have passed through the membrane by permeation, not by leakage.

From the test results, it can be concluded that water can be collected by selective permeation through a membrane after the vapor has passed through a compressor; however, since the compressor selected could not be run as desired, and since no other manufacturer contacted could supply a better compressor with the proper capacity, the complete theory could not be checked. It would be informative to run tests with a "flight type" compressor, capable of the desired vacuum and requiring low power even if the rated volumetric capacity was lower than the desired 16.3 std cc/sec. At least this type of test could be used as a step in developing flight type vapor separators.

For the two-mil membrane, the measured rate of 7.7 cc/sec of vapor is equivalent to 1.18 pounds of water/day. To obtain the desired 2.5 lbs/day at the humidity conditions of this test, a second unit in parallel would be needed. The test equipment was designed for two such units. The vendor rating for the cellulose acetate membrane is a permeability of $5000 \times 10^{-9}$ (std cc/sec) (cm)/($cm^2$) (cm Hg). The apparent permeability of the vapor compression system of $2570 \times 10^{-9}$ indicates an efficiency of 51%.

## SECTION V

## WATER VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT

### 1. THEORY AND DESIGN

As in vapor permeation with subsequent vapor compression, this method uses a membrane highly selective to the passage of water vapor to separate the vapor from an air stream. The differentiating features of the freeze-out concept are the method of obtaining vapor flow and the method of collection.

In theory, for this concept, a vacuum pump would be used to draw an initial vacuum on one side of a membrane, causing water vapor to permeate into the low pressure region. By rapidly condensing or freezing the water vapor, the vacuum would be maintained without the continuous use of a pump. Since noncondensable gases would eventually permeate the membrane and interfere with the condensation process, periodic purging of the system would be needed. To test this theory, the same equipment designed for the vapor permeation-compressor concept was used. A rearrangement of the equipment and a method of cooling the cone were the only changes needed.

### 2. TEST DESCRIPTION

#### a. Arrangement of Test Equipment

The equipment arrangement is shown in Figure 20. Figure 21 shows a close-up photograph of the cone assembly and the membrane assembly. Moist air is piped from a humidity chamber (not shown) to the "steam" duct. This is the same duct as shown in Figure 13, and the same cellulose acetate membrane assemblies were bolted to the duct. Copper tubing connects the membrane assemblies to the condenser cone (see Figure 22).

Since, by theory, the flow of water vapor into the cone was greatly dependent on the speed of condensation, dry ice was chosen as the coolant; however, the ice and cone were arranged so that the ice did not touch the cone pin fins. Instead, the cone was covered with a "blanket" of cold air caused by the sublimation of the dry ice. The Leiman compressor was used as the vacuum pump.

#### b. Periodic Pump-down Test — With Subsequent Vapor Freeze Out

The first test of this concept used a periodic pumpdown to evacuate the cone. At the start, the cone pressure was reduced to 4.31 cm Hg absolute and then the pump was turned off. After 45 minutes, a second set of readings was taken, and the cone was at 24.9 cm Hg absolute. The compressor was started, and the cone again pumped down to 4.31 cm Hg and the compressor turned off. Measurements were taken 15 minutes later. The pumping cycle was repeated, but for the next 105 minutes the time interval between "pumpdowns" was 15 minutes. In this way, the absolute pressure was not

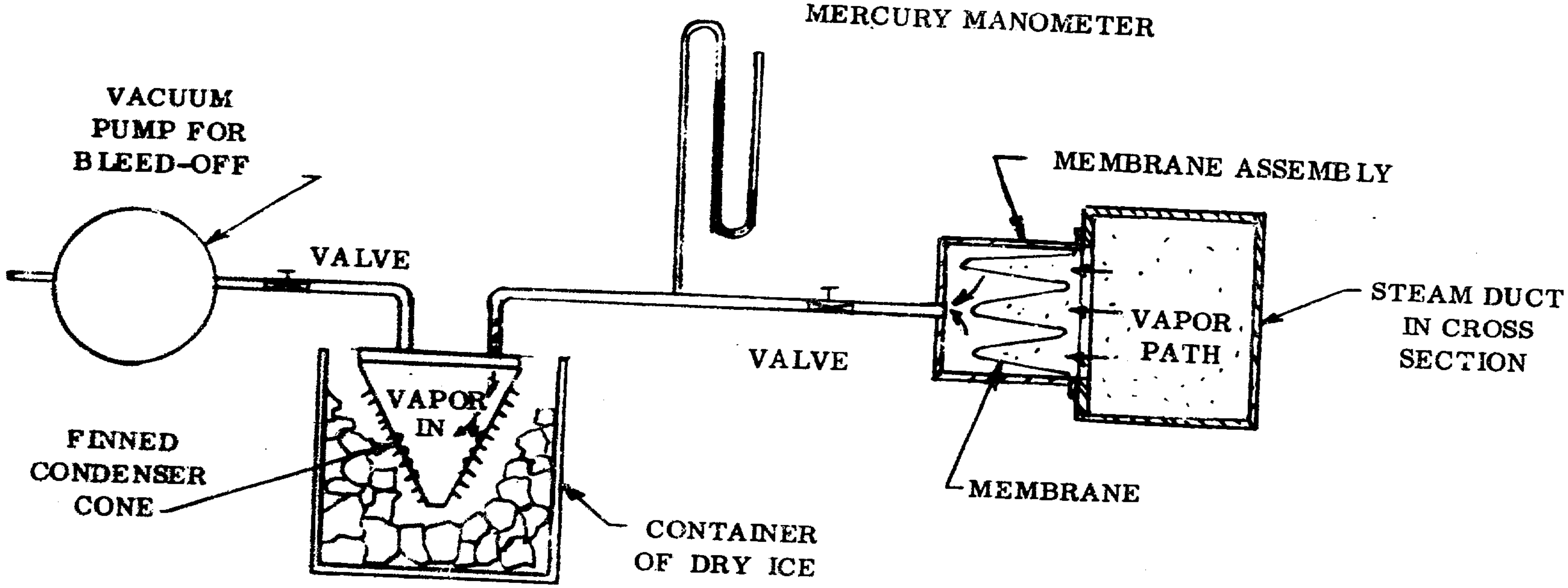



Figure 20. Setup for Vapor Permeation with Subsequent Freeze Out

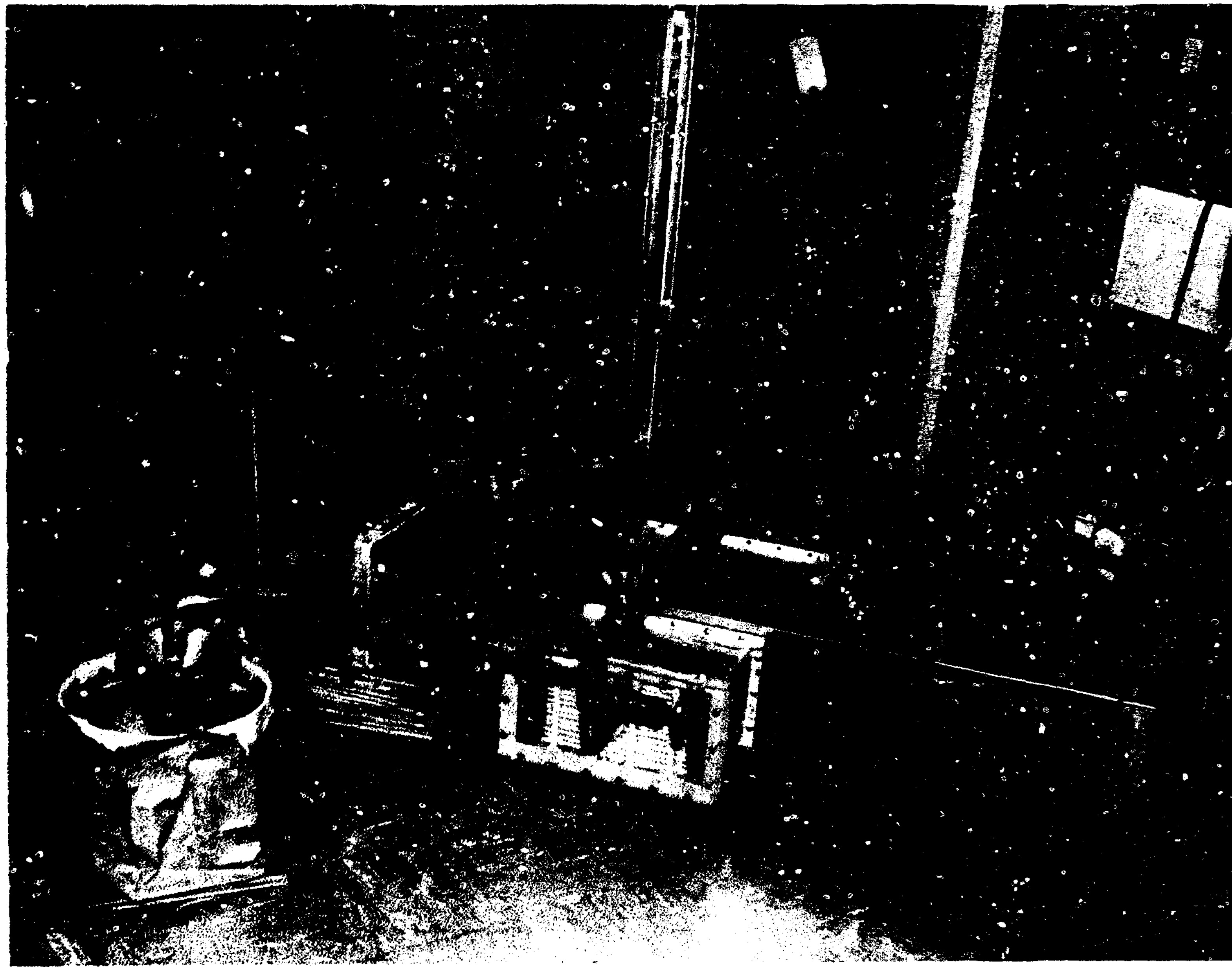


Figure 21. Cone Condenser in Dry Ice

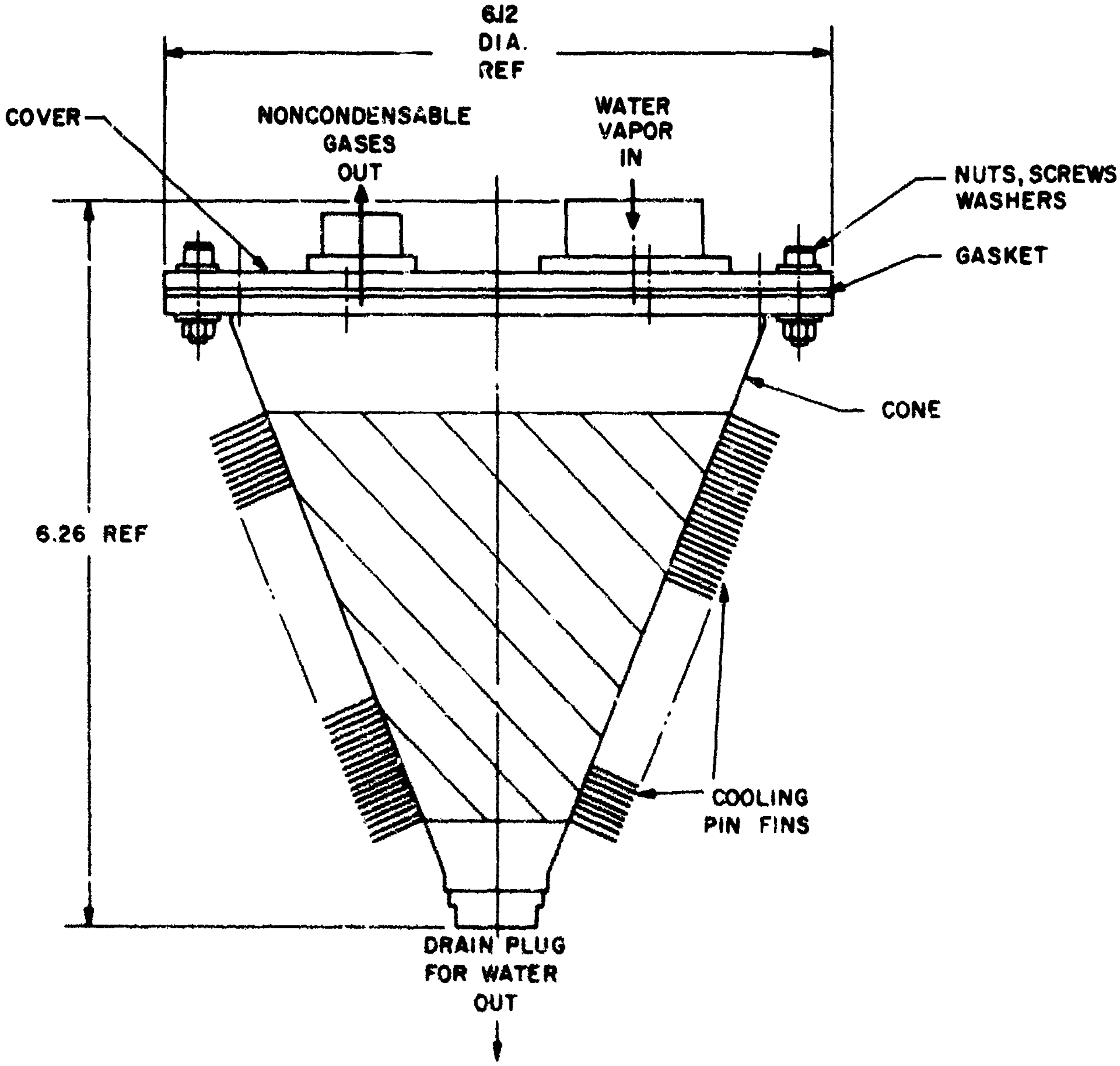



Figure 22. Water Vapor Condenser

permitted to vary as widely as in the first 45 minutes. Typically, the cone pressure at the end of a 15 minute period was 14 cm Hg.

The results are given in Table XI. Because of the two sets of test conditions, the vapor pressure was computed separately for the first 45 minutes and the second 105 minutes. The logarithmic mean pressure difference was next computed for each set of conditions.

$$\ln \Delta p = \frac{(pv_1 - pv_c) - (pv_2 - pv_c)}{\ln \dfrac{(pv_1 - pv_c)}{(pv_2 - pv_c)}} \qquad \text{V-1}$$

$pv_1$ is the vapor pressure inlet, cm Hg

$pv_2$ is the vapor pressure outlet, cm Hg

$pv_c$ is the vapor pressure cone, cm Hg

Having determined $\ln \Delta p$ for the first 45 minutes and the second 105 minutes, equation V-2 was used to compute the weighted average

$$p_w = \frac{(\Delta p_1)\ (45) + (\Delta p_2)\ (105)}{45 + 105} \qquad \text{V-2}$$

where $p_w$ is the weighted pressure difference

$p_1$ is the pressure difference for 45 minutes

$p_2$ is the pressure difference for 105 minutes.

Because the measured absolute pressure in the cone rose above the computed vapor pressure of the humid air, two calculations of permeability were made. For the first calculation, the vapor pressure of ice, 0.46 cm Hg, was taken as the minimum vapor pressure in the cone. Then equations V-1 and V-2 were used. Permeability was then computed by substituting in equation IV-6 repeated and rearranged here for convenience,

$$P = \frac{G\ell}{A\ (\Delta p)} \qquad \text{IV-6}$$

Since there was a periodic pump operation, it was possible to measure cone pressure only at the start and end of the cycle. The pumpdown resulted in a cone pressure of 4.31 cm Hg absolute; hence, for the second calculation of permeability, the vapor pressure in the cone was taken as 4.31 cm Hg, and equations V-1 and V-2 were used

TABLE XI. VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT – PERIODIC PUMPDOWN

MEMBRANE TESTED: One Mil Cellulose Acetate
Area of 8000 $cm^2$

| | |
|---|---|
| Total Test Time, Minutes | 150.0 |
| **Conditions, First 45 Minutes** | |
| Average Inlet Temp °F | 135.0 |
| Average Inlet R.H. % | 68.5 |
| Average Vapor Pressure, cm Hg | 8.52 |
| Average Outlet Temp °F | 120.0 |
| Average Outlet R.H. % | 70.2 |
| Average Vapor Pressure, cm Hg | 5.69 |
| **Conditions, Next 105 Minutes** | |
| Average Inlet Temp °F | 118.0 |
| Average Inlet R.H. % | 71.1 |
| Average Inlet Vapor Pressure, cm Hg | 5.44 |
| Average Outlet Temp °F | 107.0 |
| Average Outlet R.H. % | 70.0 |
| Average Outlet Vapor Pressure | 3.89 |
| Cone Min Pumpdown Pressure, cm Hg | 4.31 |
| Vapor Pressure of Ice, 32°F | 0.46 |
| Weighted Average Log mean Differential Pressure, cm Hg (Based on 0.46 in ice) | 4.95 |
| Water Recovered, g | 21.0 |
| Water Vapor at std cc/sec | 2.9 |
| Computed Apparent Permeability $\frac{(\text{STD cc/sec})\ (\text{cm thick})}{(\text{cm}^2)\ (\text{cm Hg})}$ | $186 \times 10^{-9}$ |
| Weighted Pressure Difference Based on pumpdown to 4.31 cm | 1.46 |
| Computed Apparent Permeability $\frac{(\text{STD cc/sec})\ (\text{cm thick})}{(\text{cm}^2)\ (\text{cm Hg})}$ | $631 \times 10^{-9}$ |

to determine the weighted pressure difference. This pressure was substituted in equation IV-6 to obtain the second value of permeability.

For both permeability calculations, the values of $186 \times 10^{-9}$ and $631 \times 10^{-9}$ are far below the vendor rating of $5000 \times 10^{-9}$ (std cc/sec) (cm thick) / ($cm^2$) (cm Hg). One major reason is the "periodic" pumpdown. The rate of pressure rise in the cone is too rapid to maintain a large pressure difference between cone inlet and the humid air stream; hence, the indications are that flow takes place for only a short period of time and not the entire 150 minutes. In addition, when the cone pressure exceeded the vapor pressure, there should have been a reverse diffusion of vapor molecules from the cone to the membrane. Nevertheless, water was collected; hence, the next test was run with only a single pumpdown to see if there would be a vapor flow despite a continuously rising cone pressure.

c. Single Pumpdown Test - with Subsequent Vapor Freeze Out

In the second 105 minutes of the periodic pumpdown test, the inlet vapor pressure at 118°F, 71.1% relative humidity was 5.94 cm Hg, and the outlet vapor pressure was 3.89 cm Hg. But the cone was pumped down to only 4.31 cm Hg. If the cone pressure were all water vapor, as was assumed for permeability calculations, there would have been a reverse flow of vapor for part of the cycle. Otherwise vapor would flow into the cone despite a rising pressure. To check this possibility, another test was run.

In this test, the equipment arrangement of Figure 20 was used again. After an initial pumpdown to 4.56 cm Hg, the pump was stopped and the valve closed. After 2.5 hours, the cone was opened and examined. Only a trace of condensate was found in the cone.

The test data showed a continuous rise in cone pressure as expected. The average condition of the humid air inlet was a temperature of 113°F and relative humidity of 71.5%, with a resulting vapor pressure of 5.15 cm. This pressure was exceeded within 15 minutes; hence, water vapor will not flow against a continuously rising pressure in the cone. Evidently, a vapor pressure equilibrium is quickly reached.

Because a record of cone pressure increase was made, the test data was used to compute leakage. This is shown in Table VII and explained in paragraph IV.2.c. The leak rate was found to be 3.06 cc/min. Because of the leak, noncondensable gases (air) would be present in the cone. Those gases would inhibit condensation of the water vapor. If the vapor would not condense, there would not be a vapor pressure differential, and subsequently vapor flow would stop; hence, a third approach to move and condense vapor was made by continuously pumping bleed air from the cone.

d. Continuous Vacuum Bleed Test - with Subsequent Vapor Freeze Out

Since water had been collected by periodic pumpdown, but not by a single pumpdown, it was desirable to determine the effect of a continuous vacuum bleed from the cone. Such a test was run with the same test setup as in Figure 20. The valve between the cone and vacuum pump was used to set and maintain the cone pressure at 6.7 cm Hg absolute for the entire test. After two hours 15 minutes of operation, the pump was turned off, and the cone was opened.

Examination of the cone showed only minor condensate; hence, the plate cover on the membrane housing was removed to determine if there was any moisture. Vapor had permeated through the membrane and condensed on the metal plate cover but because the rectangular shape of the housing made collection difficult, part of the water was lost. The total quantity of measured and estimated water was 24 cc.

Examination of the test data recorded in Table XII shows that the cone pressure had been inadvertently set higher than the vapor pressure in the duct; thus the vapor flow through the membrane should not diffuse to the cone if the cone pressure was due to vapor. Collection of water at the membrane housing probably occurred because of a low vapor pressure region immediately behind the membrane. This would occur and be maintained if the vapor condensed on the metal covers, which is what happened. Unfortunately, there was no thermocouple on the metal plate; hence, an estimate of 77°F was taken as a typical room temperature. The vapor pressure at 77°F saturated, is 2.4 cm Hg, which would provide a pressure differential for flow; thus, with two estimated values, the total water collected and the vapor pressure behind the membrane, a value of $544 \times 10^{-9}$ (std cc/sec) (cm)/(cm$^2$)(cm Hg) was computed for permeability.

In order to eliminate condensation in the membrane housing, and to permit observation of the process, the metal covers were replaced with thick plexiglas covers on both the membrane and the cone, and the continuous bleed test was repeated. With a plexiglas cover on the cone, condensation of vapor was easily observed. Table XIII shows the results, and once again the calculations showed that the vapor pressures for the temperature and humidity conditions of flow were less than the absolute pressure of the cone. Nevertheless water was collected and having observed condensation in progress, such a pressure condition was not suspected. Immediately, there must be suspicion of an incorrect pressure differential.

There are two possible conditions whereby the true pressure differential is different from those read and computed: (1) the cone pressure is not all water vapor but also contains noncondensables; (2) the temperature-humidity readings are in error by indicating values too low. For the first condition, the partial pressure of the vapor would be only a fraction of the absolute total pressure; and for the second condition, a higher temperature or humidity would give a higher water partial pressure. In either case, there would be a positive vapor pressure difference, rather than a negative vapor pressure difference.

Since the humidity sensors were checked and the instrument calibrated at the beginning of the series of tests, the first condition of air in the cone seems more likely. From the test of paragraph IV.2.c, a leak rate of 3.06 cc/min can be expected to enter the cone. If this flow is assumed to be mostly air (because the partial pressure of the air

TABLE XII. VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT — CONTINUOUS VACUUM BLEED TEST

MEMBRANE TESTED: One Mil Cellulose Acetate
Area of 8000 $cm^2$

| | |
|---|---|
| Test Time, Minutes | 135.0 |
| Average Inlet Temp, $^oF$ | 115.5 |
| Average Inlet R.H. % | 71.3 |
| Average Inlet Vapor Pressure, cm Hg | 5.5 |
| Average Outlet Temp, $^oF$ | 102.8 |
| Average Outlet R.H.% | 71.7 |
| Average Outlet Vapor Pressure, cm Hg | 3.8 |
| Average Cone Pressure, cm Hg | 6.7 |
| Vapor Pressure at $77^oF$, cm Hg * | 2.4 |
| Pressure Difference, cm Hg (Inlet/Outlet to $77^o$ Vapor) | 2.14 |
| Actual Water Recovered, g | 12.0 |
| Estimated Water Recovered, g** | 12.0 |
| Total Estimated Vapor Flow, std cc/sec | 3.7 |
| Estimated Apparent Permeability $\frac{\text{(std cc/sec) (cm thick)}}{(cm^2)\ \text{(cm Hg)}}$ | $544 \times 10^{-9}$ |

*Estimated temperature of plate.
**Estimate of spilled water.

TABLE XIII. VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT — CONTINUOUS VACUUM BLEED TEST

MEMBRANE TESTED: One Mil Cellulose Acetate
Area of 8000 $cm^2$

| | |
|---|---|
| Test Time, Minutes | 210.0 |
| Average Inlet Temp, °F | 124.5 |
| Average Inlet R.H., % | 69.6 |
| Average Inlet Vapor Pressure, cm Hg | 6.91 |
| Average Outlet Temp °F | 111.8 |
| Average Outlet, R.H. % | 69.9 |
| Average Outlet Vapor Pressure, cm Hg | 4.89 |
| Max. Cone Pressure, cm Hg | 7.20 |
| Vapor Pressure at 32°F, cm Hg | 0.46 |
| Pressure Difference, cm Hg (Based on 0.46 cm Hg of ice) | 4.93 |
| Water Collected, g | 29.0 |
| Water Vapor Flow, std cc/sec | 2.36 |
| Computed Apparent Permeability $\frac{(\text{std cc/sec})\ (\text{cm thick})}{(\text{cm}^2)\ (\text{cm Hg})}$ | $184 \times 10^{-9}$ |

is about 100 times the vapor pressure of water upstream of the membrane), then the source of air would be known. The manometer reading only gives the total pressure; thus, there would be no way to know what portion of the total pressure was the actual vapor pressure inside the cone from the test data of Table XIII. Since ice was present, it was decided to use the vapor pressure of the ice at standard temperature as the vapor pressure inside the cone, 0.46 cm Hg. On this basis, the apparent permeability of the membrane was computed as $184 \times 10^{-9}$ (std cc/sec) (cm)/(cm$^2$) (cm Hg). This checks the $186 \times 10^{-9}$ of the periodic pumpdown test, Table XI.

If the assumption that air was in the cone was correct, then the air would tend to inhibit the condensation of the vapor. With the pump operating continuously, part of the permeated vapor would be drawn off. This would explain why the amount of recovered water was low, and the resulting calculated permeability was low.

e. Continuous Vacuum Bleed Test - Air Cooled Condenser

In this test, there was a continuous vacuum on the cone similar to the tests run in paragraph V.2.d; however, instead of using dry ice as a coolant, the cone was permitted to remain at room temperature. Condensate was formed in droplets on the plexiglas cover of the cone, but at a very slow rate; hence, the test was ended after 45 minutes without a measurable amount of water being collected.

f. Continuous Vacuum Bleed Test, New Membrane - With Subsequent Vapor Freeze Out

Because of the leakage in the one mil membrane, a new assembly was built using a two mil cellulose acetate membrane. A repeat of the test of paragraph V.2.d was made using the dry ice for a coolant.

Initially for this test the pump was started and the cone evacuated. During the pumping phase, the process was observed through the plastic covers. When a pressure of 2.2 inches Hg absolute (5.6 cm) was reached, there was no ice formed in the cone. This pressure is significant because it is well below the 7.20 cm cone pressure of Table XIII where ice was formed. The evacuation of the cone was continued slowly until the pressure reached 1.20 in Hg absolute (3.05 cm). At this pressure, a virtual "cloud burst" of condensate formed resulting in crystals of ice. The control valve to the pump was opened fully to draw the maximum vacuum possible with this equipment, which was 0.9 in Hg absolute (2.28 cm Hg) average.

The test was run for two hours 15 minutes, and the results are given in Table XIV. The water collected during this time period was 35 grams which is a vapor flow of 5.38 std cc/sec. For the computed vapor pressures of 6.30 cm Hg inlet, 4.55 cm Hg outlet and the measured cone pressure of 2.28 cm Hg, the logarithmic mean pressure difference by equation V-1 is 3.05 cm Hg. The permeability was computed as $1180 \times 10^{-9}$ (std cc/sec) (cm)/(cm$^2$) (cm Hg) by equation IV-6. This permeability is of the proper order of magnitude as compared to the vendor rating.

TABLE XIV. VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT – CONTINUOUS VACUUM BLEED TEST

MEMBRANE TESTED: Two Mil Cellulose Acetate
Area of 7560 $cm^2$

| | |
|---|---|
| Test Time, Minutes | 135.0 |
| Average Inlet Temp, °F | 123.0 |
| Average Inlet, R.H. % | 66.0 |
| Average Inlet Vapor Pressure, cm Hg | 6.30 |
| Average Outlet Temp, °F | 110.5 |
| Average Outlet R.H. % | 68.5 |
| Average Outlet Vapor Pressure, cm Hg | 4.58 |
| Average Cone Pressure, cm Hg | 2.28 |
| Log Mean Pressure Difference, cm Hg (Vapor to cone) | 3.05 |
| Water Collected, g | 35.0 |
| Water Vapor Flow, std cc/sec | 5.38 |
| Computed Apparent Permeability $\frac{\text{(std cc/sec) (cm thick)}}{\text{(cm}^2\text{) (cm Hg)}}$ | $1180 \times 10^{-9}$ |

## 3. DISCUSSION OF RESULTS - ALL TESTS OF VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT

The biggest question mark in the data of these tests which greatly affects the computed permeability is the actual pressure of the water vapor in the humid air stream and in the cone. In this, the freeze-out method, the vapor flow is directly proportional to the vapor pressure difference, and if the pressures are not known precisely, the flow for the correct pressure will not be known precisely nor will the permeability calculation be precise.

In paragraph V.2.d, two conditions whereby vapor pressures could have been in error were considered: (1) the cone pressure was not all water vapor; (2) the temperature humidity readings were too low. The first condition is a very likely one especially when the pump inlet was throttled, or the pump shut off. Because of the results of Table XIII, however, the second condition was investigated by having the sensors checked at the end of the test program. The sensing element gave low readings, though they were accurately calibrated at the program outset; thus, there may have been some additional error by condition (2). While there may be some doubt to the accuracy of permeability calculations, vapor was made to flow and was recovered by the freeze-out concept.

Notice the heading "Computed Apparent Permeability" in the tables. The word "Apparent" indicates the system permeability, including the effects of support corrugations, tube elbows, and the influence of noncondensables. For the one-mil membrane, the computed permeability is very low being only 184 x $10^{-9}$ to 631 x $10^{-9}$. The vendor rating is 5000 x $10^{-9}$. For the two-mil membrane, Table XIV, the apparent permeability was computed at 1180 x $10^{-9}$. This is of the proper order of magnitude as compared to the vendor rating.

There are two significant differences between the two mil membrane assembly and the one-mil membrane assembly. First, the two-mil membrane itself appeared to be more uniform than the one-mil membrane, having less surface flaws. Secondly, much better sealing was obtained in the two-mil membrane assembly because of the "learn by experience" process. The sealing was more effective as demonstrated by the vacuum pulled during the two-mil test. This was the lowest achieved in any test, being only 2.28 cm Hg absolute. There was not even the small leakage of 3.05 cm/min as computed for the one-mil membrane. With less leakage, there would be less noncondensables, and more effective condensation in the cone. Even when the freeze-out test on the two-mil membrane was followed by the vapor compression test (paragraph IV.2.e) and a leak did develop, it was only 1.1 cc/minute. It is concluded that the two mil membrane is the more practical material for such assemblies.

In the operation of the tests, one instrumentation change was made. The "outlet" humidity and temperature sensors were moved closer to the membrane, from point B to point C in Figure 13. (The same instrumentation was used in the freeze-out concept as was used in the compressor concept.) This sensor move should have given a truer representation of humidity conditions near the membrane than in previous tests since there was less chance of condensation in a shorter length of duct; however, with less condensation, a higher vapor pressure differential would result. Since permeability is inversely proportional to the vapor pressure difference (see equation IV-6) the computed permeability would be lower. The results with the two mil readings should, if anything, be more conservative than with the one mil reading.

An important visual observation was made. Before the running of the tests with the one mil membrane, the plastic plate on the membrane housing became covered with droplets of water while the humidity chamber was being brought to stabilization. Since the valve between the cone and membrane was shut, there was no place for this vapor to go. When the valve was open and the vacuum pump turned on, only part of the plastic cover became clear of water. Later, checks in a bath of water to determine the locations of leaks showed that the largest leakage occurred in the region of the clear plate.

Before drawing a conclusion from this phenomenon, it must be reported that when the two mil membrane which had been checked prior to the test as being leak free was installed in the duct, no such vapor condensation occurred on the plastic cover. The cover was clear while the humidity chamber was stabilized and while the test was run.

Now the hypothesis can be drawn. The condensed vapor on the plastic plate (of the membrane housing) during tests of the one mil membrane caused a local buildup of vapor pressure which rendered a large part of the membrane area as ineffective. Where there was a leak, some air was drawn in. Being less than saturated, the air partially evaporated the water on the cover and carried it to the cone where condensation took place. Permeation then occurred over the portion of membrane area that opened to the region virtually clear of vapor; thus, there was a third contributing factor to the low permeability of one mil membrane and that is a reduction in effective area.

To re-enforce the hypothesis of high local vapor pressure inhibiting permeation, when the freeze-out tests were being run on the one mil membrane, the assembly was not dismantled and dried between tests. It is quite possible, therefore, for the filter paper to have gradually become "loaded" with water so that the local region between the filter paper and membrane also became a barrier to diffusion. The combination of suction from the pump combined with the small air leakage would only move a fraction of the vapor possible with a dry membrane. Then, since the tests were run over a period of several weeks, the water would have had plenty of time to evaporate off the filter paper, and form a vapor barrier behind the membrane. The action of the pump, especially during throttled bleed tests, would move some of this vapor into the cone. This would explain why there was observation of condensation even though later calculations showed the cone pressure to be higher than the humid air vapor pressures. Some of the residual vapor from previous tests must have been pumped to the cone, and that was the observed condensate. After this initial condensation, very little could permeate because of the high local vapor pressure immediately behind the membrane.

There is additional evidence that the hypothesis is correct. Chronologically, on the two mil membrane the vapor freeze-out test preceeded the vapor compressor test. The membrane assembly was dry when installed on the steam duct. The description of the pumpdown phase emphasized the fact that no condensation occurred until the "cloud burst" at a cone pressure of 3.05 cm Hg absolute. In this way, the required pressure differential was obtained to cause vapor to flow into the cone by permeation of the membrane.

On the very next day, the vapor compression test was run. This time, during the period for stabilizing the humidity chamber, a very light coat of condensate was observed on the plastic cover. The vapor deposit disappeared when the compressor was run. Again the hypothesis seems to be proven. For the initial flow of vapor could have been the residual from the previous freeze-out test. Because of the low pressure created by the compressor pulling directly on the membrane, permeation would be quickly established; however, the test was only 35 minutes. If there was some residual vapor transferred, there would have been an apparently higher permeability, and that is exactly the result. The permeability by the compressor test was $2570 \times 10^{-9}$, over twice the $1800 \times 10^{-7}$ obtained by the freeze-out method. Since the second test was short, there probably was not sufficient time to average out the effects of water collection from residual vapor of the previous day.

4. CONCLUSION - VAPOR PERMEATION WITH SUBSEQUENT VAPOR FREEZE OUT

For the one mil membrane, the average flow obtained (Tables XI, XII, and XIII) is 9.1 grams/hr. This is equivalent to 0.48 lb/day. The collection rate for the two mil membrane was 0.82 lb/day. Theoretically, the collection rate for one mil should be twice the rate for two mil film (since areas were nearly the same and pressure differences were similar). Instead, the one mil unit passed less than 60% of the water of the two mil membrane; hence, permeabilities ranged from $184 \times 10^{-9}$ to $631 \times 10^{-9}$ for the one mil assembly, but $1180 \times 10^{-9}$ for the two mil assembly. The reasons offered to explain the low permeability rate of the one mil membrane were: (1) air in the cone inhibited condensation; (2) the humidity sensors did not give proper readings; and (3) because of high local vapor pressure, there was a reduction in effective area and full permeation could not take place.

By observing the condensation of residual vapor from previous tests, there was an incorrect conclusion that permeation was taking place; hence, the apparent conditions (1) and (2) were not discovered early enough to be checked and corrected. Because the two mil membrane assembly was started dry, and the process watched from the beginning, the condition number 3 was not encountered, resulting in a good quantity of vapor flow. It is therefore concluded that the apparent permeability for the two mil membrane of $1180 \times 10^{-9}$ is a valid number.

For the two mil membrane assembly, the rate of 0.82 lb/day is approximately 1/3 the required 2.5 lbs/day; hence, three units of this size would be needed to meet the required flow. Based on the manufacturer's permeability value of $5000 \times 10^{-9}$ and the test result of $1180 \times 10^{-9}$, the efficiency of the vapor freeze-out system is 23%.

5. COMPARISON OF VAPOR PERMEATION - COMPRESSION CONCEPT TO VAPOR PERMEATION - FREEZE-OUT CONCEPT

Comparing Table X with Tables IX-XIV, the water collection rate for the vapor compression method is seen to be higher than that of the vapor freeze-out method. For the same membrane assembly, the apparent permeability by the compression test is over twice the permeability computed for the freeze-out test. This occurs despite the high rate of weep-hole air flow in the compressor test. The reasons offered are: (1) The water vapor which may be a residual from previous tests (or from vapor permeation of the membrane during stabilization of the humidity chamber) is definitely drawn through the compressor. This lowers the local vapor pressure behind the membrane and permits effective permeation. Diffusion to the cone, however, is less positive, since diffusion flow is dependent on vapor pressure and can be hindered by a local high vapor pressure, or noncondensable gases in the flow path. (2) The compressor should deliver all the vapor to the cone, but the freeze-out method may not result in all the vapor entering the cone, (as witnessed in the test of Table XII, where the vapor condensed on the membrane housing cover plate). (3) As a final plus for the compressor method, the discharge of air and vapor from the compressor exit was turbulent. This would statistically favor more vapor impingement on the cone than in the freeze-out concept.

On the negative side for the compressor test are the following: (1) the possibility of water collection including residual vapor from a previous test giving a high permeability reading; (2) a colder cone was needed in the compressor test, but this was because of the heat of compressing weep-hole air; (3) the compressor power was very high because of weep-hole flow. The vacuum pump in the freeze-out test needed only to move small quantities of bleed air, and could, therefore, be of lower power than in the compressor test.

In neither concept did the water recovered in the cone ever equal the change of water vapor in the steam duct. Condensation in the duct accounted for the bulk of the water removal (see Appendix IV).

# SECTION VI

## MATERIALS SECTION

### 1. POROUS MATERIALS FOR WATER DROPLET REMOVAL

Several materials potentially suitable for use in the water droplet concept were tested for pressure-flow relationship. The test setup is shown in Figure 23. A sample of porous material was clamped between two flanges that were soldered to 2-inch O.D. copper tubes. Air was forced through the sample by a blower and the flow was controlled by varying the power supplied to the blower. The total pressure drop was measured by the piezometric ring located upstream of the sample. The flow was measured by a calibrated venturi meter.

With the instruments available in the laboratory, no measurable flow could be obtained from three material samples: (1) Millipore of 0.45 $\mu$ average pore size, (2) woven teflon of 25$\mu$ average pore size and (3) sintered teflon of 9$\mu$ average pore size. Later in the program, during the water droplet recovery tests, sufficient flow data was obtained for sintered teflon to plot the curve of Figure 24 (see paragraph III.3.

The pressure flow characteristic for air for each of the material samples listed below is best described by curves. Figures 24 through 26 are curves derived from the test data. Published curves exist for two of the sintered stainless steel materials except that the samples received were silicone treated; therefore, while the curves do not coincide, there is sufficient correlation of the curves to give a high degree of confidence in the test (see Figure 24).

The points were plotted on log-log paper, and the curves are straight lines. The basic equation of a straight line on this paper is:

$$Y = BX^A$$

where B is the value of Y when X = 1

and A is the slope of the curve if the length of the X scale and Y scale are the same.

Thus, the following equations were derived:

- Sintered stainless, 35$\mu$ — $\Delta p = 0.03\ V^{1.025}$
- Sintered stainless, 65$\mu$ — $\Delta p = 0.0021V^{1.428}$
- Sintered Kel-F 15$\mu$ — $\Delta p = 0.023V^{1.058}$
- Silicon treated cotton — $\Delta p = 0.0056V^{1.28}$

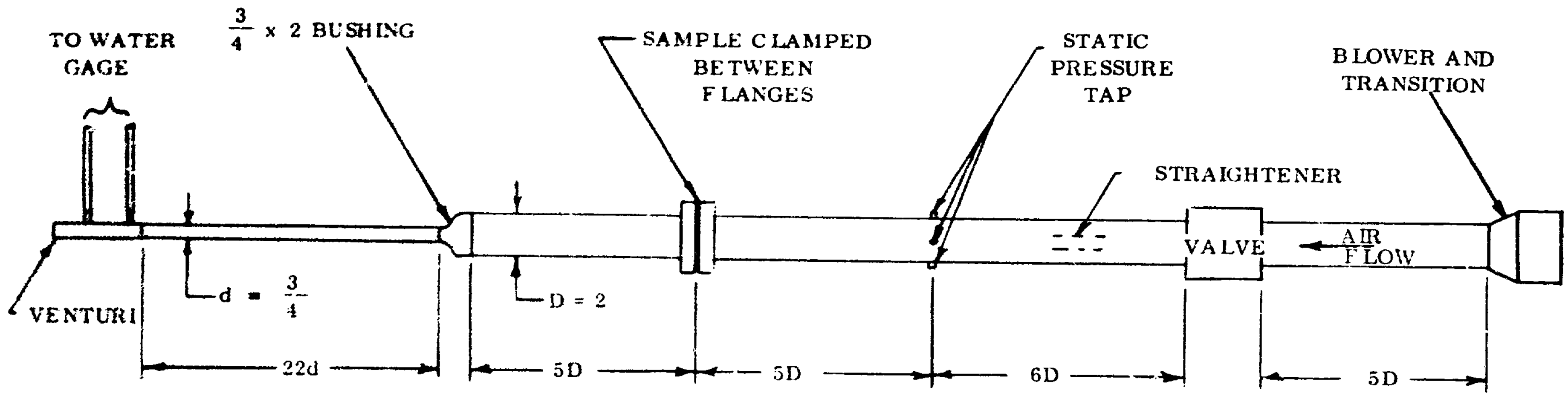



Figure 23. Schematic Drawing of Pressure Drop Test Setup

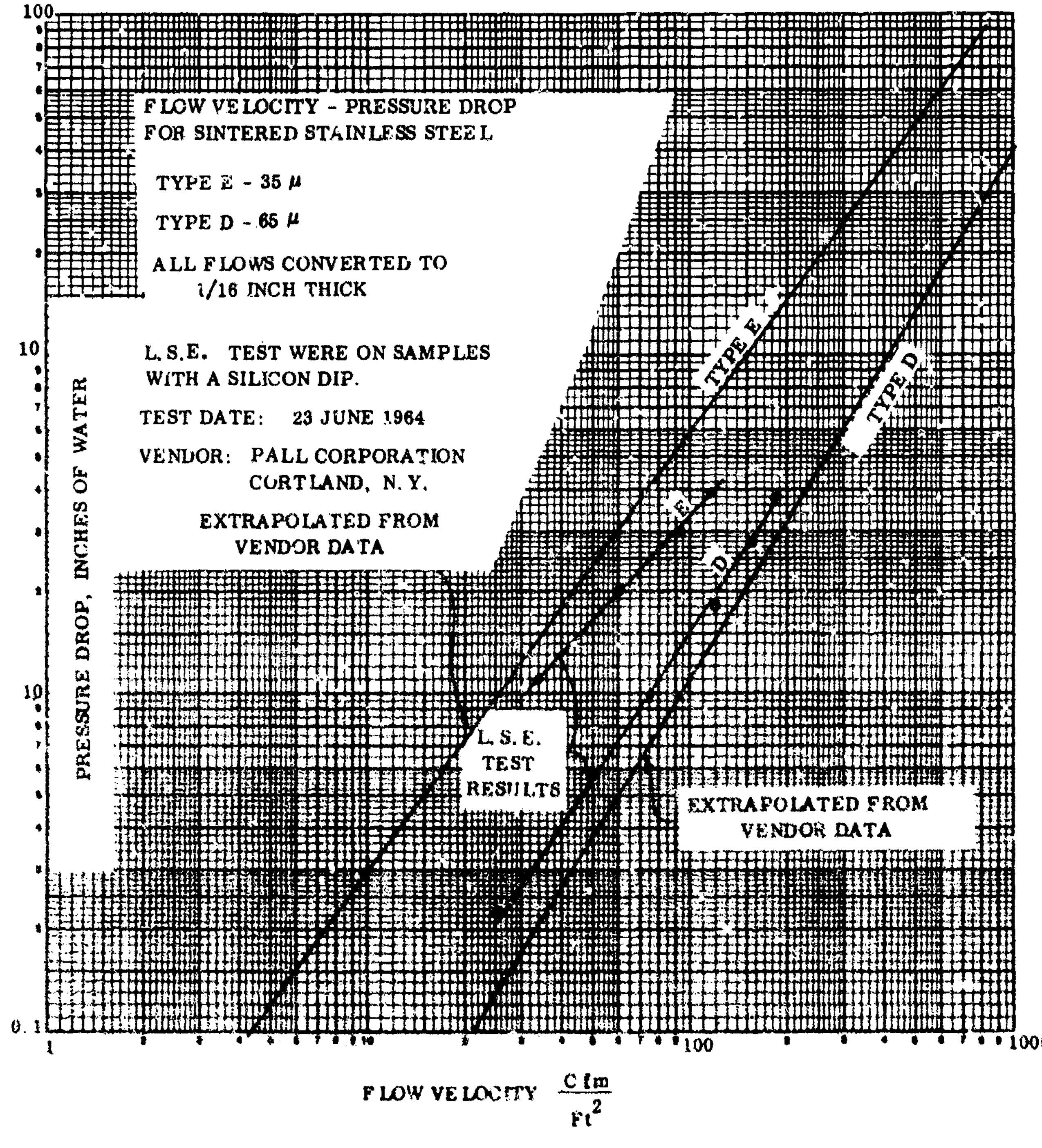



Figure 24. Velocity-Pressure Drop Curves for Sintered Stainless Steel

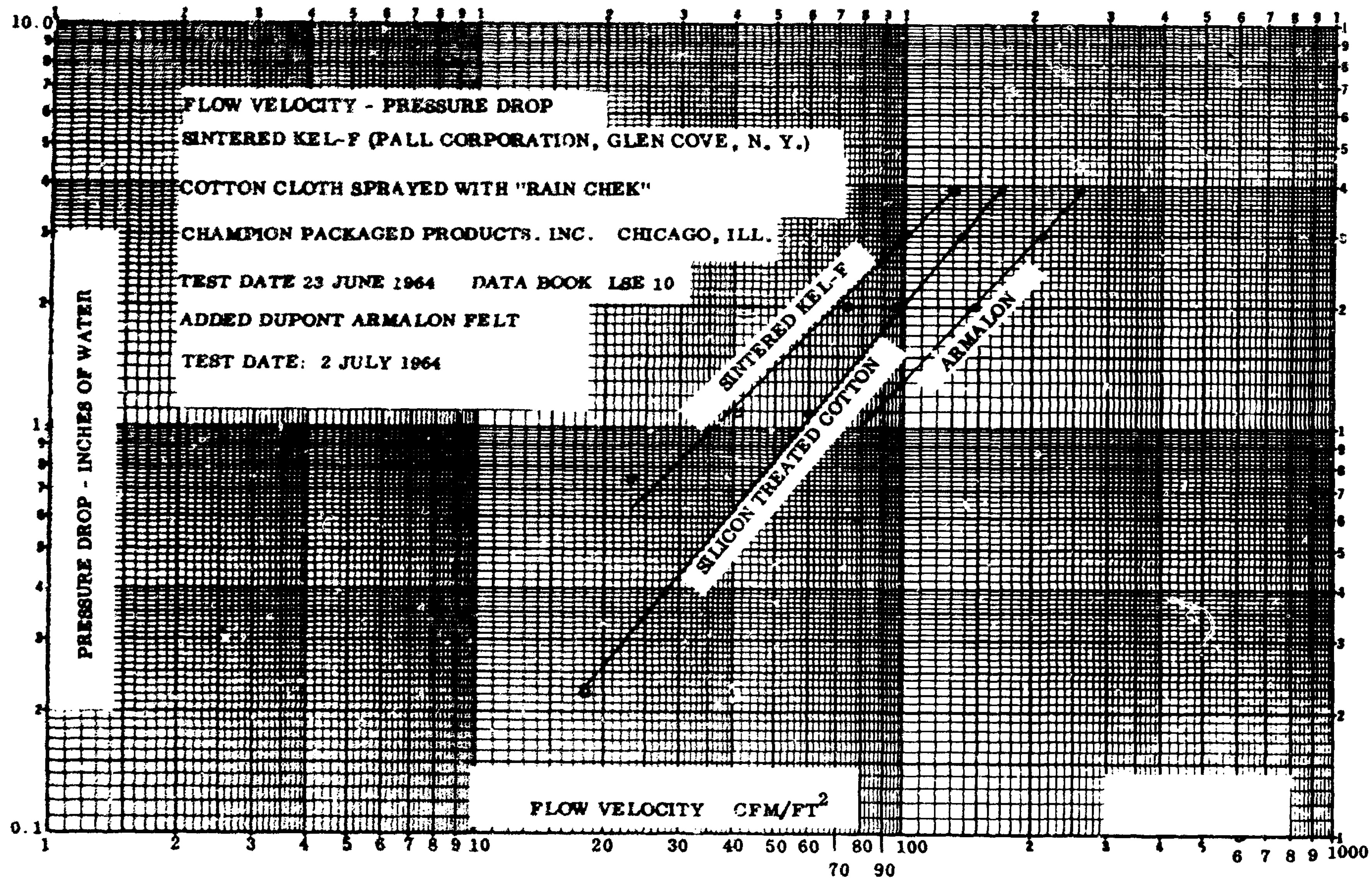



Figure 25. Velocity-Pressure Drop Curves for Sintered Kel-F and Coated Fabric

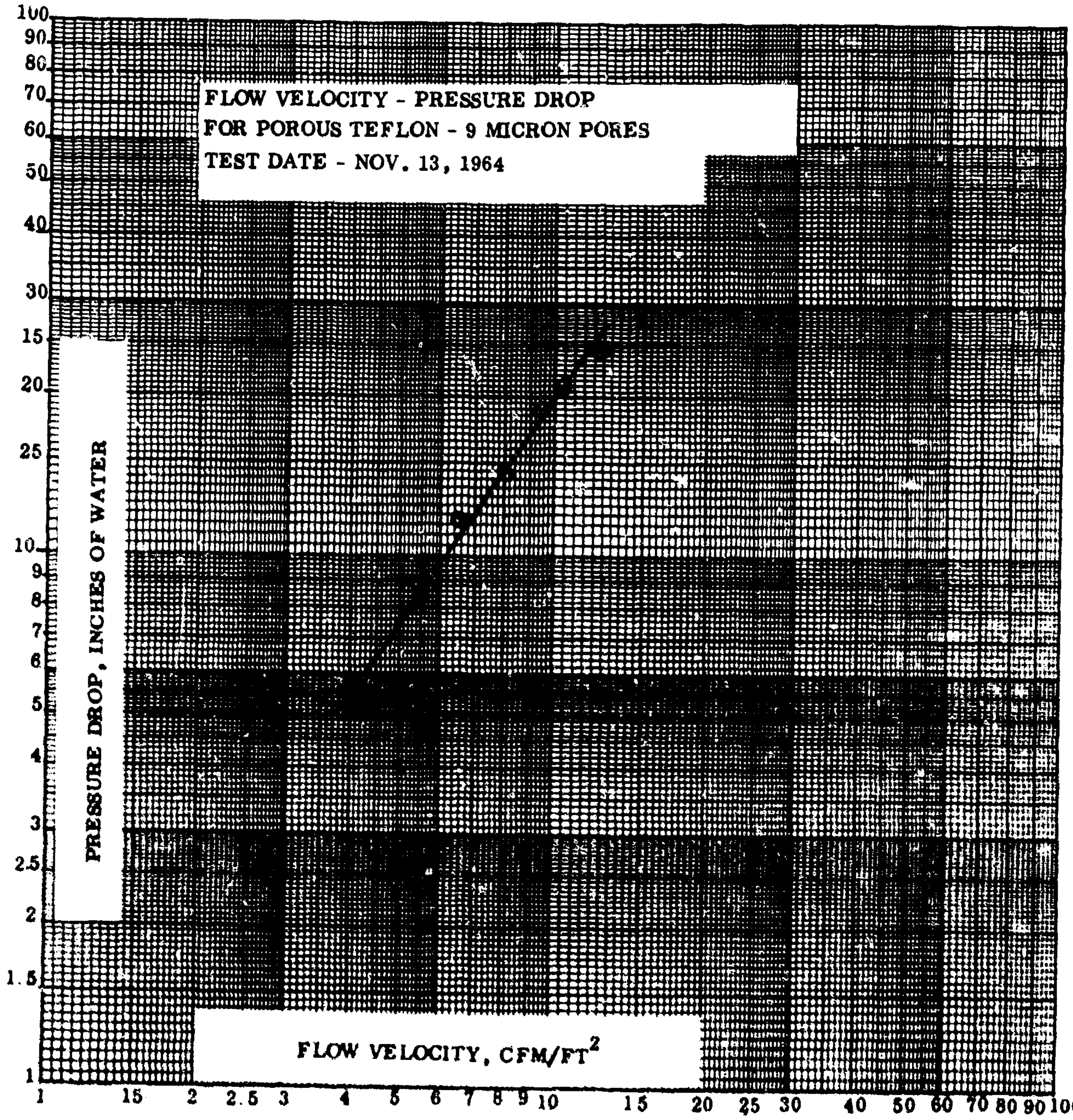



Figure 26. Velocity-Pressure Drop Curve for Porous Teflon

- Teflon felt (DuPont Armalon) $\Delta p = 0.007V^{1.138}$
- Sintered Teflon 9μ $\Delta p = 0.78V^{1.4}$

where $\Delta p$ is the pressure drop, inches of water

$V$ is approach velocity, cfm/ft$^2$.

A trade off of material properties was made before selecting a membrane for the water droplet removal concept. The stainless steels have the highest strength, but also the highest weight; hence, for a "flight type" design, the nonmetallics would be preferable. In addition, the nonmetallics have a lower thermal conductivity than the steels and should provide less change of unwanted condensation. Of the nonmetallics, Kel-F has the best stiffness; hence, Kel-F was the material choice.

For the cone collector assembly, Figure 4, the net exposed area of the Kel-F was computed to be 15.34 square inches or 0.1068 ft$^2$. At a flow of 2.35 cfm, $V = 22$ feet/min and $\Delta p = 0.61$ inch of water. In the actual test, with dry nitrogen only, the measured pressure drop was 0.9 inch of water. As reported in the water droplet section, the stable pressure with droplets flowing was 11.6 inches of water. There is, therefore, correlation with the dry membrane only.

## 2. MATERIALS PERMEABLE TO WATER VAPOR

The material chosen for the vapor-compression and vapor freeze-out tests was cellulose acetate. This was because of the high selectivity to passing water vapor. DuPont CA-148 was the particular type chosen because it had the highest water permeability of the various references. See Tables XV and XVI for properties of cellulose acetate and other film materials.

TABLE XV. PERMEABILITY OF CELLULOSE ACETATE SHEET 0.001 INCH THICK

| MATERIAL | WATER VAPOR | $CO_2$ | $N_2$ | $O_2$ |
|---|---|---|---|---|
| Eastman Kodak Kodacel A-30 | 1000 | 0.53 | 0.018 | 0.090 |
| DuPont CA-148 | 5200 | 0.60 | 0.024 | 0.07 |
| Tappi Monograph No. 23 (Reference 2) | 550 | --- | 0.028 | 0.076 |
| Industrial and Eng. Chemistry (Reference 3) | 1500 | -- | -- | -- |
| Units Are | $\frac{(\text{Std cc})}{(\text{sec})} \times \frac{(\text{cm thick})}{(\text{cm}^2\ \text{area})} \times \frac{10^{-9}}{(\text{cm Hg})}$ | | | |

TABLE XVI. PERMEABILITY OF VARIOUS FILM MATERIALS

| MATERIAL | WATER VAPOR | $CO_2$ | $N_2$ | $O_2$ |
|---|---|---|---|---|
| Ethyl Cellulose (Dow Chemical Co.) | 1100 | 3.08 | 0.36 | 1.2 |
| Ethyl Cellulose (Reference 2) | 1300 | 20.0 | 0.84 | 2.65 |
| PVA (Polyvinyl Alcohol) | 3200 | 0.012 | 0.01 | 0.01 |
| Lexan (Polycarbonate) | 100 | 1.0 | 0.02 | 0.1 |
| Dimethyl Silicone Rubber | 3800 | 325.0 | 28 | 60 |
| Polyethylene | 7.6 | 46.0 | 0.19 | 7.2 |
| Natural Rubber | 2800* | 131.0 | 8.08 | 23.3 |

Units Are $\frac{(\text{Std cc})}{(\text{sec})} \times \frac{(\text{cm thick})}{(\text{cm area})} \times \frac{(10^{-9})}{(\text{cm Hg})}$

*Extrapolated from References 2 and 4.

To have a minimum area, the permeation of the film should be maximum. Of the materials in Table XV or XVI, CA-148 has the highest permeability rate to water; however, there are other high permeability materials. PVA not only has high permeability, but also has extreme selectivity; that is, it has the lowest ratio of gas to water vapor permeability for the gases of a cabin atmosphere, $O_2$, $N_2$ and $CO_2$. Unfortunately, PVA readily dissolves in water, and hence cannot be used. Indeed, there is a slow, but unknown rate of hydrolization for cellulose acetate, hence, ethyl cellulose was considered as a possible back-up material (although it was not tested in this program). Silicone rubber was the next high-permeability material investigated. It was not used because of the poor selectivity. Similarly, natural rubber was rejected. To illustrate, assume that an atmosphere exists with the following conditions:

| | |
|---|---|
| Nitrogen | 58.1 cm Hg pressure |
| Oxygen | 15.4 cm Hg |
| Carbon Dioxide | 0.3 cm Hg |
| Water Vapor | 2.2 cm Hg |
| | 76.0 cm Hg Total |

The membrane area will be taken as 8000 $cm^2$ of one-mil thickness, similar to the unit actually tested.

The computed flows are:

| GAS | CELLULOSE ACETATE (CA-148) Gas Flow Rate cc/sec | | SILICONE RUBBER Gas Flow Rate cc/sec | |
|---|---|---|---|---|
| $N_2$ | 0.0044 | | 5.1 | |
| $O_2$ | 0.0034 | | 2.9 | |
| $CO_2$ | 0.0006 | | 0.3 | |
| $H_2O$ | -- | 36.0 | -- | 26.3 |
| Total Gases | 0.0084 $\frac{cc}{sec}$ | | 8.3 $\frac{cc}{sec}$ | |

Observe that silicone rubber would pass 8.3 cc/sec out of a total of 34.6 cc/sec or 24%. The cellulose acetate would pass only 0.023% of noncondensable gases. Silicone rubber assemblies would need several stages in order to approach the selectivity of cellulose acetate; hence, for this contract, cellulose acetate was chosen, even though hydrolization was a potential hazard.

To check the strength of the film, a sample of acetate was placed in the fixture of Figure 27. The fixture has a slot opening greater than the unsupported edge in an actual assembly. A vacuum of -29 inches of mercury was pulled. There was deformation of the membrane, but no tearing, so cellulose acetate was considered strong enough for the effort in this contract.

Samples of 0.001 and 0.002 cellulose acetate film were soaked in a beaker of water. At various time periods up to 30 days, samples of each thickness were removed from the beaker and tested in the fixture of Figure 27 to check the vacuum-sealing capabilities. No leaks developed. In fact, the materials actually seemed to become stronger. The conclusion, therefore, is that hydrolization for a 30-day mission should not be a problem.

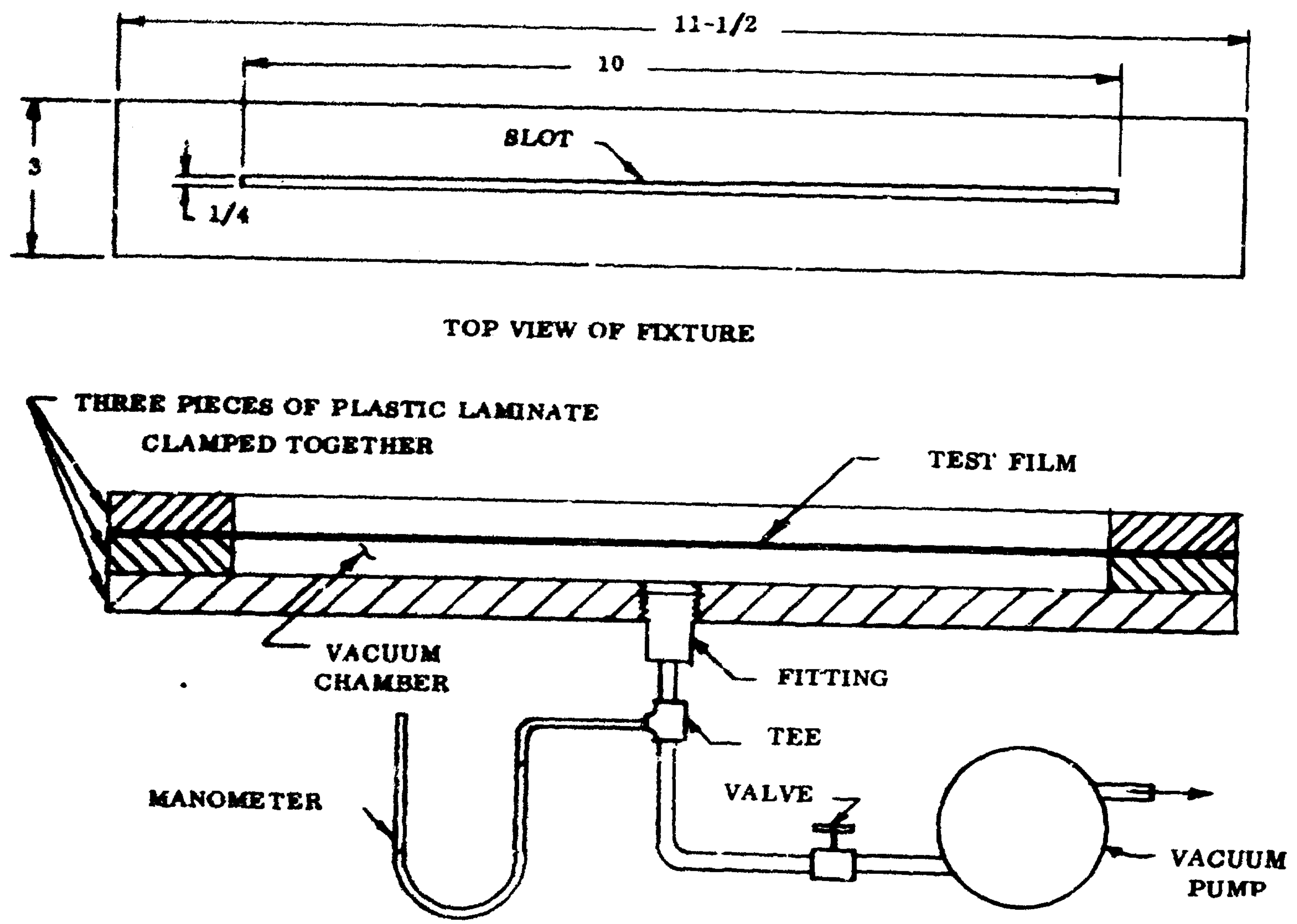



Figure 27. Schematic Drawing of Vacuum Test on Cellulose Acetate Film

## APPENDIX I

### COMPRESSOR CHOICE AND POWER CALCULATIONS

The following companies or their representatives were contacted to see if they could provide a compressor or pump that could evacuate to a pressure of 5 to 25 mm, while pumping 0.0013 lb/min of water vapor.

| | | |
|---|---|---|
| 1. | Binks Manufacturing Co. | Chicago, Ill. |
| 2. | Bell and Gossett | Morton Grove, Ill. |
| 3. | Nash Engineering | South Norwalk, Conn. |
| 4. | Gelman Instruments | Ann Arbor, Mich. |
| 5. | Chicago Pneumatic Tool Co. | New York, N.Y. |
| 6. | Eastern Industries | Hamden, Conn. |
| 7. | The Kraisal Co. | Hackensack, N.J. |
| 8. | Gast Manufacturing Co. | Benton Harbor, Mich. |
| 9. | Great Lakes | Cleveland, Ohio |
| 10. | Leiman Brothers, Inc. | East Rutherford, N.J. |
| 11. | Kinney Vacuum | Camden, N.J. |
| 12. | Fairchild Stratos | Bay Shore, N.Y. |
| 13. | Worthington Pump | Bala Cynwyd, Pa. (Office) |
| 14. | Pressure Products, Inc. | Hatboro, Penna. |
| 15. | Tecumseh Compressors | Smithtown, N.Y. (Office) |
| 16. | Dunham Busch | West Hartford, Conn. |

Of these companies, only two felt confident that the requirements could be met. One company, Pressure Products, had made similar pumps, but they were very big, heavy, and cost $5000 minimum. The most optimistic company was Leiman Brothers. They believed that their Model 295 - 2 x 3 two-stage oilless pump with slight modification would suffice.

On a visit to Leiman Brothers, June 24, 1964, a model 295 - 2 x 3 unit was under test (for a different customer). This pump was able to draw a vacuum of -28 inches of mercury gage when discharging to the atmosphere. By covering one of the weep holes (a small hole for cooling air near the bearings) the vacuum drawn was -29.5 inches of mercury. Covering the weep holes, however, might cause overheating during steady state operation. This was to be checked by the vendor and the weep holes are needed. Tests at GE confirm the need for weep holes.

A short test was run on a Leiman pump in the GE Life Support Engineering Laboratory. Model 295-2, (which is only a single stage pump) was available in the laboratory with a one horsepower motor. This unit pulled a vacuum of -26 inches of mercury gage. When weep holes were blocked, a vacuum of -27.5 inches of mercury gage was measured. Reducing the outlet pressure to -26.5 inches gag. with a Welch Duo-Seal vacuum pump enabled the Leiman pump to pull -29.5 inches of mercury gage; however, the Leiman pump began heating rapidly (see Figure I-1 for test schematic).

With these tests in mind, there was reasonable assurance that the Leiman pump would work although weep holes could cause troubles. The Leiman Company supplied a curve to show the capacity of their pump (see Figure I-2).

1. CALCULATION OF COMPRESSOR POWER

The ideal theoretical work of a compressor is (Reference 6):

$$W = p_1 v_1 \left(\frac{k}{k-1}\right) \left[\left(\frac{p2}{p1}\right)^{\frac{k-1}{k}} - 1\right]$$

Where W is work in ft lb/min done on the "fluid."

$p_1$ is compressor inlet pressure, lb/ft$^2$ absolute

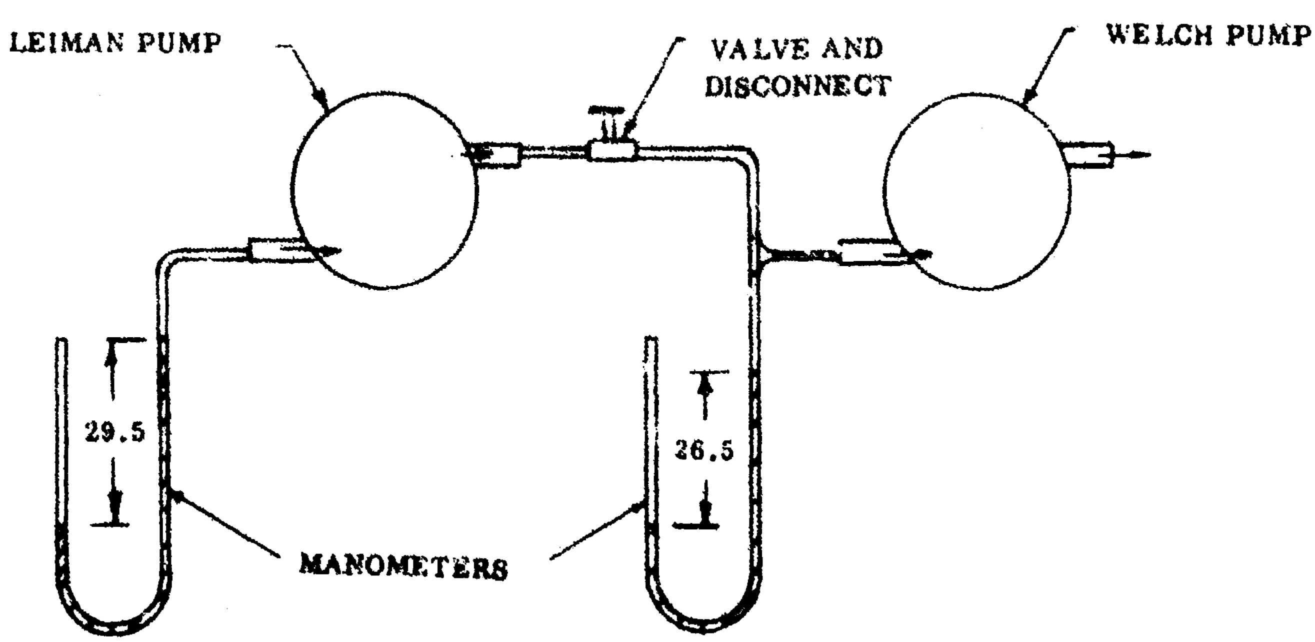



Figure I-1. Schematic Drawing of Pump Test Setup

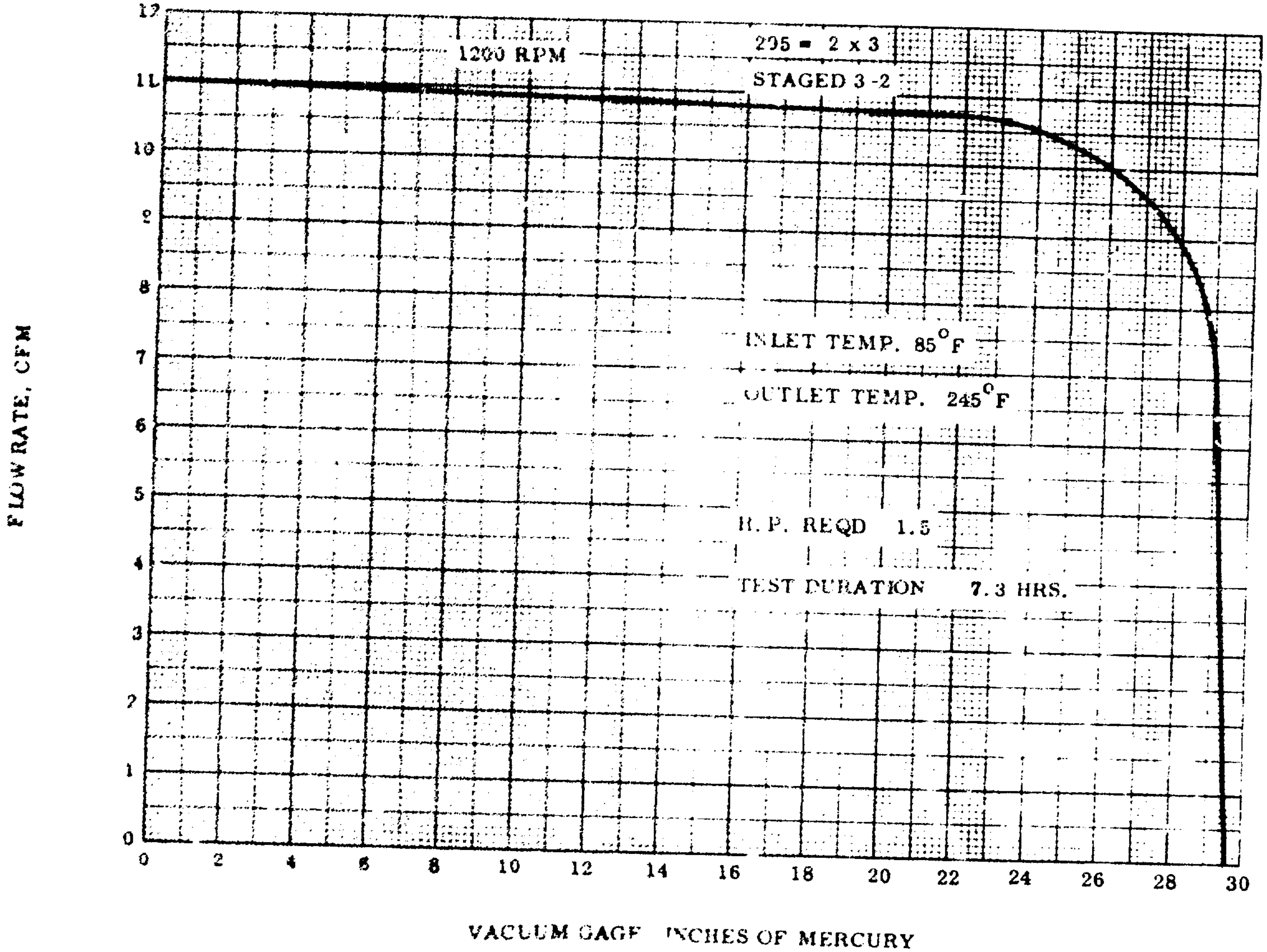



Figure I-2. Pressure-Volume Curve of Leiman Pump Model 295-2x3 Staged 3-2.

$p_2$ is compressor outlet pressure, $lb/ft^2$ absolute

$v_1$ volume flow $ft^3/min$

k is the ratio of specific heats

From Section IV and Figure 8.

$p_1$ = 1.5 cm Hg = 0.59 in. Hg = 0.29 psia = 41.8 psfa

$p_2$ = 7.5 cm Hg = 2.95 in. Hg = 1.45 psia = 209 psfa

k = 1.35 for steam

$$\frac{k-1}{k} = 0.260$$

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \left(\frac{209}{41.8}\right)^{.260} = 1.70$$

$$\frac{k}{k-1} = \frac{1}{0.260} = 3.84$$

To determine $v_1$, the weight flow desired is $2.5 \frac{lb}{day}$

$$2.5 \frac{lb}{day} \div 1440 \frac{min}{day} = 0.001735 \frac{lb}{min}$$

Assume that the temperature at the compressor inlet is the mean duct temperature of 90°F. The specific volume at 90°F, saturated is 468.4 $ft^3/lb$ at a pressure of 0.698 psia.

$$v_1 = (.001735)(468.4)\frac{(0.698)}{0.290} = 1.96 \frac{ft^3}{min}$$

$$W = (41.8)(1.96)(3.84)(1.70 - 1) = 220 \frac{ft\ lb}{min}$$

$$W = \frac{220\ ft\ lb/min}{33000 \frac{ft\ lb/min}{HP}} \times 746 \frac{Watts}{HP} = 4.98\ watts = 5\ watts$$

For the test conditions recorded in Table X,

$$p_1 = 3.50 \text{ cm Hg} = 1.38'' \text{ Hg} = 0.676 \text{ psia} = 97.5 \text{ psfa}$$

$$p_2 = \text{Barometer reading} = 30.08 \text{ in Hg} = 14.8 \text{ psia} = 2130 \text{ psfa}$$

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \left(\frac{2130}{97.5}\right)^{.260} = 2.23$$

Assume that the temperature at the compressor inlet is the average of the steam duct inlet and outlet temperatures; thus T = 118°F. Specific volume is 214.5 ft$^3$/lb at 1.600 psia and 118°F. The measured flow was 13 grams in 35 minutes = 0.000818 lb/min.

$$v_1 = (0.000818)\ (214.5) \left(\frac{1.600}{.676}\right) = 0.415\ \frac{ft^3}{min}$$

$$W = (97.5)\ (0.415)\ (3.84)\ (2.23-1) = 191 \text{ ft lb/min}$$

$$W = \frac{191 \times 746}{33000} = 4.32 \text{ watts}$$

For the typical weep hole air flow of 33 l/min at STP and the condition of Test Er, Table IX (see paragraph IV.b.4)

$$p_1 = 0.610 \text{ psia} = 87.9 \text{ psfa}$$

$$p_2 = 8.58 \text{ psia} = 2180 \text{ psfa}$$

$$v_1 = \frac{33 \text{ l/min}}{28.3 \text{ l/ft}^3} \times \frac{14.7 \text{ psia}}{0.61 \text{ psia}} = 28.1\ \frac{ft^3}{min}$$

For air, k = 1.4

$$k-1/k = 0.286$$

$$\frac{k}{k-1} = 3.50$$

$$\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} = \left(\frac{2180}{87.9}\right)^{.286} = 2.13$$

$$W = (87.9)\,(28.1)\,(3.5)\,(2.13-1) = 9750\,\frac{\text{ft lb}}{\text{min}}$$

$$w = (9750)\,\frac{(746)}{33000} = 220 \text{ Watts}$$

## APPENDIX II

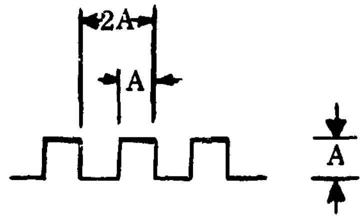
## CORRUGATED ALUMINUM SHEET FOR MEMBRANE SUPPORT

In order to support a thin film in a folded position, the need for a corrugated shape was determined. Various corrugated shapes were made out of 0.005 in. aluminum and tested. For example, Figure II-1 shows several shapes in profile. The corrugation had to be strong enough to support 14.7 psia, (a full atmosphere) without deformation; thus, the membrane would be supported even if the compressor pulled a vacuum to 5 mm Hg absolute.

The pressure loading was simulated by adding weights to two samples acting as supports for a metal beam. The total weight, including beam, was divided by the projected area of support. The square bend of 1/8 in. x 1/8 in. or the sharp 1/8 in. high sawtooth with a vertical bend were found to be satisfactory. Since the sawtooth corrugation offered less contact to the membrane and more open area for flow, it was the first choice; however, in order to prevent nesting of the corrugations, it was decided to have the corrugations run on an angle, as in Figure II-2, and then cross stack the corrugations. An inquiry reply from the vendor showed the cost and delivery time of this design were prohibitive; whereas a corrugation similar to the second choice of a 1/8 x 1/8 square was readily available. To prevent nesting, the corrugation is not straight but staggered (see Figure II-3). By changing to the modified square, cost and delivery were favorable.

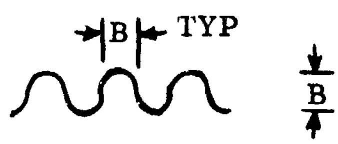
The vendor also stated that corrugating the tempered aluminum alloy 1145H19 was too difficult. A soft alloy such as 3003-0 was needed; hence, the thickness was increased from 0.005 to 0.010 to compensate for the lack of temper. When the corrugation was received, it was tested for strength and found to be more than adequate.

From previous work independent of this contract (Appendix III), it was found that the use of filter paper between a membrane and its support tends to aid the flow of vapor by giving additional flow area. The vapor has room to pass through the filter media instead of being blocked by a bare support corrugation or screen. The use of filter paper would also reduce the likelihood of the membrane tearing as it was wound over the edge of a corrugation support. The result was the design of this contract (see Figures 9 through 12).

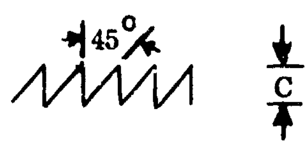
a. 


WITH A = 1/4, CORRUGATION DISTORTS AT 7.2 PSI

WITH A = 1/8 CORRUGATION SPREADS SLIGHTLY AT 15.1 PSI

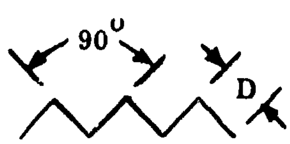
b. 


WITH B = 1/4 CORRUGATION COLLAPSED AT 2.88 PSI

c. 


WITH C = 3/16 CORRUGATION SUPPORTED 12.2 PSI FAILED AT 16 PSI

C = 1/8 SUPPORT AT 21.7 PSI

d. 


WITH D = 1/8 CORRUGATION SPREAD AT 25 PSI

MATERIAL: ALUMINUM ALLOY 1145 H 19

THICKNESS: 0.005

Figure II-1. Sections of Corrugations Tested for Membrane Support

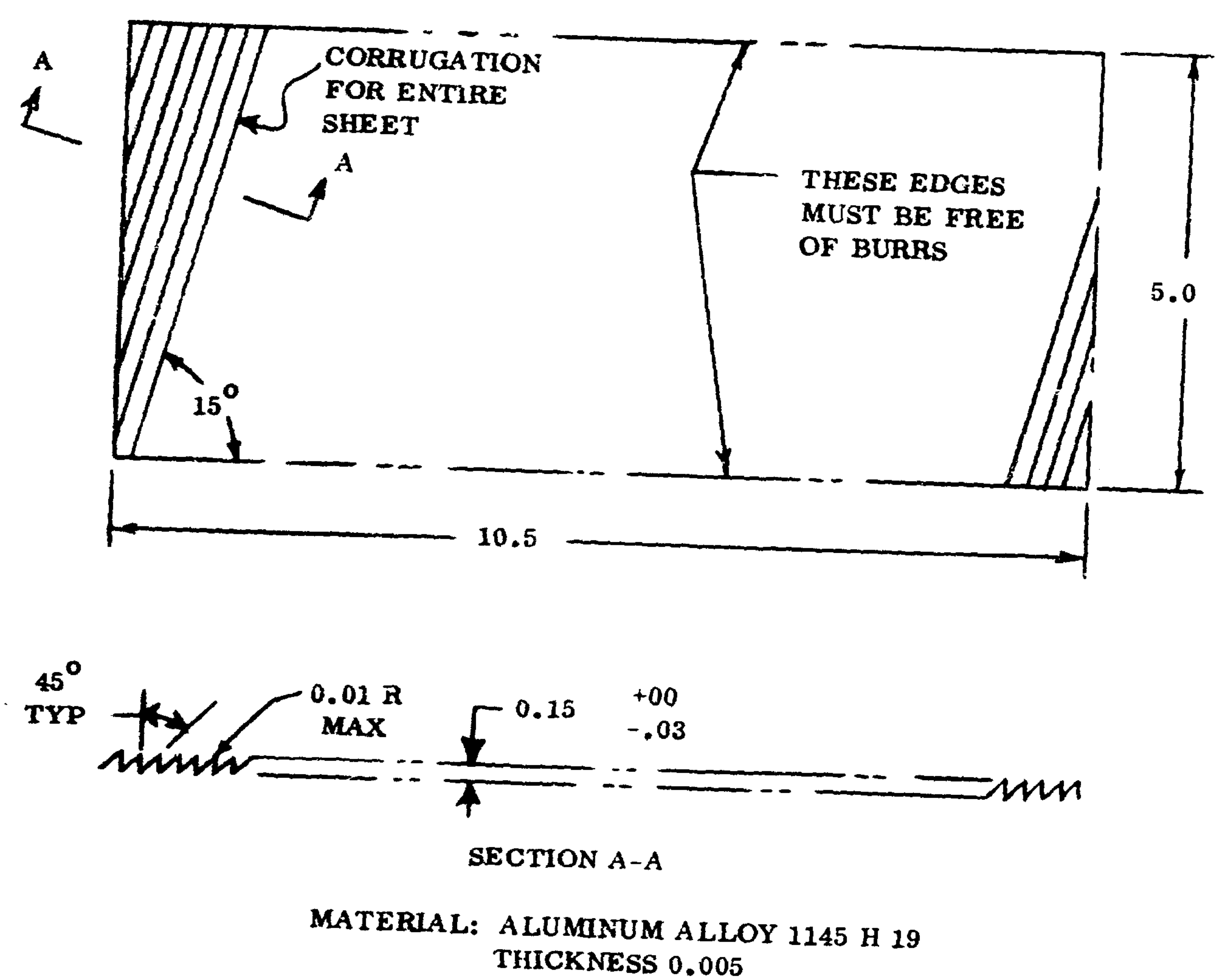
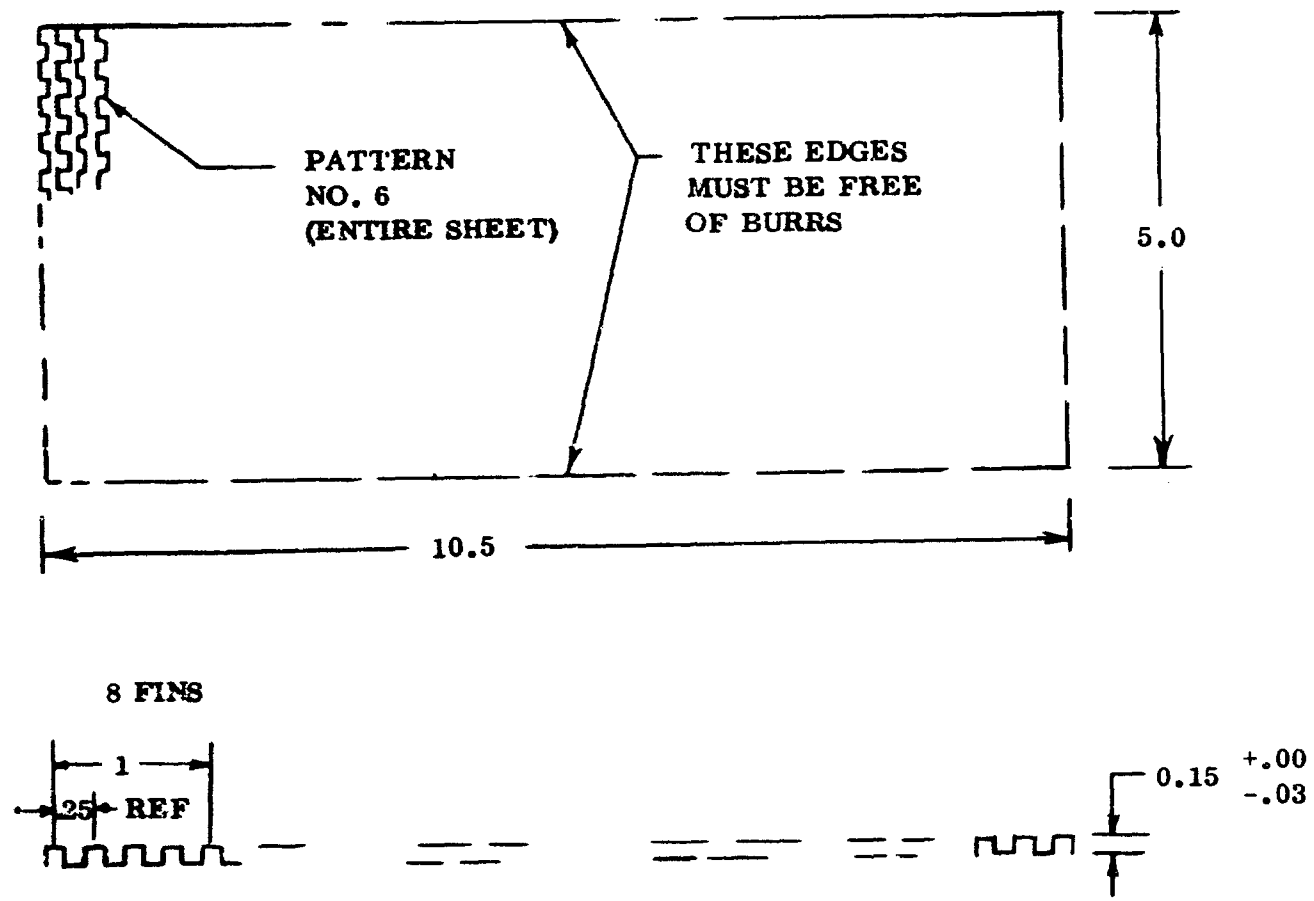



Figure II-2. Sawtooth Corrugation for Membrane Support




MATERIAL: ALUMINUM ALLOY 3003-0
THICKNESS 0.010

Figure II-3. Staggered Square Corrugation for Membrane Support

## APPENDIX III

## EFFECT OF MEMBRANE SUPPORT STRUCTURE ON WATER VAPOR PERMEATION THROUGH A MEMBRANE

The effort reported in this appendix was performed independently of contract AF 33(615)-1475; however, since the results were related to the design effort of this contract, the description of the test is reported.

The purpose of the work was to determine the effect of support structure on the rate of water vapor permeation through a membrane. Figures III-1, III-2 and III-3 show the test schematic, support details, and a photograph of the assembly. The membrane tested was cellulose acetate film (DuPont CA-148) 0.001 thick. This is the same material used in the membrane assembly detailed in Figure 9.

Four different support structures were tested. They were (1) a fine mesh stainless steel screen; (2) a coarse mesh screen; (3) a sintered stainless disk; (4) and a coarse screen with filter paper between the screen and membrane.

To run the test, a burette was filled with water and the assembly was installed in the bath. The flask was evacuated creating a difference in pressure across the membrane, resulting in permeation of water through the membrane into the flask. By measuring the change in water volume at the burette for a given test time, the flow rate can be computed. Then by equation IV-6, repeated here in rearranged form, the permeability can be computed.

$$P = \frac{Qt}{A \Delta p} \qquad \text{(IV-6) repeated}$$

For the pressure gradient term $\Delta p$, the vapor pressure of water at the temperature of the flask was subtracted from the vapor pressure of water at the burette temperature. Since the amount of water that can be forced through the membrane by the hydrostatic pressure of one atmosphere is negligible, the vapor pressure at the burette temperature was used even though the pressure of the burette is nominally one atmosphere.

The details of support characteristics, and the results of the tests are given in Table III-1. The amount permeated through the membrane varies from $400 \times 10^{-9}$ to $5100 \times 10^{-9}$. Since the same type of membrane was used, in every test, the change in permeability must have been because of the change in support structure.

Figures III-4 and III-5 are photographs of the membrane taken at the end of the tests. The membrane was "dimpled" with the pattern of the screen, indicating that the change in permeability rate was due entirely to a reduction in effective area of the membrane. This is further substantiated by the coarse screen-filter paper

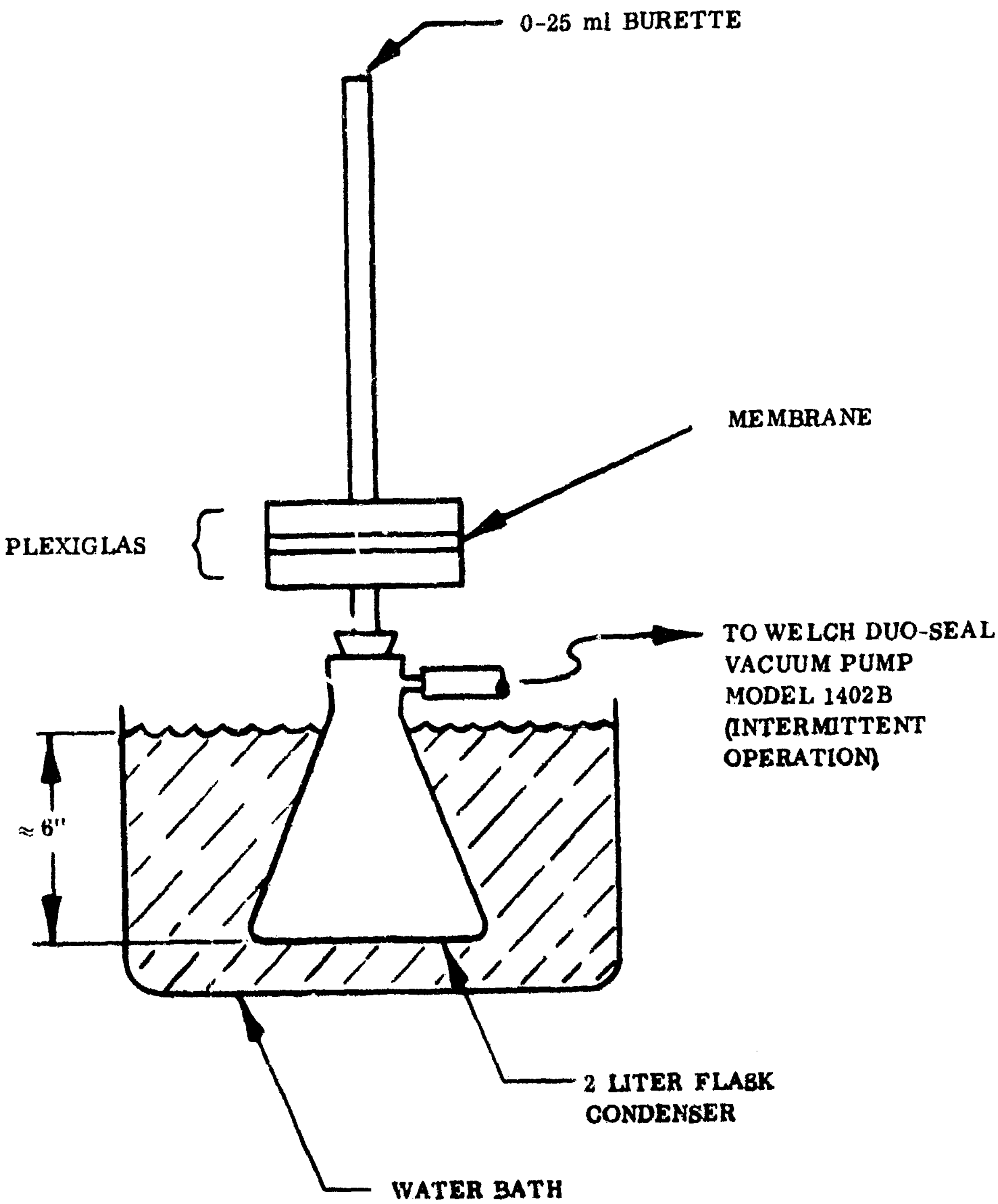



**Figure III-1. Test Setup - Effect of Membrane Structure on Water Vapor Permeation**

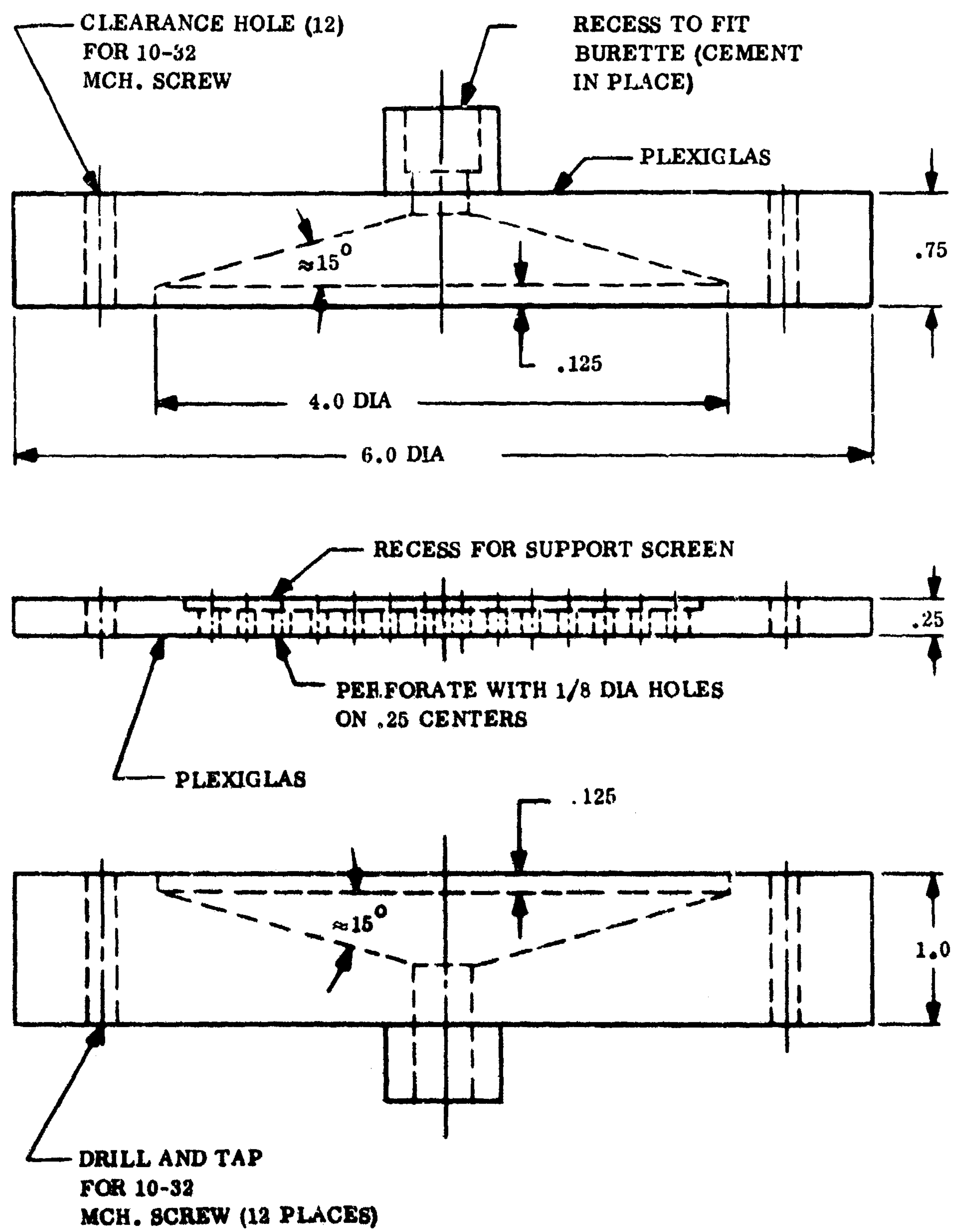



Figure III-2. Test Setup - Membrane Holder Detail support which gave a permeability constant in agreement with DuPont data for the unsupported acetate film (5000 x $10^{-9}$).

An additional test was performed using the coarse screen-filter paper support wherein the hydrostatic head across the membrane was reduced to about 12 inches of water (by not evacuating the condenser). A surprisingly low permeability constant was obtained for this condition i.e. $P = 7.5 \times 10^{-9}$. Whether this low constant was due to the effect of the reduction in hydrostatic pressure or the impeding action of the residual air within the condenser was not determined.

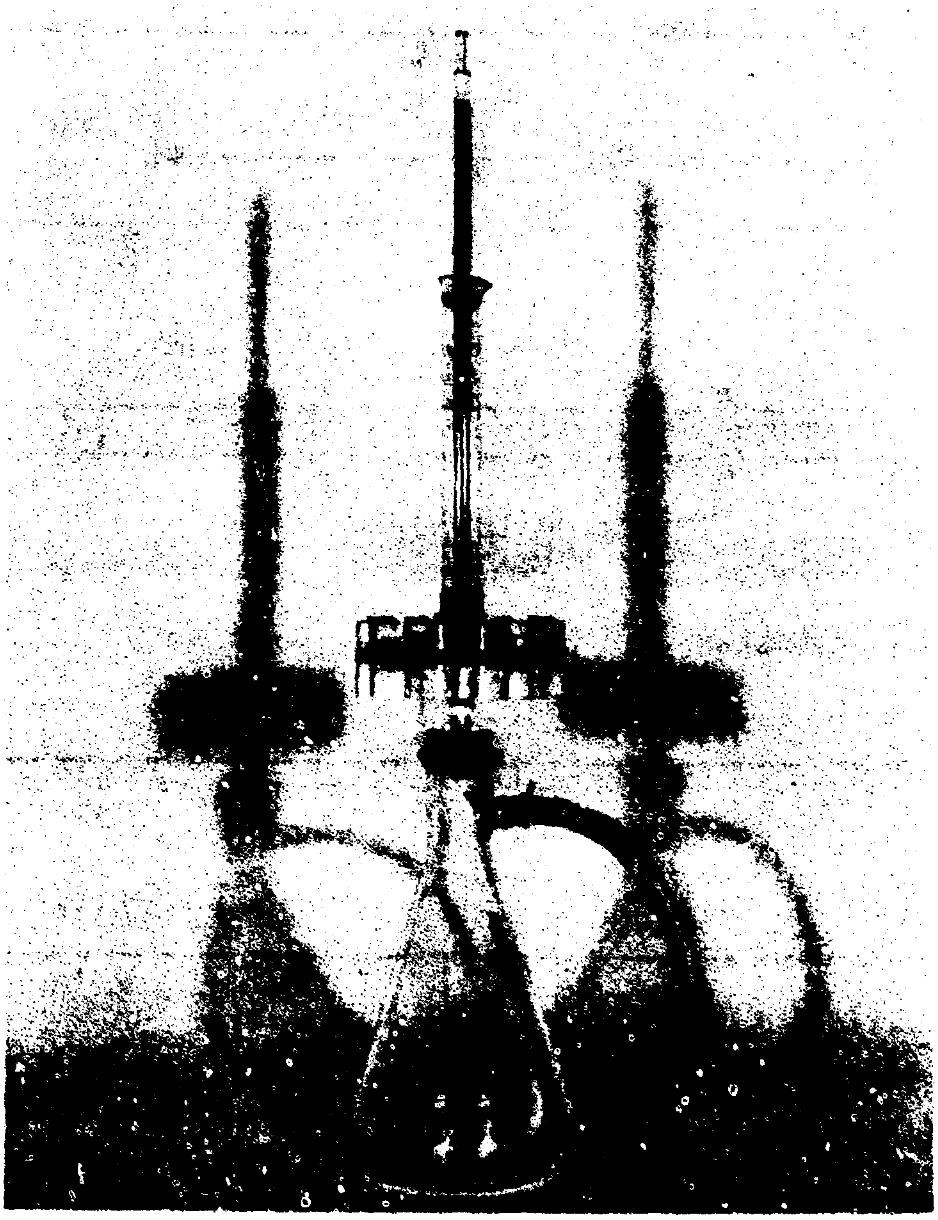


**Figure III-3. Flask and Membrane Assembly**

TABLE III-1. WATER VAPOR PERMEABILITY OF ONE MIL CELLULOSE ACETATE FILM

| Test No. | Test Configuration | Avg Flow gms/hr | Equivalent Differential Pressure cm Hg (1) | Permeability Constant (2) |
|---|---|---|---|---|
| 1 | Fine screen (3) | 0.0475 | 1.37 | $400 \times 10^{-9}$ |
| 2 | Coarse screen (4) | 0.136 | 1.15 | $1280 \times 10^{-9}$ |
| 3 | Sintered disc (5) | 0.128 | 1.24 | $1130 \times 10^{-9}$ |
| 4 | Coarse screen plus filter paper (6) | 0.64 | 1.37 | $5100 \times 10^{-9}$ |

(1) Based on vapor pressure at the equivalent up and downstream temperatures

(2) $\dfrac{\text{(std cc) (cm)}}{\text{(sec) (cm}^2\text{) (cm Hg}\Delta\text{p)}}$

(3) 84 mesh stainless steel

(4) 0.028 dia wire; square weave; 17 strands/inch

(5) Type H, 5 micron pore size, 0.062 in. thick

(6) Qualitative Filter Paper, No. 5160, Arthur H. Thomas Company

NOTE: In the membrane assembly of this contract (see Figure 9), the filter paper used was No. 5161. This is the identical grade of paper as No. 5160 except that it is available in sheets 480 mm square.

Figure III-4. Membrane From Test No. 2

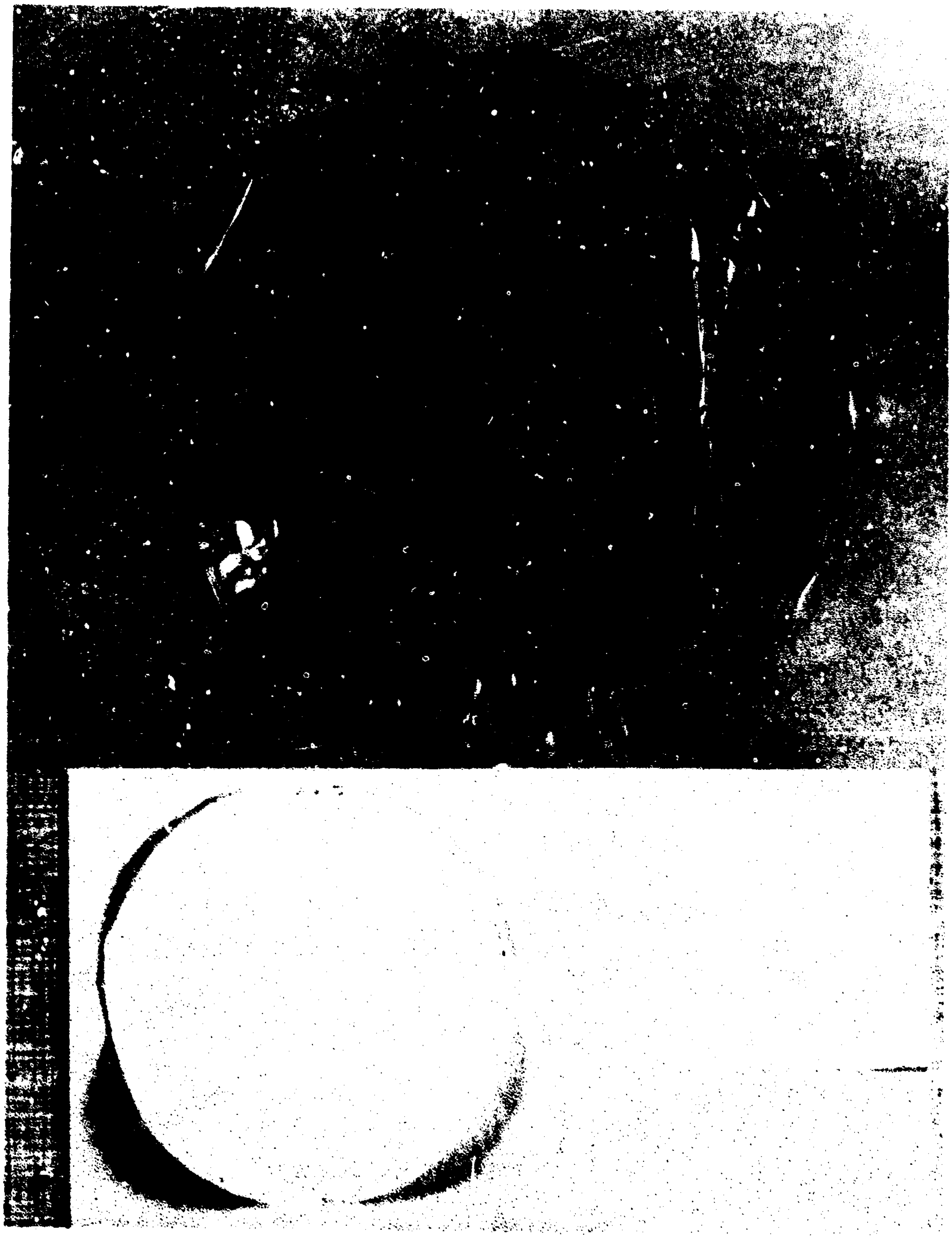


Figure III-5. Membrane and Filter Paper (Placed Between Coarse Screen and Membrane) from Test No. 4

## APPENDIX IV

## SAMPLE CALCULATIONS

### 1. WATER DROPLET REMOVAL CONCEPT - DATA FOR TABLES

A set of sample calculations will be presented here suitable for explaining Tables I, II, and III. Test No. 2 will be used as the sample.

a) Determine Flow of Air, Mositure

From Figure IV-1, a flow of 2.35 cfm was read for a venturi pressure differential of 1.0 inch of water. The basic equation of ideal velocity through a venturi (approximate form) is given in Reference 6 as:

$$V_{s2} = \left[ 2\, g\, RT_1 \frac{(p_1 - p_2)}{p_1} \right]^{1/2} \qquad \text{(A. IV-1)}$$

where $V_{s2}$ = velocity, throat of venturi ft/sec

$g$ = acceleration of gravity ft/sec$^2$

$R$ = specific gas constant ft/$^{\circ}$R

$T_1$ = temperature venturi inlet degrees $^{\circ}$R

$p_1$ = venturi inlet pressure, psfa

$p_2$ = throat pressure, psfa

From equation A. IV-1 above, it is seen that for a given gas, temperature, and pressure, the change in velocity by varying temperature is:

$$\frac{V_{s2}}{V_{s2}'} = \left[ \frac{T_1}{T_1'} \right]^{1/2} \qquad \text{(A. IV-2)}$$

Similarly, as the gas varies, all other terms being constant,

$$\frac{V_{s2}}{V_{s2}'} = \left[ \frac{R}{R'} \right]^{1/2} \qquad \text{(A. IV-3)}$$

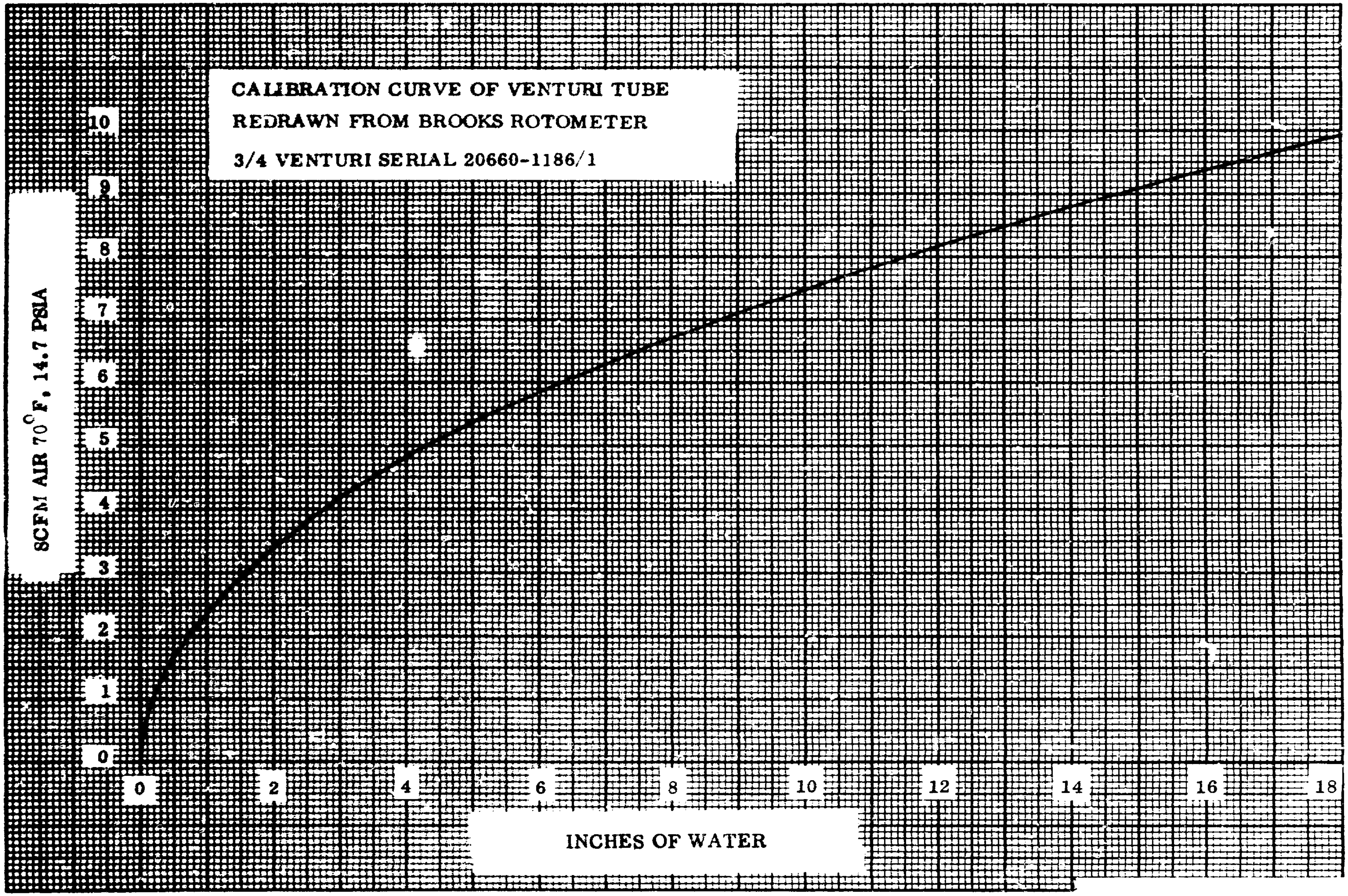



Figure IV-1. Calibration Curve of Venturi Tube

A mixture of a vapor and a gas results in a new gas constant, R' such that

$$R' = \frac{W_v R_v + W_n R_n}{W_m} \qquad \text{(A. IV-4)}$$

where $R'$ = resultant gas constant ft/$^{o}$R

$R_n$ and $R_v$ = gas constants for gas-vapor constituent

$W_n$, $W_v$ = weight of individual gas-vapor constituents

$W_m$ = weight of mixture.

And for a mixture of nitrogen and water vapor

$$W_s = \frac{p_v R_n}{p_n R_v} \qquad \text{(A. IV-5)}$$

where $W_s$ = humidity ratio $lbH_2O/lb\ N_2$

$p_v$ = partial pressure of the vapor

$p_n$ = partial pressure of nitrogen

$R_n$ = gas constant nitrogen

$R_v$ = gas constant vapor

From measured data, inlet conditions of nitrogen and water is 71.5$^{o}$F, 84.0% RH. To correct for air at 70$^{o}$F, (the calibration curve of the venturi)

$$\frac{V_{s2}}{V_{s2}'} = \left[\frac{460 + 70}{460 + 71.5}\right]^{1/2} = 0.998$$

Thus it is seen that temperature corrections are negligible. To determine $p_n$, the barometric pressure (14.7 psi) was added to the piezometric pressure of 10.6 inches of water (.382 psia) giving the total pressure of 15.08 psia.

From Figure IV-2, the saturation vapor pressure at 71.5$^{o}$F is 20 Torr. At 84% R.H., $p_v$ = (.84) (20) = 16.8 Torr = 0.325 psia

$$p_n = 15.08 - 0.325 = 14.75 \text{ psia}$$

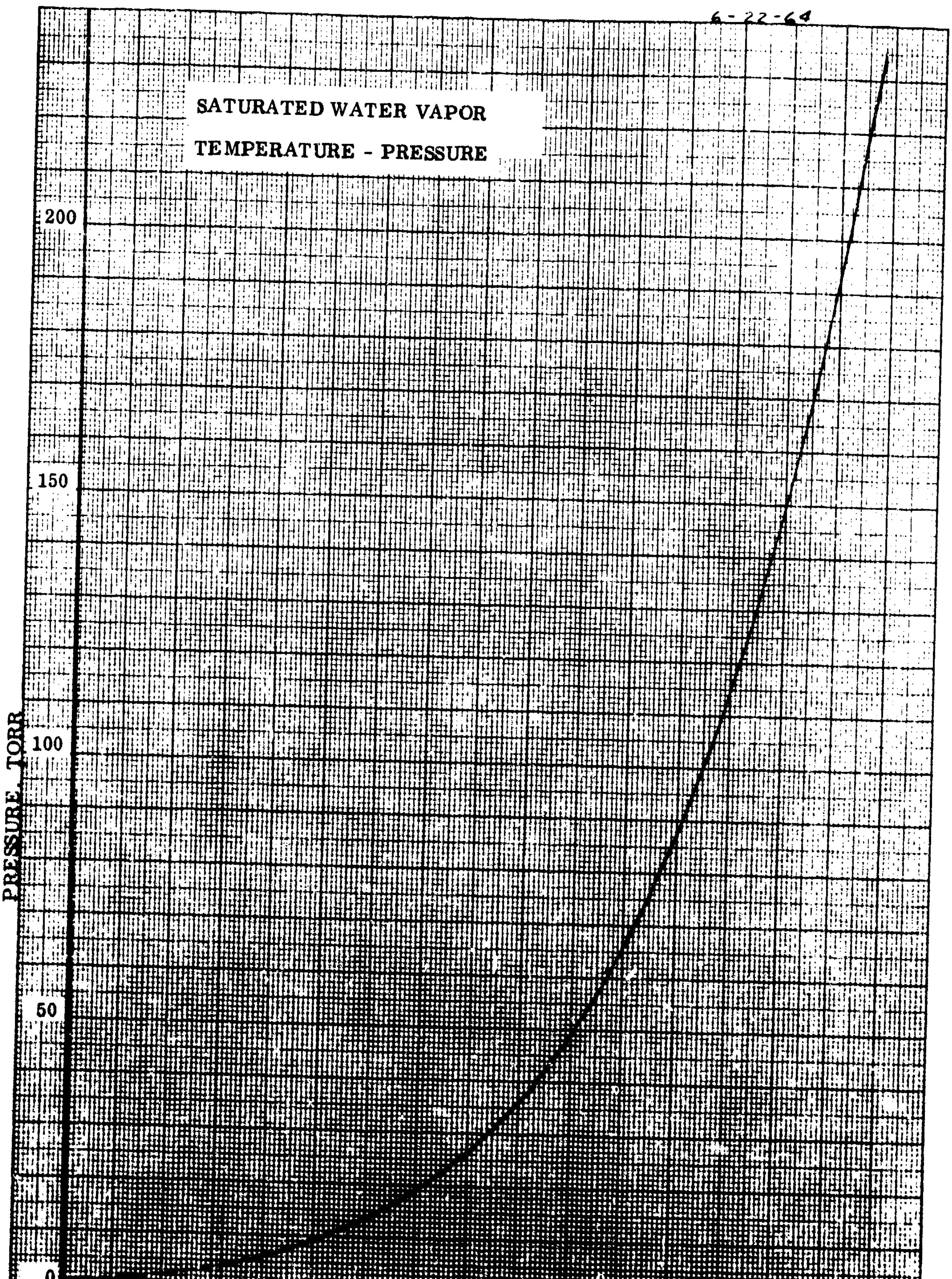



Figure IV-2. Temperature - Pressure Curve of Saturated Water Vapor

$$R_n = 55.1$$

$$R_v = 85.7$$

$$W_s = \frac{(.325)(55.1)}{(14.75)(85.7)} = 0.0142 \ \frac{\text{lb } H_2O}{\text{lb } N_2}$$

NOTE: By using a psychrometric chart, the humidity ratio at 71.5°F, 84% R.H. is 0.0140 lb $H_2O$/lb d.a. Thus it can be seen that using nitrogen instead of air does not make a significant difference in moisture carrying capabilities. Returning to equation A.IV-4 above, assume

$$W_n = 1 \text{ lb Nitrogen}$$

$$W_v = 0.0142 \text{ lb Water}$$

$$W_m = 1.014 \text{ lb mixture}$$

$$R' = \frac{(0.0142)(85.7) + (1)(55.1)}{1.014} = 55.5$$

Substitute into equation IV-3

$$\frac{V_{s2}}{V_{s2}'} = \left[\frac{53.3}{55.5}\right]^{1/2} = 0.98$$

The corrected flow would be proportional to the corrected velocity.

$$\text{Flow} = \frac{2.35}{.98} = 2.40 \text{ cfm}$$

From this example, it can be seen that flow correction is small, and the calculations are tedious. For this report, therefore, although the nitrogen was set for 2.35 cfm in test by the venturi reading, 2.40 cfm was used for calculations. Even without correction, the error would be only 2%. In addition, tables of Moist Air Properties (Reference 5) were used considering nitrogen-water vapor mixtures to be very close to air-water vapor mixtures.

b) Vapor Flow

From Figure IV-3, or for more accurate results using a steam table, the specific volume of saturated steam at 71.5°F (inlet temperature) is 827.22 ft³/lb. The reciprocal is the density of 1/827.22 lb/ft³. But the inlet relative humidity was 84%; hence, the density ($\rho_1$) = $\frac{0.84}{827.2}$

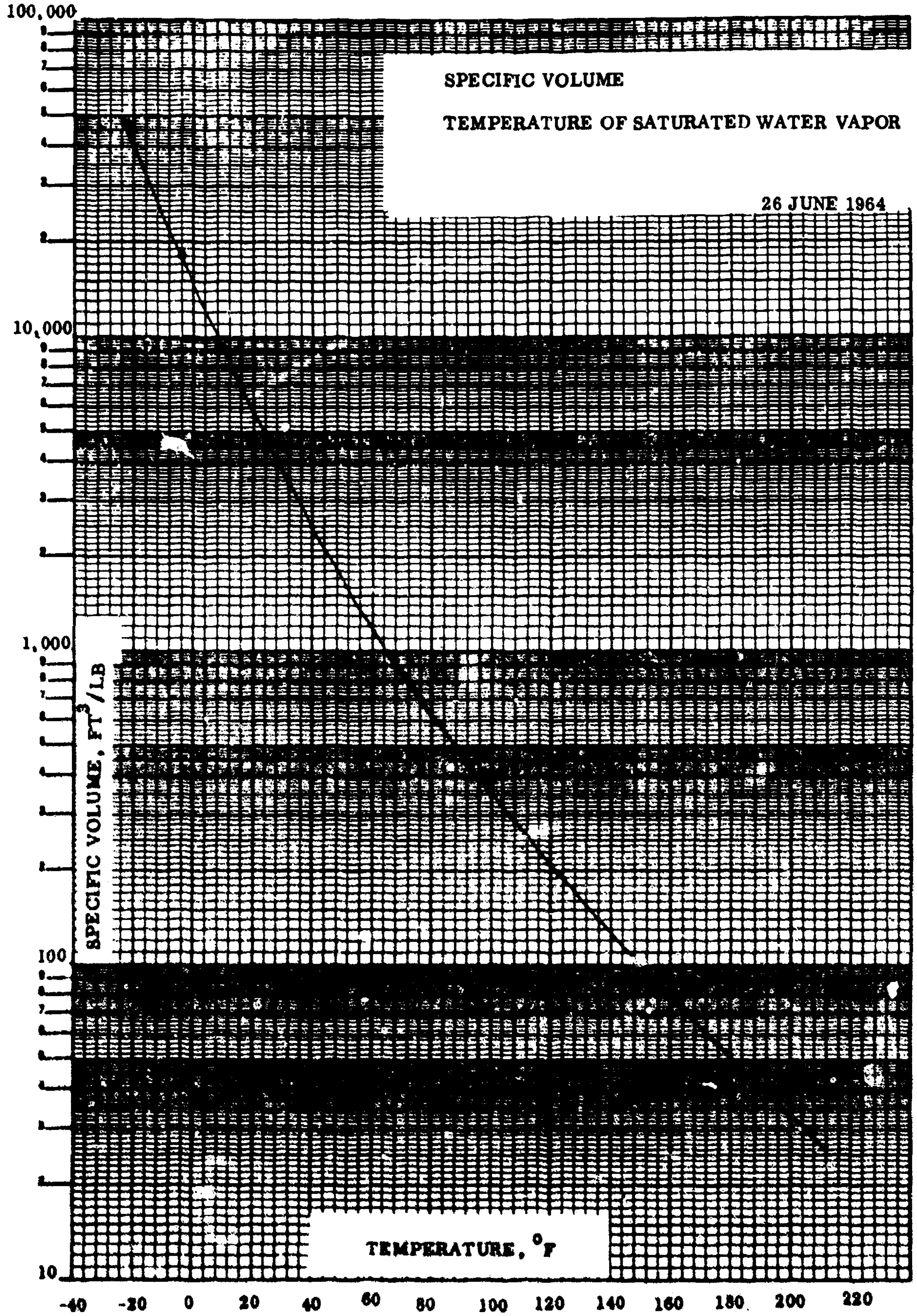



Figure IV-3. Temperature - Specific Volume Curve of Saturated Water or $$\rho_1 = 0.001015 \frac{\text{lb}}{\text{ft}^3}$$

For a flow of 2.40 cfm, and a test time of 44 minutes, the inlet water that becomes vapor is (0.001015) (2.40) (44) = 0.1075 lb water.

Converting to grams gives 48.7 g inlet. Similarly, for the outlet condition of 71.5°F, 81.0% R.H.

$$W_v = \left(\frac{0.81}{827.2}\right)(2.40)\ (44)\ (454) = 47.0 \text{ g outlet}$$

c) Water To Entrained Droplets, and Water Balance

| | | | |
|---|---|---|---|
| Total Injected, g | 795 | | measured |
| Excess water in Mix chamber, g | | 702.5 | measured |
| Excess water in lower duct, g | | 23.0 | measured |
| Water to vapor, g | ___ | 48.7 | calculated |
| Totals, g | 795 | 774.2 | |

Net water to droplets = 795 - 774.2 = 20.8 g
Water in Cone = 20.0 g

Water Balance - 0.8 g (loss)

d) Percent Water Removal

Water vapor in - water vapor out = 48.7 - 47.0
= 1.7 g

The +1.7 g difference indicates a condensation in the cone; therefore, since 20.0 grams were measured in the cone, the droplets trapped by the membrane were 20.0 - 1.7 = 18.3 g. For test time of 44 minutes, membrane collection rate =

$$\frac{18.3}{44} = 0.415 \frac{\text{g}}{\text{min}}$$

Entrained Droplet Rate $= \dfrac{20.8}{44} = 0.472 \dfrac{\text{g}}{\text{min}}$

Percent Water Removal $= \dfrac{0.415}{0.472} \times 100 = 88\%$

## 2. WATER VAPOR PERMEATION CONCEPT - DATA FOR TABLES

The data of Tables X through XVIII can be determined by the method to be given in this section. Even though Table X listed the results of a test in which the compressor was located between the membrane and the cone, and in Table XIV, the compressor was downstream of the cone, all the significant temperature, humidity and pressure readings were taken at the same location for both concepts; hence, it is sufficient to explain one set of data, and Table XIV will be that set.

a) Permeability Equation

Equation IV-6 from the text is repeated here for convenience in rearranged form. In the following paragraphs the quantities defined are computed.

$$P = \frac{Qt}{A \, \Delta \, p} \qquad \text{(IV-6 repeated from text)}$$

Where Q is the flow through the membrane, std cc/sec

t is the membrane thickness, cm

A is the membrane area, $cm^2$

$\Delta$ p is the pressure gradient, cm Hg

P is the permeability, $\frac{\text{(std cc)}}{\text{(sec)}} \; \frac{\text{(cm)}}{(cm^2)\text{ (cm Hg)}}$

b) Membrane Dimensions, A and t

The free area of membrane after potting in epoxy and sealing leaks, consisted of 26 sides of 9 in. wide x 5 in. deep.

$$A = (26)\ (9)\ (5) = 1180\ in^2 = 7560\ cm^2$$

Since there was some wrinkling and pinching of the edges of the filter paper and membrane when the assembly was under vacuum, the face area (of 9 in. wide x .15 in. per corrugation x 27 corrugation = 36.5 $in^2$) was neglected This would, at most, introduce an error of approximately 3%.

Membrane thickness for this assembly was 0.002 inches = 0.00508 cm.

(NOTE: That in Tables XI, XII, and XIII, t = 0.001 = 0.00254 cm)

c) Flow Volume, Q

The measured collected water from the cone was 35g. Test time was 135 minutes or 8100 seconds. At standard temperature and pressure, ($0^{\circ}$C, 760 Torr). The specific volume of water vapor is 1245 cc/g. Therefore,

$$Q = \frac{35\ g}{8100\ sec} \times 1245\ \frac{cc}{g} = 5.38\ \frac{\text{std cc}}{\text{sec}}$$

d) Pressure Gradient, $\Delta p$

This term involves the mean logarithmic vapor pressure, equation V-1 from the text is repeated here.

$$\ln \Delta p = \frac{(pv_1 - pv_c) - (pv_2 - pv_c)}{\ln \dfrac{(pv_1 - pv_c)}{(pv_2 - pv_c)}} \qquad \text{(V-1 repeated from text)}$$

$pv_1$ is the vapor pressure inlet

$pv_2$ is the vapor pressure, outlet

$pv_c$ is the vapor pressure, cone

(NOTE: Equation IV-7 in the compression concept is identical except that $pv_c$ is replaced by $pv_3$, the inlet pressure to the compressor.)

From Table XIV, the inlet conditions in the steam duct are 123 $^{O}$F, 66% R.H. The outlet conditions are 110.5$^{O}$F, 68.5% R.H. At saturation, 123$^{O}$F, Figure IV-2, (or steam tables) the vapor pressure is 9.5 cm. If relative humidity is only 66%, $pv_1$ = (.66) (9.5) = 6.30 cm. Similarly, $pv_2$ = (0.685) (67) = 4.58. The average cone pressure was a gage of -29.3 in. Hg with a barometer of 30.2 in Hg.

$$pv_c = (30.2 - 29.3)\ (2.54) = 2.28 \text{ cm Hg}$$

$$\ln \Delta p = \frac{(6.30 - 2.28) - (4.58 - 2.28)}{\ln \dfrac{(6.30 - 2.28)}{(4.58 - 2.28)}} = 3.05 \text{ cm Hg}$$

e) Permeability Computation

Combining the computed terms in equation IV-6,

$$P = \frac{(5.38)\ (.00508)}{(7560)\ (3.05)} = 1180 \times 10^{-9} \frac{(\text{std cc})\ (\text{cm})}{(\text{sec})\ (\text{cm}^2)\ (\text{cm Hg})}$$

f) Efficiency

The manufacturer, DuPont, rates the CA-148 as $P = 5000 \times 10^{-9}$, and this value was virtually attained in test 4 of Appendix III; therefore, an overall efficiency of the system collection based on permeability can be made as follows.

$$\text{Efficiency} = \frac{1180 \times 10^{-9} \text{ test value} \times 100}{5000 \times 10^{-9} \text{ vendor value}} = 23.6\%$$

3. WATER VAPOR PERMEATION CONCEPT - WATER BALANCE

A complete water balance was not run in the vapor concept, since much of the humid air was discharged to the room ambient. A check of vapor flow was made, however, and that will be reported here for Table XIV.

From equations A. IV-1, 2, and 3 of this appendix correction factors for flow were determined. For the air-steam mixture, the inlet was 123°F, 66% R. H.

At 123°F saturated air, there is 0.08955 lb $H_2O$/lb. d.a. (Ref. 5). At 66% R.H., the ratio is (.66) (.08955) = 0.0592 lb $H_2O$/lb d.a.

From equation IV-4, $R' = \frac{(.0592)(85.7) + (1)(53.3)}{1.0592} = 55.0$

Then $$\frac{Vs_2}{Vs_2'} = \left[\frac{53.3}{55.0}\right]^{1/2} = 0.985$$

The temperature correction factor from 70° to 123°F is:

$$\frac{Vs_2}{Vs_2'} = \left[\frac{460 + 70}{460 + 123}\right]^{1/2} = 0.955$$

Then for the venturi reading of 1.0 inch water giving an apparent 2.35 cfm, the corrected flow is $\frac{2.35}{(0.955)(0.985)} = 2.50$ cfm

The specific volume @ 123°F dry air = 14.687 $ft^3$/lb. For full evaporation of water, the specific volume increases by 2.103 $ft^3$/lb. But since R.H. is 66%, Specific Volume Mixture = (14.687) + (.66) (2.103)
= 16.077 $ft^3$/lb d.a.

Then the inlet air flow is $\frac{2.50 \text{ cfm}}{16.077} = 0.156 \frac{\text{lb}}{\text{min}}$ of dry air
= 224 lb/day

The moisture flow is 0.0592 lb vapor/lb d.a.

Therefore weight flow = (.0592) (.156) = 0.0092 lb/min of vapor
= 13.3 lb/day

The outlet condition of Table XIV was 110.5°F, 68.5% R.H. This reduces to a ratio of 0.0405 lb $H_2O$/lb d.a. For continuous weight flow, the water vapor outlet is flowing at (.156) (.0405) = 0.00631 lb/min
= 9.1 lb/day

The vapor change is .0092 - .0063 = 0.0029 lb/min. For a test time of 135 minutes weight = (135) (.0029) = 0.392 lb = 177 grams. But the actual water collected was 35 grams. The conclusion is that most of the vapor condensed in the steam duct, and was not recovered.

4. PERMEATION OF THE MEMBRANE BY OXYGEN AND NITROGEN

Paragraph IV.4 referred to computations of gas leakage. Assume, for simplicity, an atmosphere of 76 cm Hg total pressure, of which 15 cm Hg is oxygen and 61 cm Hg is nitrogen. From Table XV, for DuPont CA-148, P is $0.07 \times 10^{-9}$ for $O_2$, and $0.029 \times 10^{-9}$ for $N_2$.

By equation IV-6 from the text, and the 2 mil, 7560 $cm^2$ membrane, oxygen flow rate is

$$Q = \frac{PA\,\Delta p}{t} = \frac{(.07 \times 10^{-9})\,(7560)\,(15)}{(.00508)} = 0.00157\ \frac{cc}{sec}$$

$$= .094\ cc/min$$

Similarly, for nitrogen,

$$Q = \frac{(.024 \times 10^{-9})\,(7560)\,(61)}{(.00508)} \times 60\ \frac{sec}{min} = 0.131\ \frac{std\ cc}{min}$$

Combined flow is $0.094 + .131 = 0.225\ \frac{std\ cc}{min}$

If the membranes are only 1 mil thick, (0.00254 cm) twice the flow will be obtained. For two assemblies in parallel, twice the flow would be obtained, give a fourfold increase in volume:

$$Q = 4(.225) = 0.900 \approx 1\ \frac{std\ cc}{min}$$

## APPENDIX V

## LIST OF KEY TEST EQUIPMENT

1. Nozzles - Type H 601F

   Monarch Manufacturing Works
   2501 East Ontario Street
   Philadelphia 34, Penna.

2. Laboratory Hygrometers - Model 101

   Eltronics Incorporated
   11 South Irvine Street
   Warren, Penna.

3. Pressure Regulator - Type 2A2

   Norgren Co.
   Denver, Colorado

4. Blower - Tube Axial (27 V DC)

   Globe Industries
   Dayton, Ohio

5. DC Power Supply Model MR 28-5 Type A 28 V DC

   Magnetic Research
   3160 W. El Segundo Blvd.
   Hawthorne, California

6. Humidity Chamber - Model T3 OUFR-100-350

   Tenny Engineering Co.
   Union, N. J.

7. Recording AC Watt Meter 8CH - 3 KBF10 0-1 Kw
   with
   WATT VAR Meter 890-1089 G13 for 3 wire 3 phase

   General Electric Company

8. Chain Compensated Gasometer (Spirometer)
   120 L Capacity Serial No. 1106

   Warren E. Collins Co.
   Boston, Mass.

9. Gauges Mercury Column 0 - 30 inches
Water Column 0 - 2.0 inches
Water Column 0 - 30 inches

10. Venturi - (Special Design)

See curve included for calibration, Figure IV-I

| | | |
|---|---|---|
| Nominal Dimensions: | 0.340 in. | Throat |
| | 0.650 in. | Entrance/Exit |
| | 0.882 in. | Entrance to Throat |
| | 2.230 in. | Throat to Exit |

## REFERENCES


1. Donald Petrash and Edward Otto, Studies of the Liquid Vapor Interface Configuration in Weightlessness, NASA Cleveland, Ohio (Prepared for American Rocket Society September, 1962).

2. V. Stannet, M. Swarc, R. L. Bhargava, J. A. Meyer, A. W. Myers, C. E. Rogers, Permeability of Plastic Films and Coated Papers to Gases and Vapors, TAPPI No. 23, Technical Association of the Pulp and Paper Industry, New York, N. Y., 1962.

3. Paul M. Hauser and A. Douglas McLaren "Permeation through and Sorption of Water Vapor by High Polymers." Industrial and Engineering Chemistry, Vol 40, No. 1, pp. 112-117, January, 1948.

4. International Critical Tables of Numerical Data, Physics Chemistry and Technology, Vol II, p. 272, McGraw-Hill Book Co., Inc., New York, N. Y., 1927.

5. Heating Ventilating and Air Conditioning Guide, American Society of Heating and Air Conditioning Engineers, New York, N. Y., 1956.

6. V. M. Faires, Applied Thermodynamics, Macmillan Company, New York, N. Y., 1941.

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13. ABSTRACT The objective of this program was to investigate the feasibility of separating water vapor and/or droplet water from air by selective permeation through microporous materials. Two concepts were studied with the goal of removing 0.1 pound per hour of water or water vapor from approximately ten pounds of air per hour.

In the first concept, a 15 micron average pore size barrier of sintered Kel-F was positioned perpendicularly across a droplet laden air stream. Positive removal of droplet water was observed.

A second concept studied utilized a continuous sheet membrane of cellulose acetate to transmit water vapor but to block the passage of permanent gases. A modified dry-vane type commercial compressor was used to produce a high suction level. While positive evidence of vapor transfer and water condensation were observed, problems of complete edgewise sealing of the cellulose acetate membrane and the cooling and suction limitations of the oil-free, modified commercial pump precluded achievement of design water removal rates.

Further work to perfect water droplet separation from air through porous media should be preceeded by pressure drop determination of droplet laden air through unit areas of porous media. Further work to perfect the membrane permeation concept would have to include a thorough search for or development of a high suction, oil-free compressor for compression of water vapor and more effective means for edgewise sealing of sheet membrane materials.


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| 14. KEY WORDS | LINK A ROLE | LINK A WT | LINK B ROLE | LINK B WT | LINK C ROLE | LINK C WT |
|---|---|---|---|---|---|---|
| Microporous Materials | | | | | | |
| Membranes | | | | | | |
| Water Removal | | | | | | |
| Permeation | | | | | | |
| Droplet Removal | | | | | | |
| Cellulose Acetate | | | | | | |
| Sintered Kel-F | | | | | | |
| Sintered Teflon | | | | | | |
| Water Management | | | | | | |

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