THE DEVELOPMENT OF GAS AND LIQUID CIRCULATING DEVICES FOR MANNED SPACE ENCLOSURES

JOSEPH FLATT

FOREWORD

This investigation was initiated by the Biomedical Laboratory of the Aerospace Medical Research Laboratories, Aerospace Medical Division, Wright Patterson Air Force Base, Ohio. The development program was carried out by Tech Development, Inc., Dayton, Ohio under Contract No. AF33(615)1251. Mr. Joseph Flatt was the principal investigator for Tech Development, Inc. Contract monitors for the Aerospace Medical Research Laboratories were Mr. Irving H. Lantz and Mr. D. A. Keating. The work was performed in support of Project No. 6373, "Equipment for Life Support in Aerospace" and Task No. 637302, "Respiratory Support Equipment." The work sponsored by this contract was started in February 1964 and completed in February 1965.

This technical report has been reviewed and is approved.

WAYNE H. McCANDLESS
Technical Director
Biomedical Laboratory
Aerospace Medical Research Laboratories

ABSTRACT

An analytical and experimental investigation was conducted to determine the feasibility of utilizing the energy available in breathing oxygen, when expanded from storage pressure to breathing pressure, to power two types of pumping devices; (1) an oxygen-circulating blower, and (2) a water-circulating pump. With an oxygen weight flow of 2.65 lb/hr and an ambient environmental pressure of 5 psia, the required performance of 5 cfm at 10 inches H_2O static pressure for the circulating blower and 1 lb/min at a back pressure of 10 inches H_2O for the water pump were met. Performance considerably in excess of the requirement was obtained for the water pump. Both units were powered by single-stage impulse turbines and weighed less than the 1-pound requirement. Both units operated reliably without mechanical malfunction and appear to be suited for manned aerospace enclosures.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

SECTION I

INTRODUCTION

This study covers the design, fabrication and testing of two types of gas-driven, impulse-turbine pump devices. These devices were designed to circulate a habitable atmosphere and to circulate cooling liquids. This technique was used to reduce the electrical power required by life support systems. The energy to drive the circulating devices is obtained from the pressure and flow inherent to the source of breathing oxygen when compressed oxygen is used. This study has direct application to manned space flight life support systems, particularly to extravehicular operations and to emergency mode operations within the vehicle.

GENERAL CONSIDERATIONS

Fabrication considerations primarily dictated selection of a turbine-type motor for these devices. With the minute flow areas resulting from the low drive oxygen flows, the prevention of leakage in valves and around sliding fits in positive displacement motors would require unrealistic, if not impossible, tolerances. There are no leakage problems in the turbine approach, and oxygen flow is positively controlled by the turbine nozzle.

Maximum efficiency of a single-stage, impulse turbine is obtained, theoretically, when the nozzle velocity is twice that of the bucket velocity. Because of the windage drag of the buckets when a single nozzle is used in devices of this size, this ratio cannot be achieved practically. Maximum turbine power output is obtained at a bucket speed where there is maximum difference between gross turbine power output and windage power. The pumping impellers are sized for maximum efficiency and capacity at a speed corresponding to the optimum for the turbine.

Nozzle drive pressure is a compromise, with high pressures desired to provide maximum thrust per unit weight flow and low pressure being required to permit a nozzle diameter of reasonable size from a fabrication standpoint. Efficient operation requires careful attention to tolerance control, running clearances, bearing friction, and dynamic balancing.

SECTION II

TYPE I - OXYGEN CIRCULATING DEVICE

ANALYTICAL CONSIDERATION

The theoretical expression for static power required to pump fluid is

$$HP = \frac{CFM \times Ps}{6341} \qquad (1)$$

where CFM is quantity flow in cubic feet/minute
Ps is static pressure rise across the pump in inches of water
HP is horsepower

For the type I device, the required circulation is 5 cfm at a static pressure rise of 10 in. of water.

$$HP = \frac{5 \times 10}{6341} = 0.0079 \text{ Horsepower} \qquad (2)$$

Ideal theoretical horsepower of a single-stage impulse turbine is

$$HP = 2\, T_o \left(1 - \frac{V_B}{V_N}\right) \times \frac{V_B}{550}$$

where T_o is nozzle momentum $= \frac{V_N}{32.2} \times W_a$

V_B is bucket speed - ft/sec

V_N is nozzle jet speed - ft/sec

W_a is weight flow of nozzle - lb/sec (3)

For a convergent-divergent nozzle considered feasible from a fabrication standpoint, the minimum size nozzle throat was 0.0135 inch (number 80 drill). With this size throat, the computed absolute drive pressure is 205 psia for a weight flow of 2.65 lb/hr or 0.000736 lb/sec. At the design ambient pressure of 5 psia, the pressure ratio of $\frac{5}{205} = 0.0244$ yields (from supersonic flow tables) a nozzle Mach number of 3.07, a throat to exit area ratio of 0.2209 and a nozzle velocity of 2,020 ft/sec, assuming 60° F drive oxygen temperature. Theoretically, maximum turbine power occurs (single-stage impulse) when $V_B = \frac{1}{2} V_N$.

At this condition, for the stated oxygen weight flow and drive pressure,

$$\text{HP} = \left(\frac{2 \times 2020 \times 0.000736}{32.2}\right)\left(1 - \frac{1010}{2020}\right)\left(\frac{1010}{550}\right)$$

$$= 0.0849 \text{ HP @ } V_B = 1010 \text{ ft/sec} \qquad (4)$$

From this analysis ideally the required efficiency for the blower-turbine train need be only

$$\frac{0.0079}{0.0849} = 9.32\%$$

Blower efficiencies of 40% should be obtainable, and adequate margin should exist. This margin, however, can be affected seriously by turbine efficiency.

Actual turbine power that can be realized is influenced by several items:

a. Nozzle discharge angle

b. Bucket velocity coefficient

c. Bucket windage drag

d. Nozzle thrust coefficient.

Bucket force varies directly as the cosine of the nozzle discharge angle. Practical fabrication considerations determine this angle and, for these devices, a value of 20° was selected. This angle is typical for impulse turbine arrangements. Bucket velocity coefficient is dependent primarily on nozzle Mach number, and the applicable value for the drive pressure selected ($M - 3.07$) is approximately 0.5. Bucket windage drag and nozzle thrust coefficient were determined experimentally and are discussed next. The resulting net power output of the turbine was found to be barely adequate to meet the required performance.

DESIGN

The general arrangement selected for this unit was that of a shrouded centrifugal blower having, integrally attached, a peripheral single-stage impulse turbine with the turbine discharge mixing with the wheel efflux in the exit scroll of the blower. Prior to selection of the final geometry, tests were conducted on nozzle efficiency and on turbine windage drag.

Nozzle thrust was measured, under sea level conditions, by impinging the nozzle efflux on the platform of a sensitive balance. Height of the nozzle above the platform was varied to obtain the maximum thrust. Optimum height was not critical and occurred in the range of 1/16 to 1/4 inch above the plate. At sea level conditions, maximum measured nozzle thrust for the design value of 2.65 lb/hr was found to be 0.0340 pounds. Theoretically, at sea level, this nozzle should have a thrust of 0.0415 pounds, indicating a nozzle efficiency of $\frac{0.0340}{0.0415} = 82.5\%$.

Turbine windage drag was measured in the following manner. A turbine wheel 2.278 inches in diameter with 1/16-inch-high buckets was attached to a variable speed d-c motor and spun at various speeds. The motor voltage and current were recorded for each speed. The motor was then calibrated on an eddy current dynamometer, with torque measured at each voltage and current flow condition observed during the turbine tests. From the torque and speed, windage horsepower was computed at sea level and corrected to 5 psia by the ratio of the densities.

Using the observed nozzle efficiency and including the effects of nozzle discharge angle and bucket velocity coefficient, the gross horsepower output, windage horsepower, and net horse-power output at 5 psia were determined and are plotted in figure 1[1]. The results indicate a large loss due to windage drag and that optimum blade speed falls in the 325 to 375 ft/sec range. Maximum power output to be expected is 0.0190 hp.

A turbine 2.52 inches in diameter was arbitrarily selected as a reasonable size. With this diameter and a 375 ft/sec bucket speed, the design rpm was 33,900. The blower was sized accordingly by conventional methods to yield the required output at peak efficiency for this speed.

Mechanically, the unit was designed for compactness and minimum weight. All parts, with the exception of the stainless steel shaft, ball bearings, and nozzle, were constructed of aluminum. Ball bearings are class VII precision, permanently lubricated with grease compatible with oxygen. The nozzle is brass. Figure 2 shows the assembly. Figures 3 and 4 show the unit assembly and component breakdown. The complete unit weight is 0.955 pounds. Overall dimensions are 3-1/8 by 3-1/2 by 2-3/4 inches.

[1] Figures are located on pages 9 through 27.

TEST APPARATUS AND TESTING

A special test apparatus was designed and built to permit measurement of the blower performance over a range of flow and pressures and at several ambient pressures. Under high pressure, drive oxygen passed through a reducer and regulator holding the outlet pressure at 190 psig (205 psia), thence through a flowmeter and into the blower nozzle. Blower inlet pressure was ambient static. Blower discharge exited into a flow channel with a flush static pressure tap to measure back pressure, from there through a calibrated sharp edge orifice for flow measurement, and then exited through a variable size exit orifice to ambient conditions. The entire apparatus was constructed so that it could be housed in a bell jar for operation under low pressure conditions. The operating ambient pressure was controlled by a variable capacity vacuum pump. A schematic of the apparatus is shown in figure 5. Figures 6 and 7 show the equipment.

Tests were conducted at three ambient pressures: 14.7 psia (sea level), 9.84 psia and 4.92 psia. At each ambient pressure, flow rate was measured for various blower back pressures obtained by changing the exit flow area. All tests were made at the nominal drive pressure of 205 psia and 2.65 lb/hr oxygen flow. Performance of the type I unit is plotted in figure 8. At 4.92 psia, the minimum requirement of 5 cfm at 10 in. H_2O is barely met. Large increases in airflow are possible with slight reductions in back pressure, pointing out the importance of keeping back pressure to low levels. At 5 in. H_2O back pressure, better than 10 cfm flow rate is available. Mechanical operation of the unit was good, and no problems were encountered. Referring to maximum power expected, the indicated net efficiency of the blower is

$$N_B = \frac{0.0079}{0.0190} = 41.6\%$$

SECTION III

TYPE II - LIQUID CIRCULATING DEVICE

ANALYTICAL CONSIDERATION

The theoretical static horsepower required to pump the required minimum water flow of 1 lb/min at 10 in. H_2O pressure rise is .0000253 hp. This amount of power is so minute as to permit almost a complete disregard for efficiency in the interest of satisfying other desirable design features such as low speed for bearing life and absence of cavitation, low noise level, low vibration, compactness and low weight.

DESIGN

The general arrangement selected was that of a nozzle-turbine combination, similar to the type I device, magnetically coupled to a conventional centrifugal water impeller. Design speed was arbitrarily selected as 3,500 rpm. The turbine and impeller are coupled through a pair of permanent magnets, with no direct mechanical connection. The turbine and impeller each has its own pair of bearings and spindle and are separated by a nonmagnetic stainless steel diaphragm. The impeller bearings are permanently lubricated with water compatible grease, and the turbine bearings with oxygen compatible grease. The casing and turbine are aluminum; the water impeller and nozzle are brass; and the housing, spindle, and diaphragm are stainless steel. Figure 9 shows the assembly. Figures 10 and 11 show the unit assembly and component breakdown. The complete unit weight is 0.5 pound. Overall dimensions are 2-5/8 in. diameter by 2-5/8 in. long.

TEST APPARATUS AND TESTING

As in the type I testing, a special apparatus was designed and constructed to permit testing at various absolute pressures. The nozzle oxygen drive system and measuring equipment was the same as for the type I device. The water performance was measured by means of a flowmeter housed within the bell jar and two standpipe water tubes to measure inlet and exit water heads, with the difference being the total pressure head of the pump. Water flow was from the low end of a reservoir through the pump, then through a flowmeter into an exit nozzle which could be varied in height to control discharge head, and finally back to the reservoir. The height of the exit nozzle was varied by a motorized device controlled from the outside of the bell jar. A schematic of the apparatus is shown in figure 12. Figures 13 and 14 show the equipment. Tests were conducted at three ambient pressures: 14.7 psia (sea level), 9.84 psia, and 4.92 psia. At each ambient pressure, flow rate was measured for various head differentials across the pump. It was not possible to operate at full drive pressure at any ambient condition, because the performance was so high as to exceed the scale ranges of the head-measuring tubes and flowmeter. Runs were made at several reduced oxygen drive pressures and flow rates at each ambient pressure condition. The results of these tests are shown in Figures 15, 16 and 17. At the ambient pressure of 4.92 psia, the minimum required performance of 1 lb/min at 10 in. head was met with only 85 psia drive pressure and 1.13 lb/hr oxygen flow as compared to the design allowable flow of 2.65 lb/hr. Performance decreased slightly at the higher ambient pressures because of lower nozzle thrust and higher turbine windage drag. At rated oxygen flow, it is estimated that this unit could pump approximately 7 lb/min at 20 in. H_2O head at 5 psia ambient pressure. Mechanical operation of the unit was good, and no problems were encountered.

SECTION IV

CONCLUSIONS

1. The feasibility of utilizing the energy in high pressure oxygen to power an oxygen circulating blower or water pump has been demonstrated.
2. Blower efficiencies in the order of 40% are indicated for small centrifugal units.
3. Miniature turbine power output can be severely restricted by windage drag and bucket losses.
4. Miniature nozzles can be expected to perform at reasonable (82%) efficiencies.
5. Further improvement in performance of both the type I and type II devices could probably be realized by arrangements whereby the turbine windage losses and bucket losses are reduced.

SECTION V

RECOMMENDATIONS

Experimental studies should be conducted on turbine design variations to establish maximum power output for prescribed oxygen weight flows. This would consist primarily of variations in types of buckets as concerns momentum transfer and windage drag and the effect of nozzle drive pressure at constant flow. Once the optimum turbine parameters are established, peak efficiency turbine-blower or turbine water pump combinations can be designed for any specific performance requirement.

Oxygen flow - 2.65 lb/hr @ 205 psia

Ambient Conditions - 5 psia, 60° F

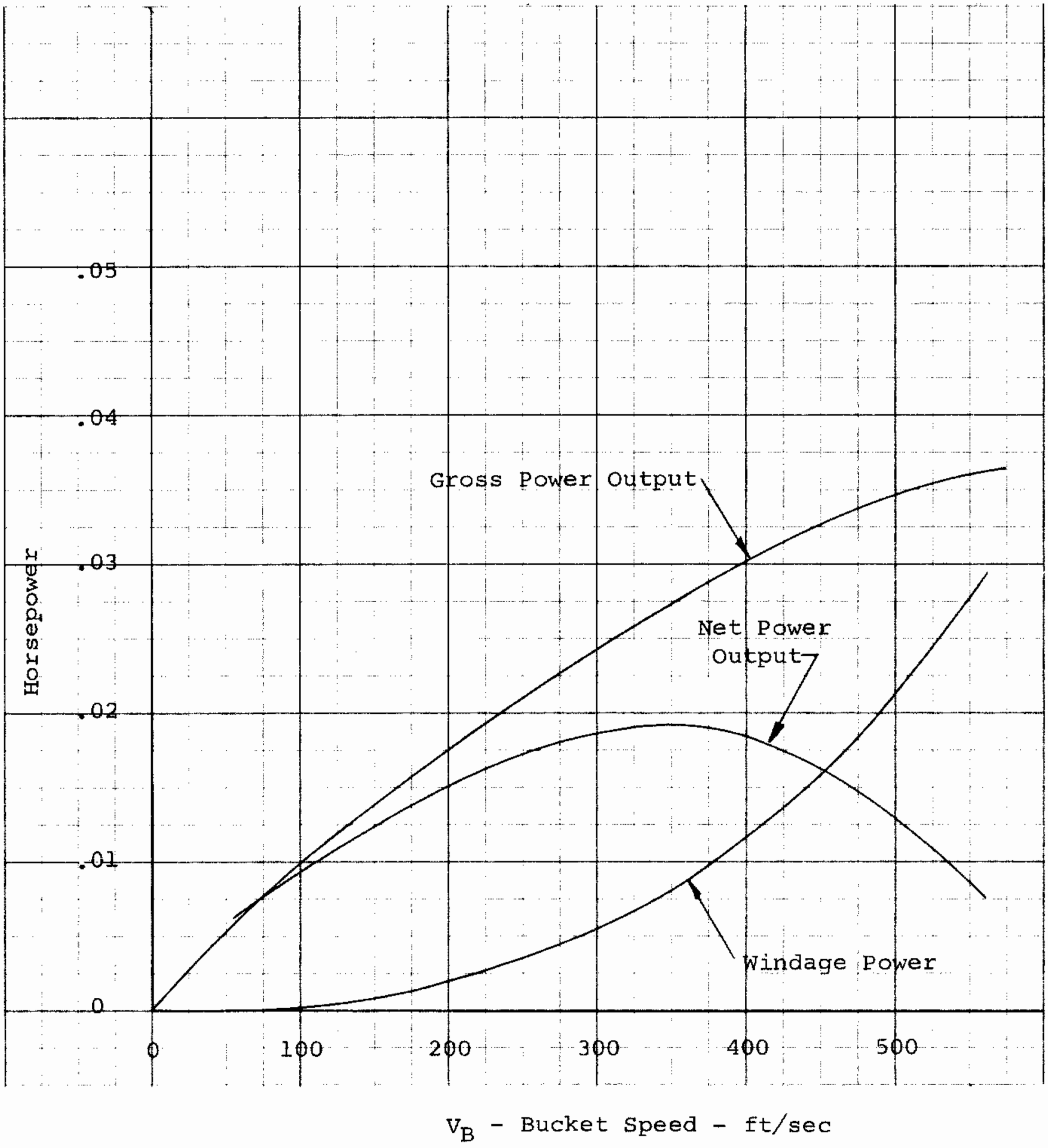
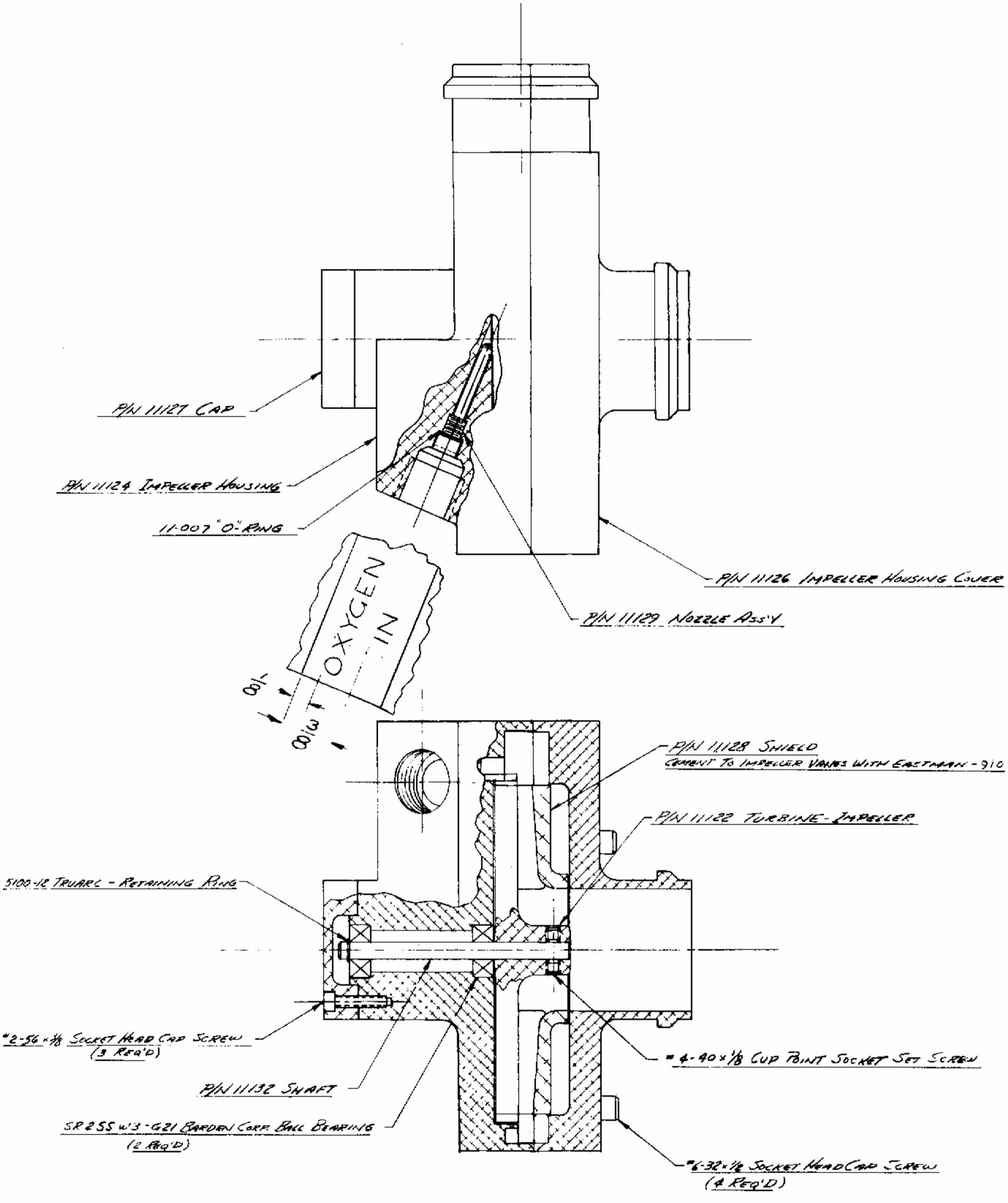


Figure 1

Turbine Power Characteristics


P/N 11127 CAP
P/N 11124 IMPELLER HOUSING
11-007 "O" RING
P/N 11126 IMPELLER HOUSING COVER
P/N 11129 NOZZLE ASS'Y
OXYGEN IN
1/8
3/8
P/N 11128 SHIELD
CEMENT TO IMPELLER VANES WITH EASTMAN - 910
P/N 11122 TURBINE-IMPELLER
5100-12 TRUARC - RETAINING RING
#2-56 × 1/8 SOCKET HEAD CAP SCREW
(3 REQ'D)
#4-40 × 1/8 CUP POINT SOCKET SET SCREW
P/N 11132 SHAFT
SR2SS W3-G21 BARDEN CORP. BALL BEARING
(2 REQ'D)
#6-32 × 1/2 SOCKET HEAD CAP SCREW
(4 REQ'D)

SECTION A-A

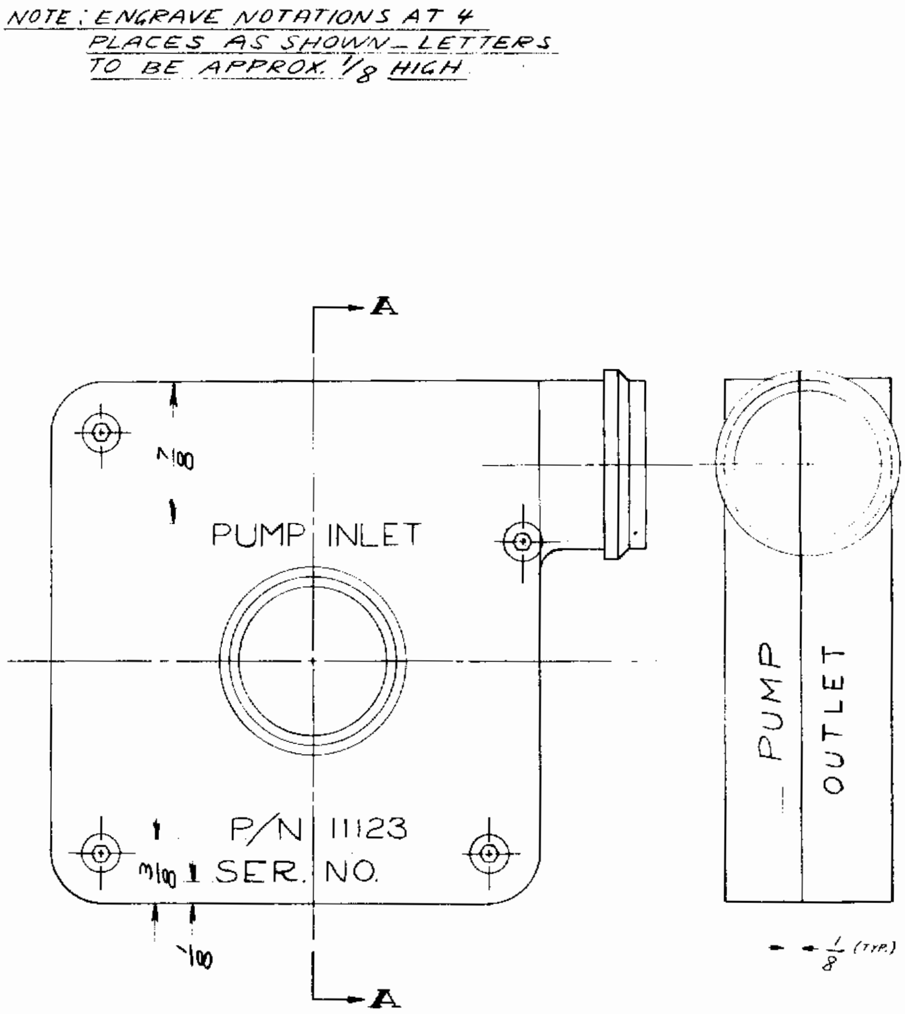


Figure 2

Type I Device Assembly

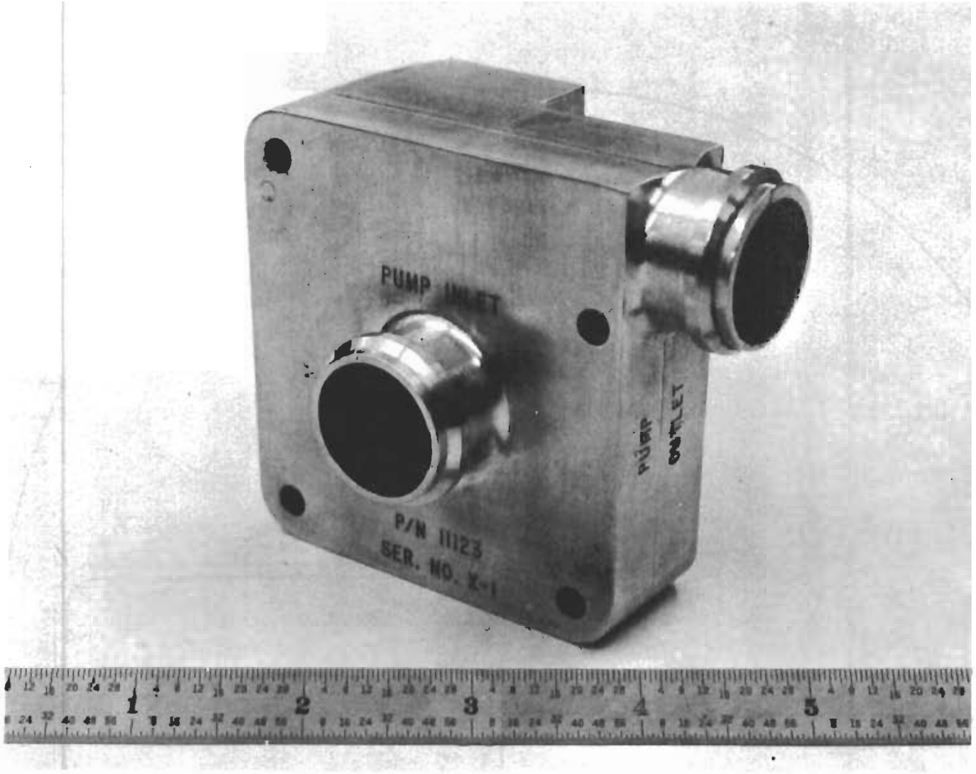


Figure 3. Type I Device

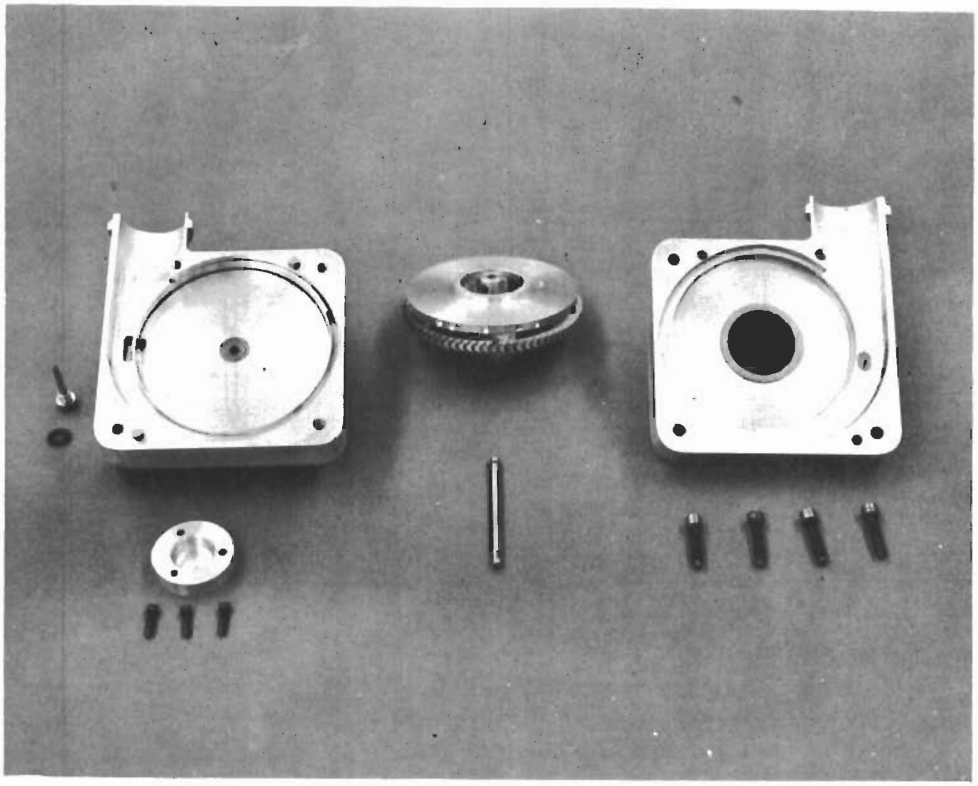
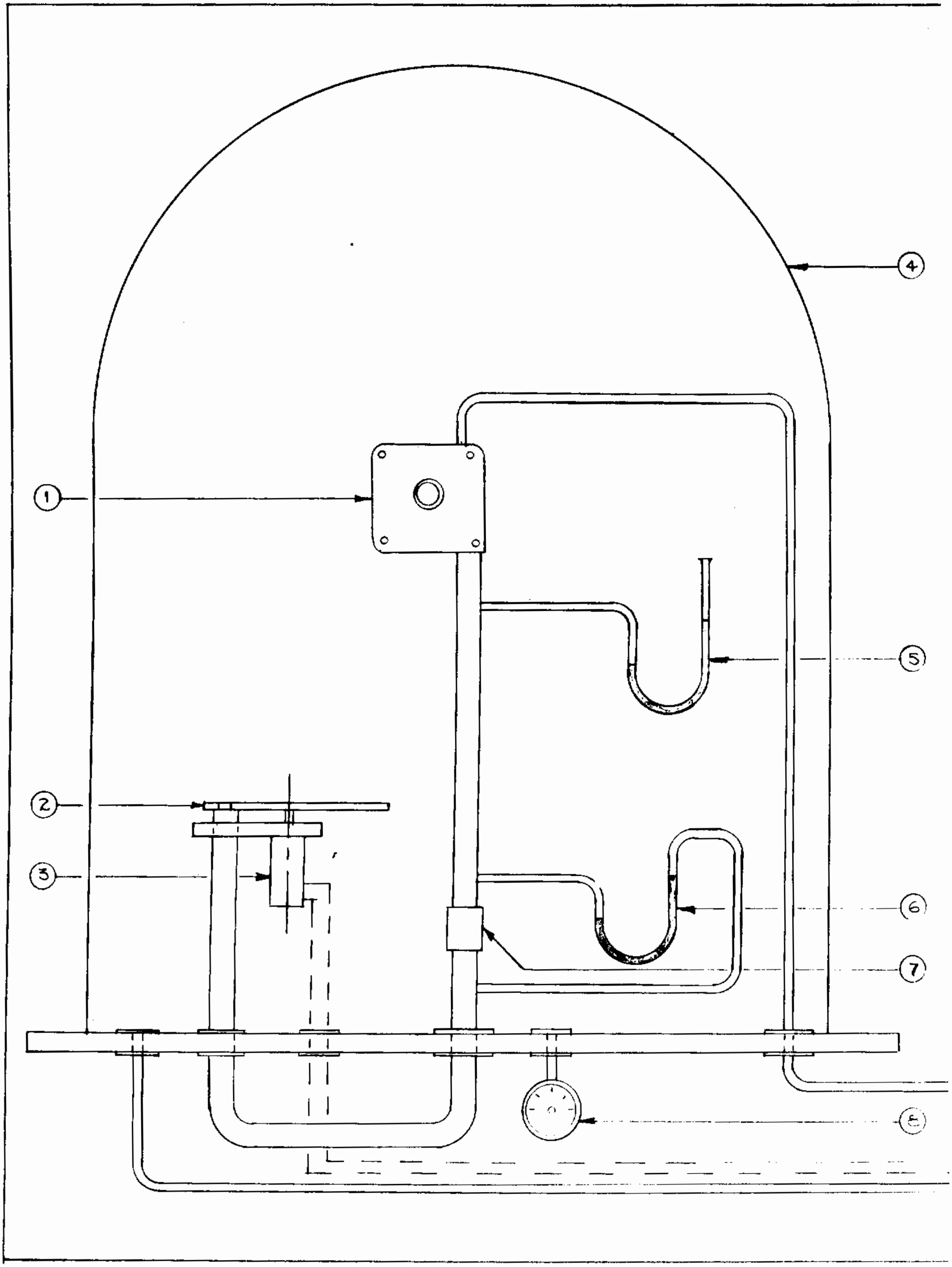


Figure 4. Type I Device Components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No.	Description
1	Type I Device
2	Variable Area Exit Plate
3	Exit Plate Drive Motor
4	Bell Jar
5	Back Pressure Manometer
6	Flow Manometer
7	Orifice Plate
8	Absolute Pressure Gauge
9	Oxygen Drive Pressure Gauge
10	Oxygen Flow Meter
11	Oxygen Pressure Regulator
12	High Pressure Oxygen Bottle
13	Exit Area Motor Control
14	Vacuum Pump

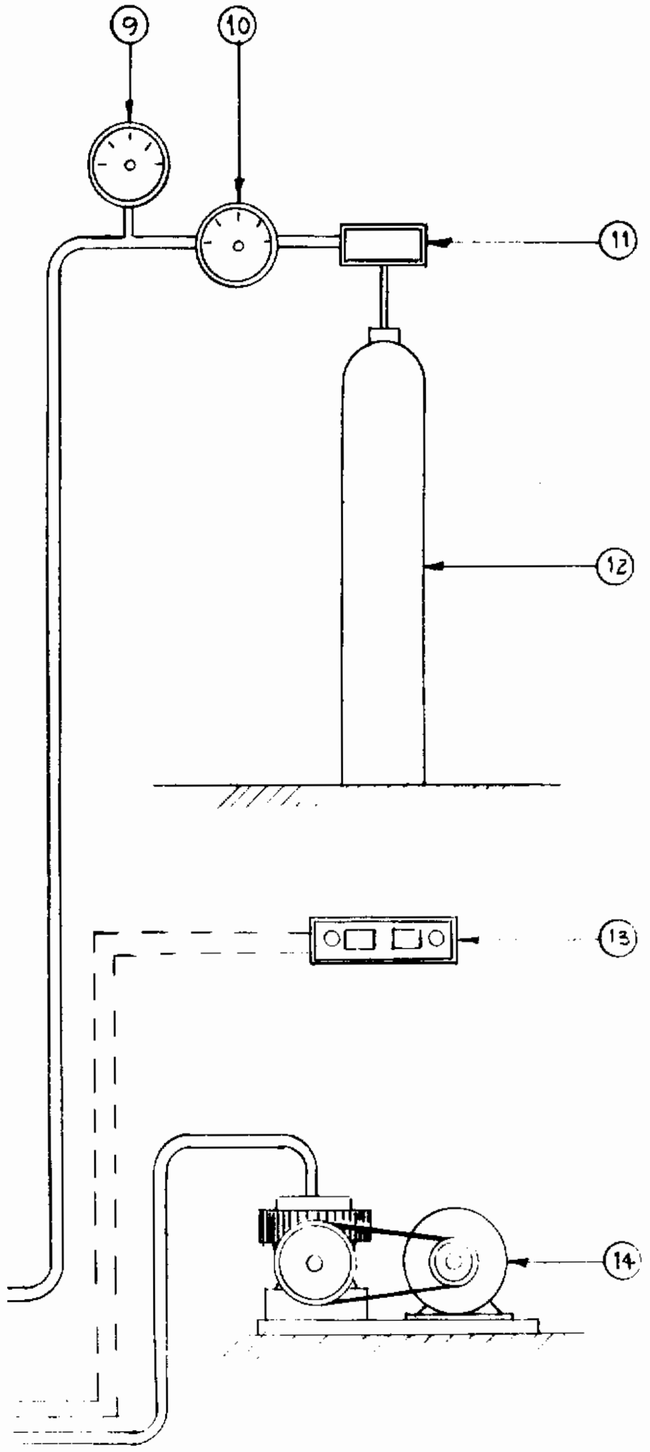


Figure 5

Type I Device

Test Apparatus Schematic

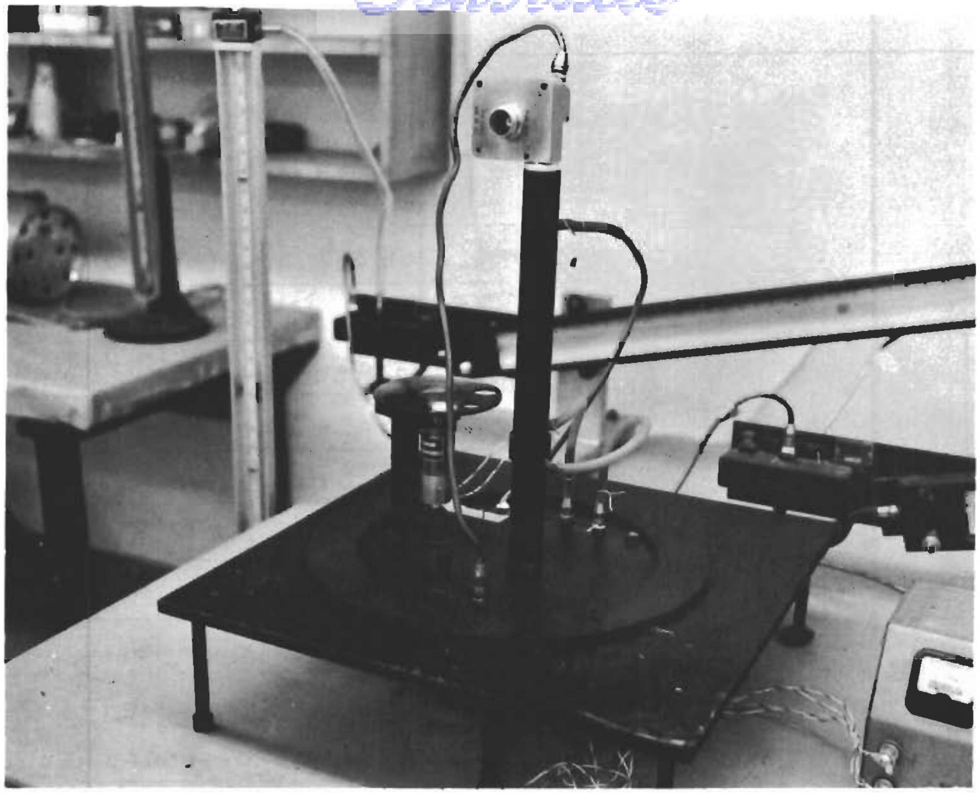


Figure 6. Type I Device Test Apparatus without Bell Jar

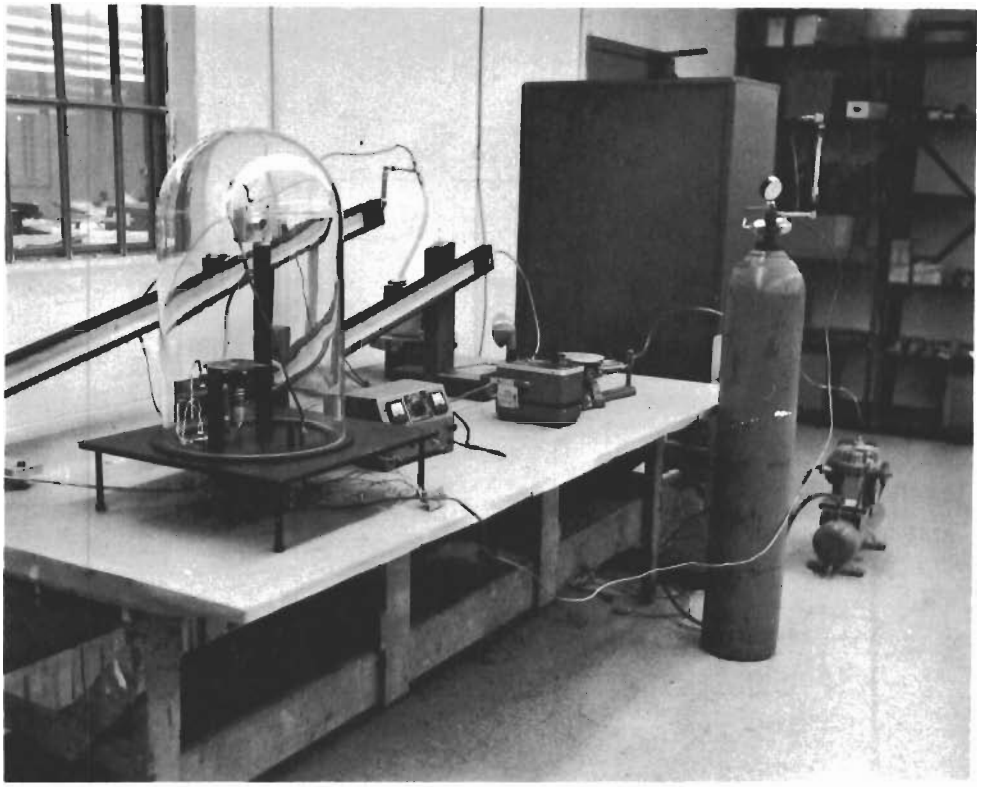


Figure 7. Type I Device Test Apparatus with Bell Jar

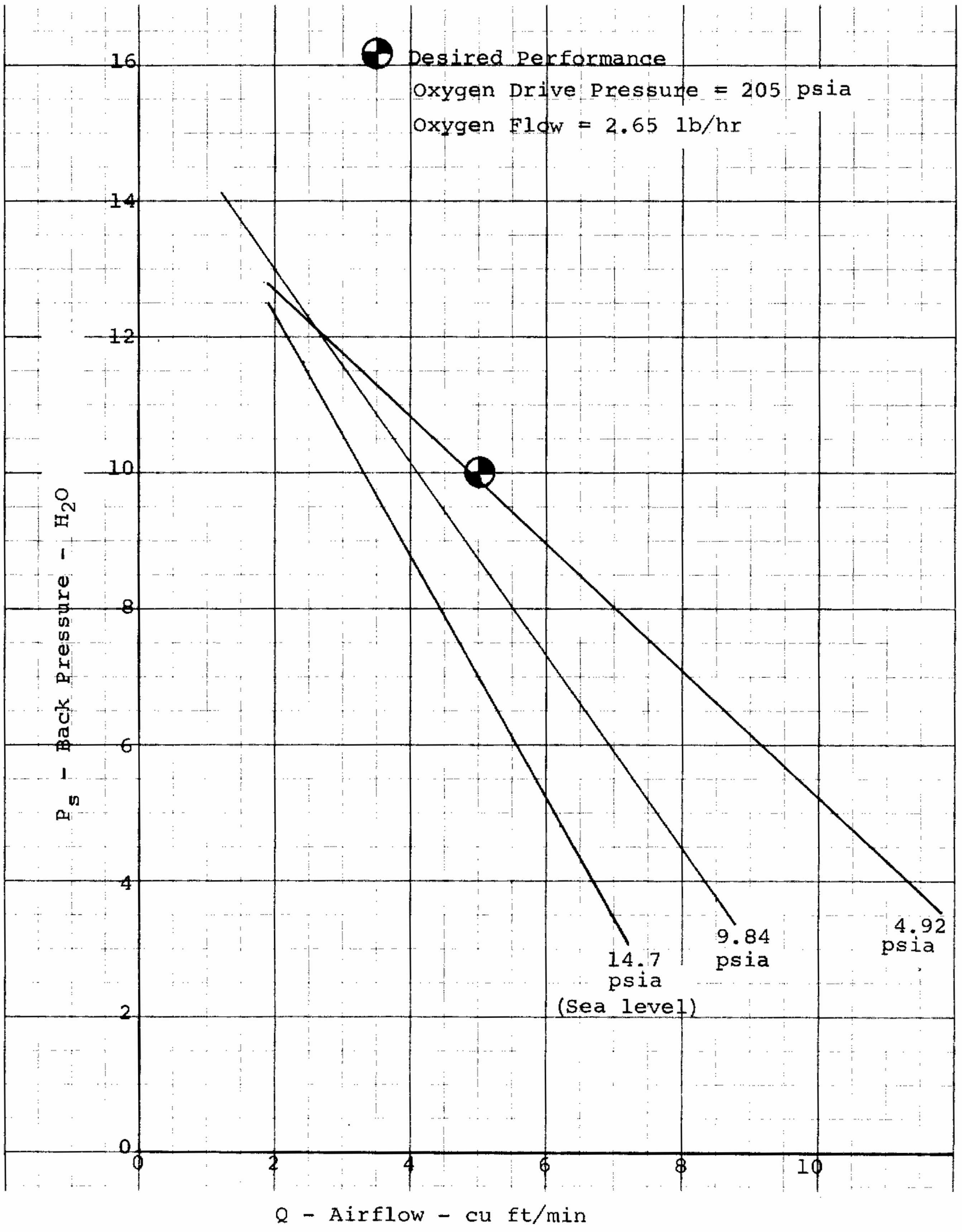
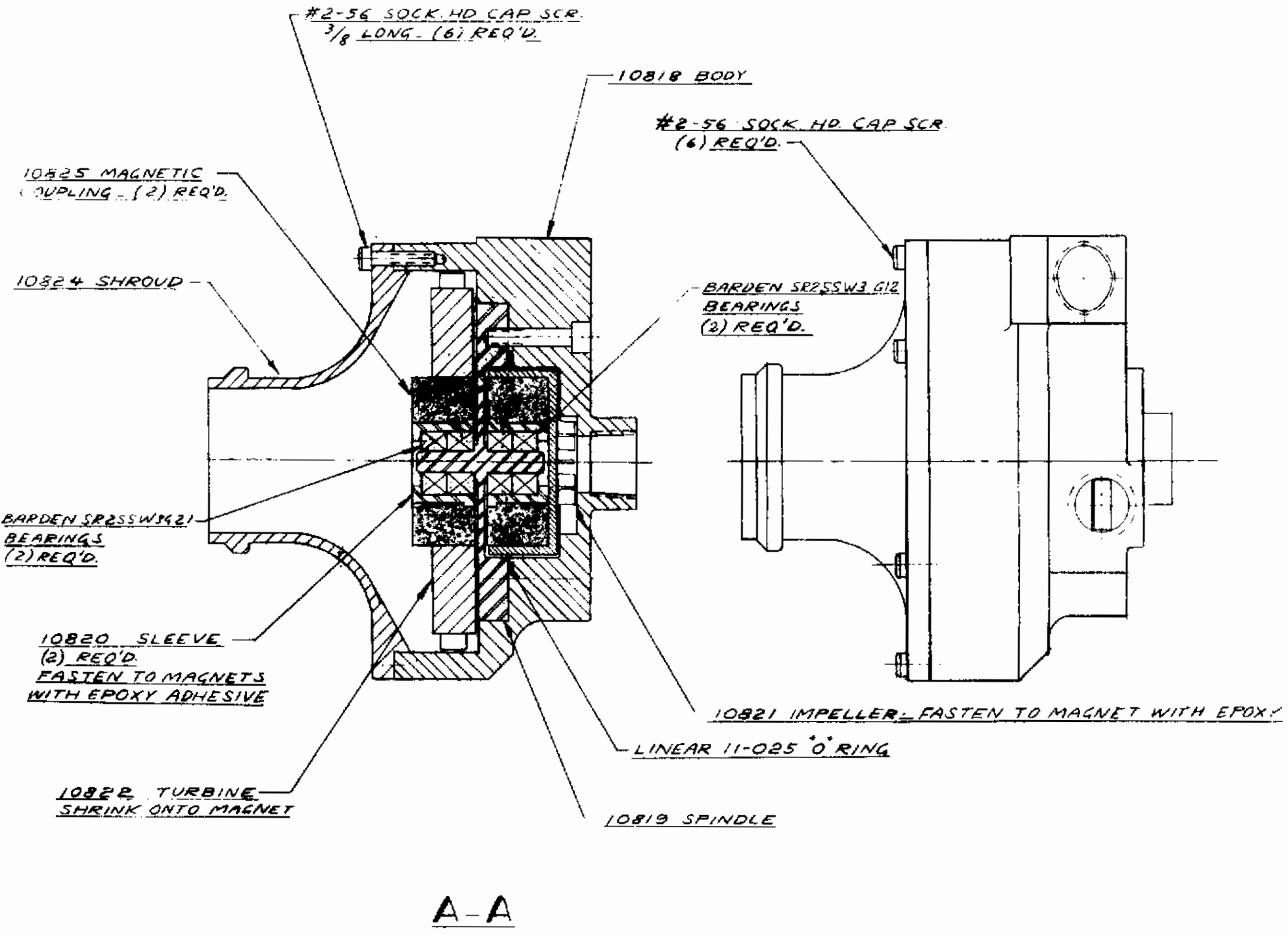


Figure 8
Type I Device Performance


#2-56 SOCK. HD. CAP SCR.
3/8 LONG - (6) REQ'D.
10818 BODY
#2-56 SOCK. HD. CAP SCR.
(6) REQ'D.
10825 MAGNETIC
COUPLING - (2) REQ'D.
10824 SHROUD
BARDEN SR2SSW3 G12
BEARINGS
(2) REQ'D.
BARDEN SR2SSW3G21
BEARINGS
(2) REQ'D.
10820 SLEEVE
(2) REQ'D.
FASTEN TO MAGNETS
WITH EPOXY ADHESIVE
10821 IMPELLER - FASTEN TO MAGNET WITH EPOXY
LINEAR 11-025 "O" RING
10822 TURBINE
SHRINK ONTO MAGNET
10819 SPINDLE
A-A

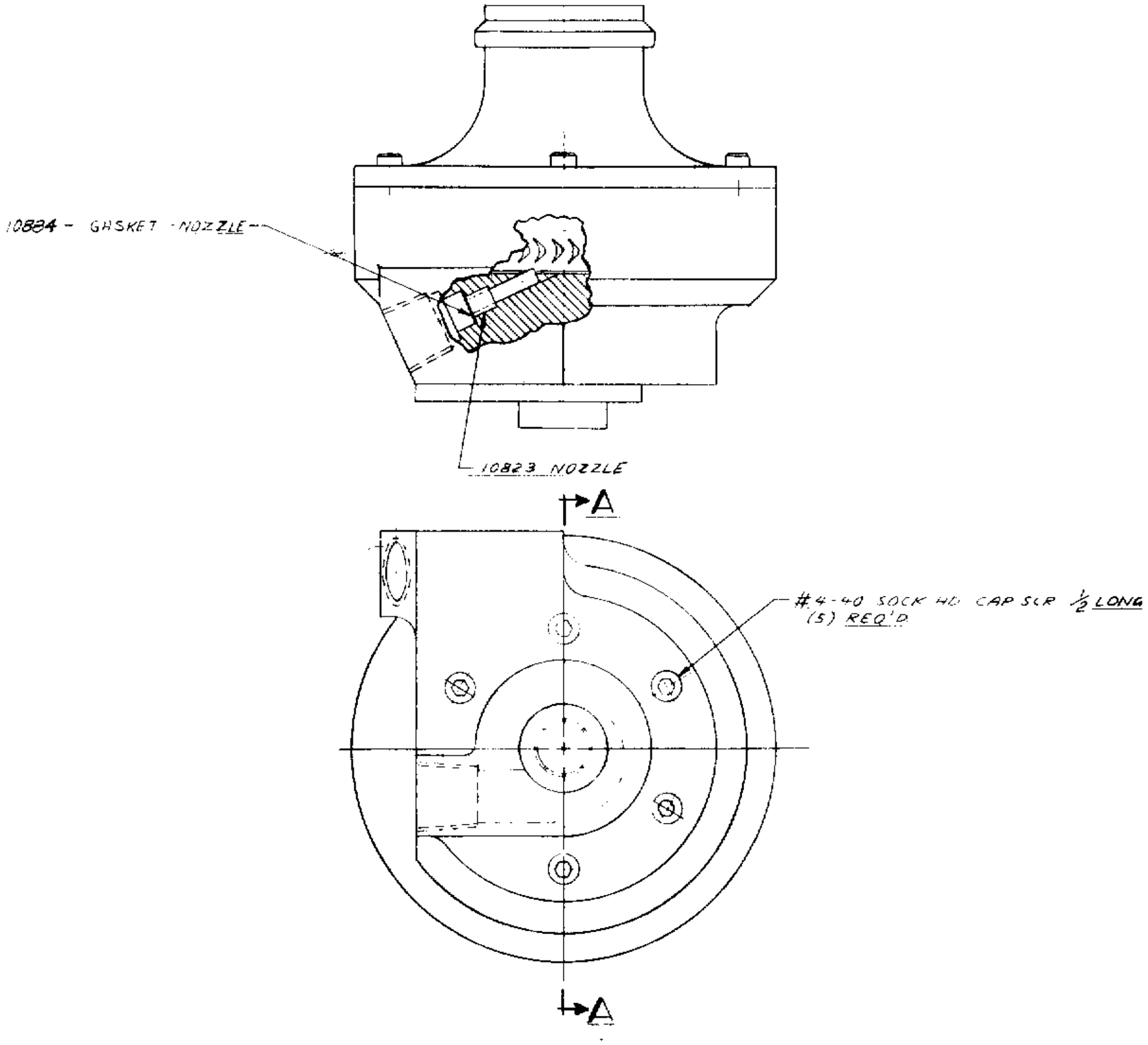


Figure 9

Type II Device Assembly

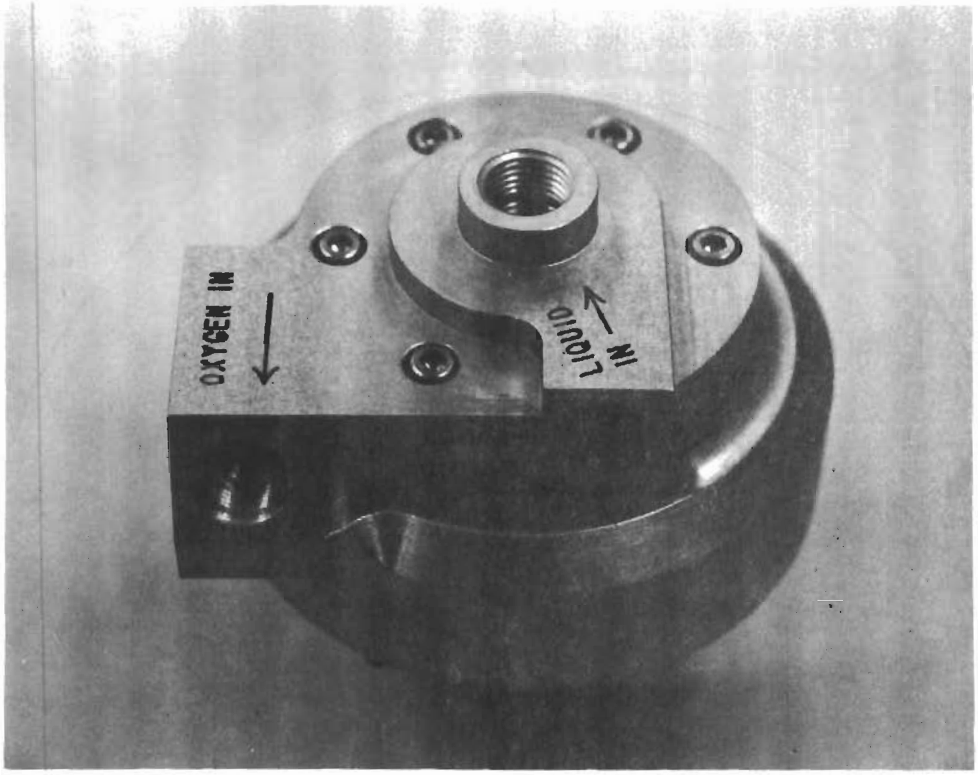


Figure 10. Type II Device

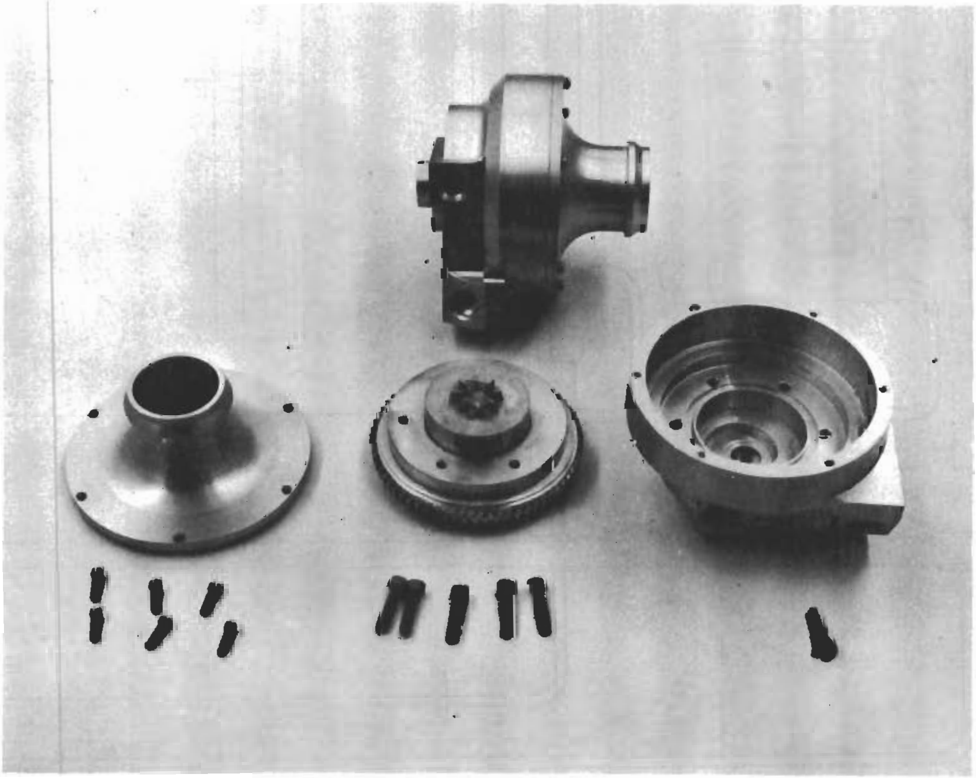
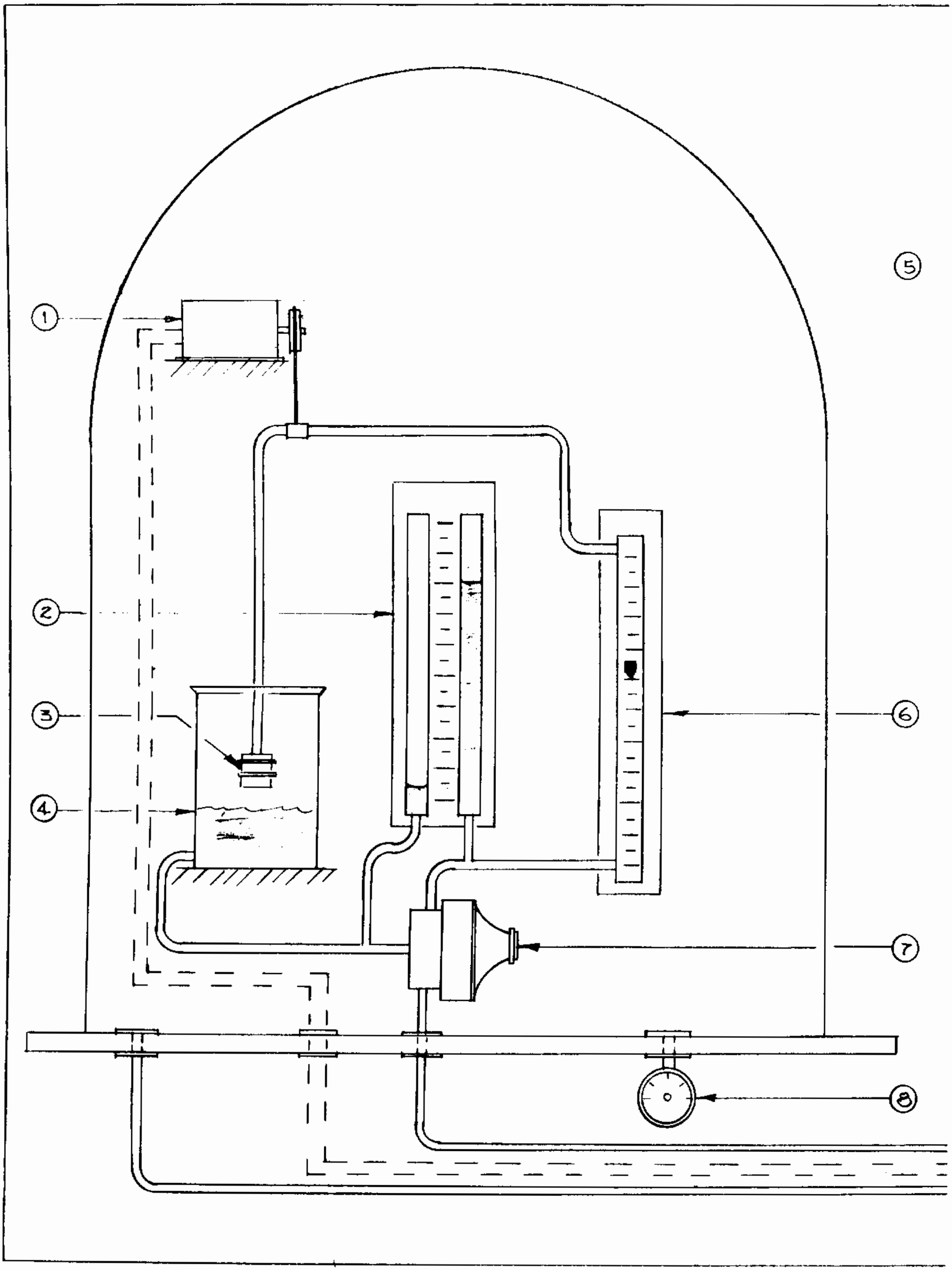


Figure 11. Type II Device Components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No	Description
1	Head Control Motor
2	Head Differential Manometer
3	Water Exit Nozzle
4	Water Resevoir
5	Bell Jar
6	Water FlowMeter
7	Type II Device
8	Absolute Pressure Gauge
9	Oxygen Drive Pressure Gauge
10	Oxygen FlowMeter
11	Oxygen Pressure Regulator
12	High Pressure Oxygen Bottle
13	Motor Control
14	Vacuum Pump

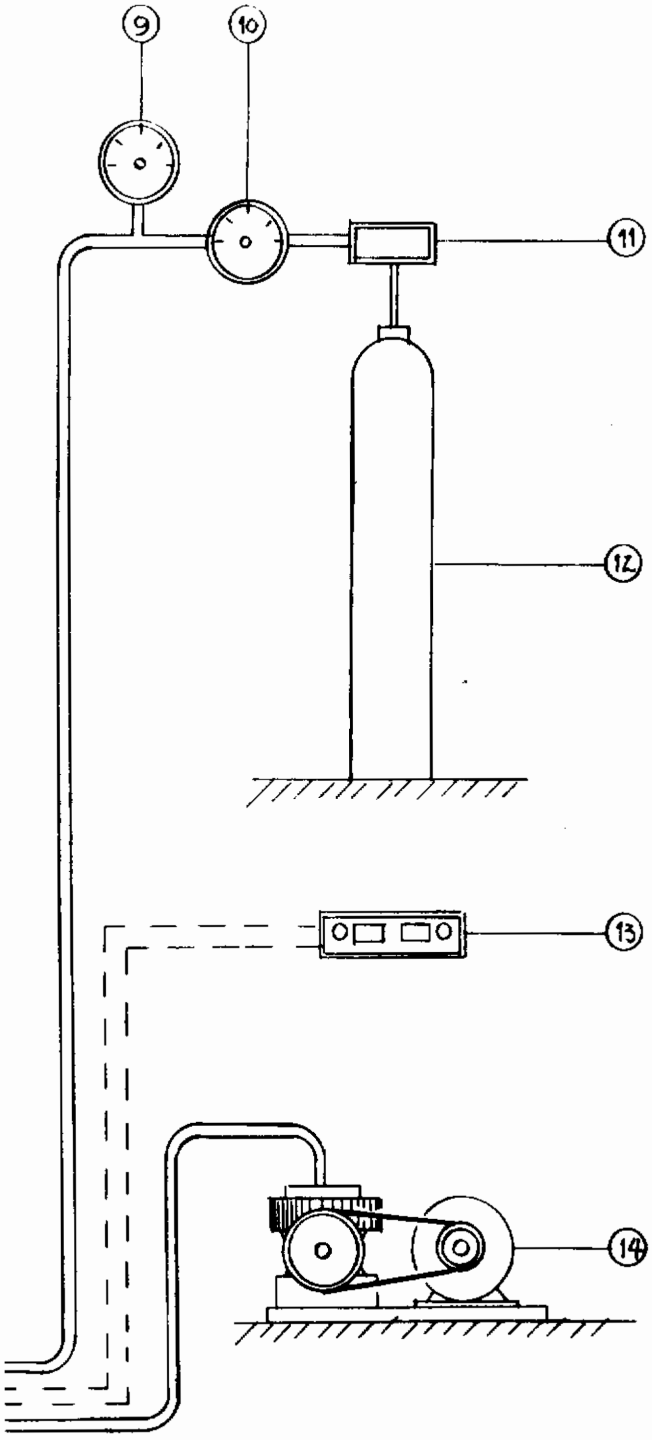

Figure 12

Type II Device

Test Apparatus Schematic

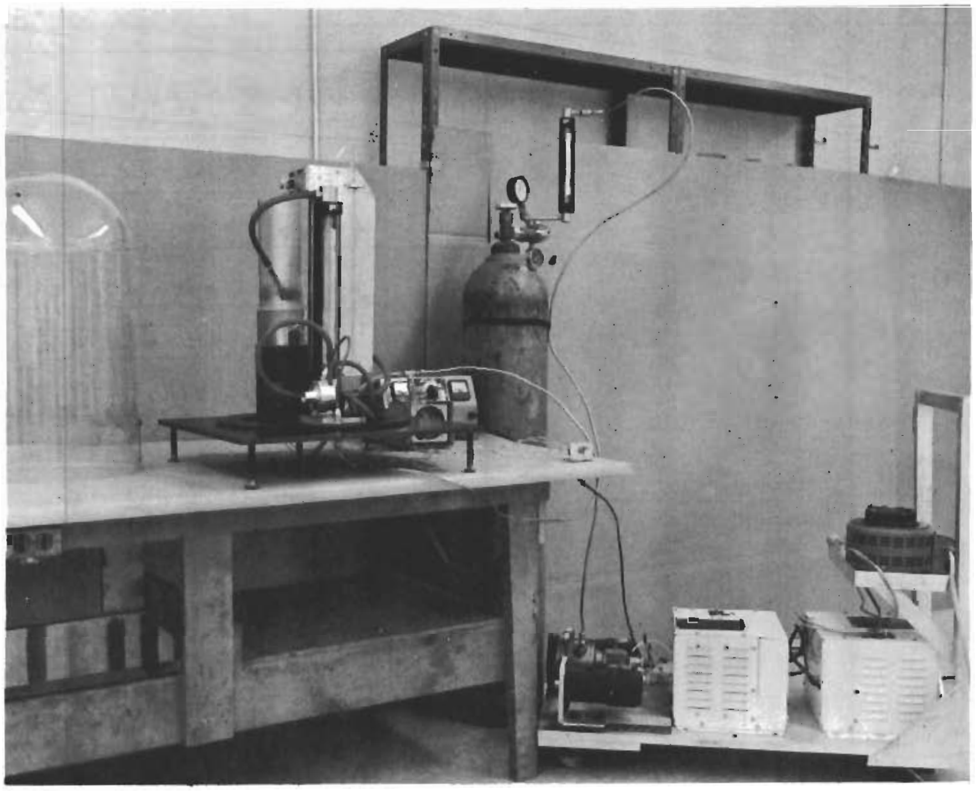

Figure 13. Type II Device Test Apparatus without Bell Jar

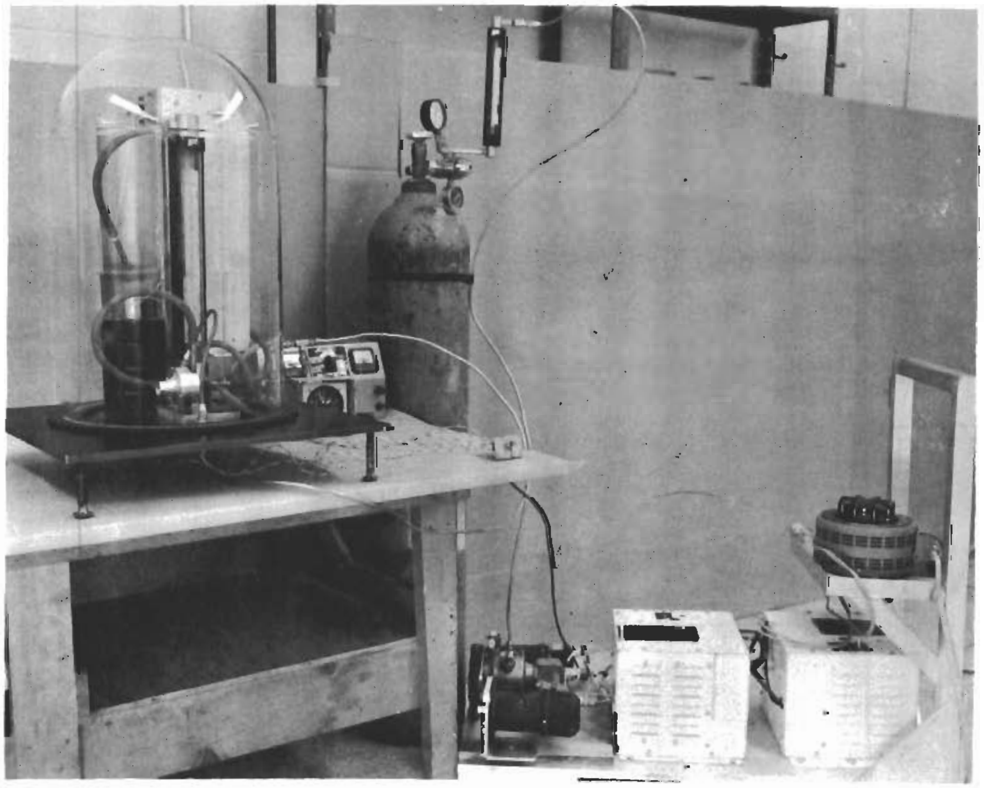

Figure 14. Type II Device Test Apparatus with Bell Jar

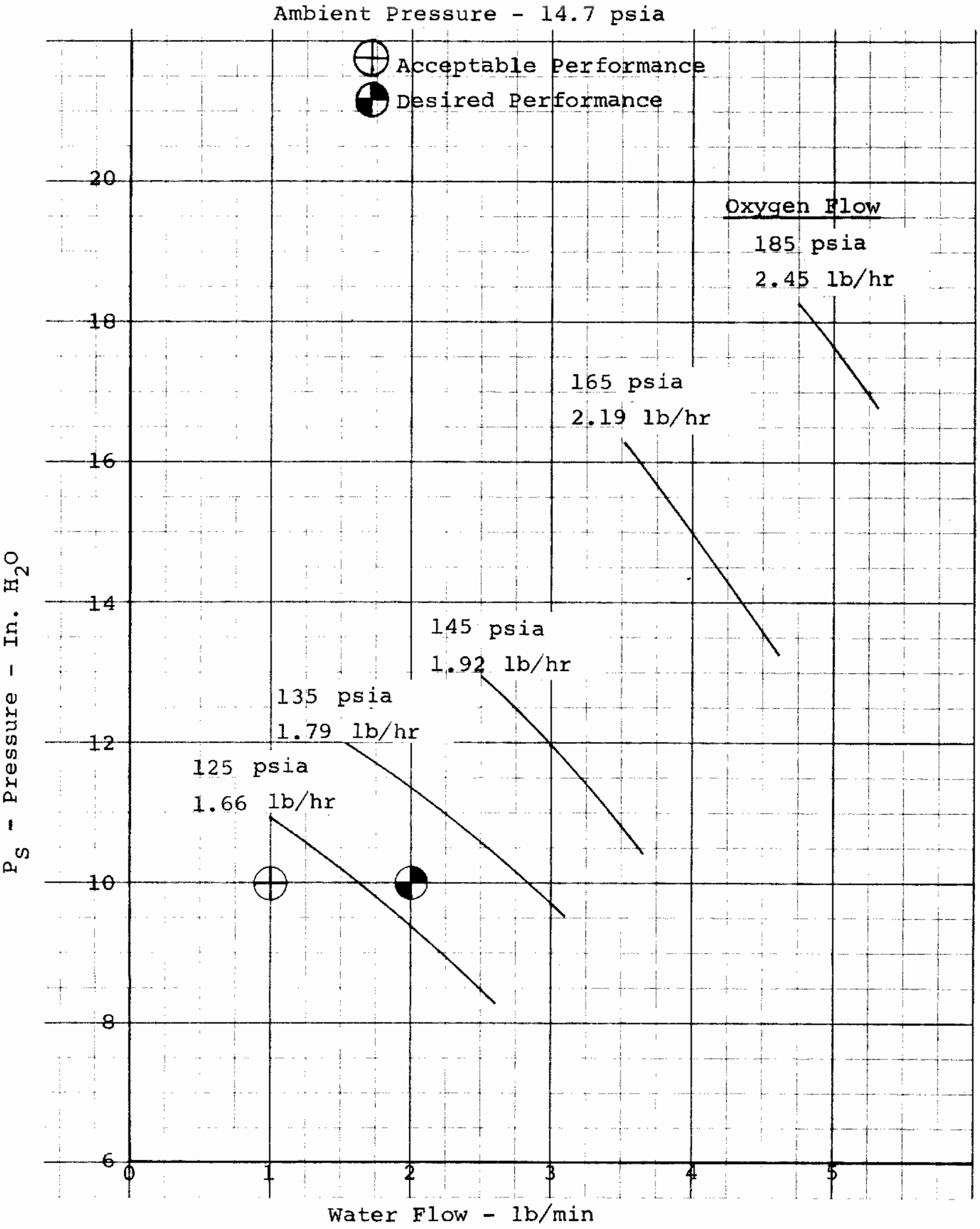

Figure 15

Type II Device Performance - Sea Level

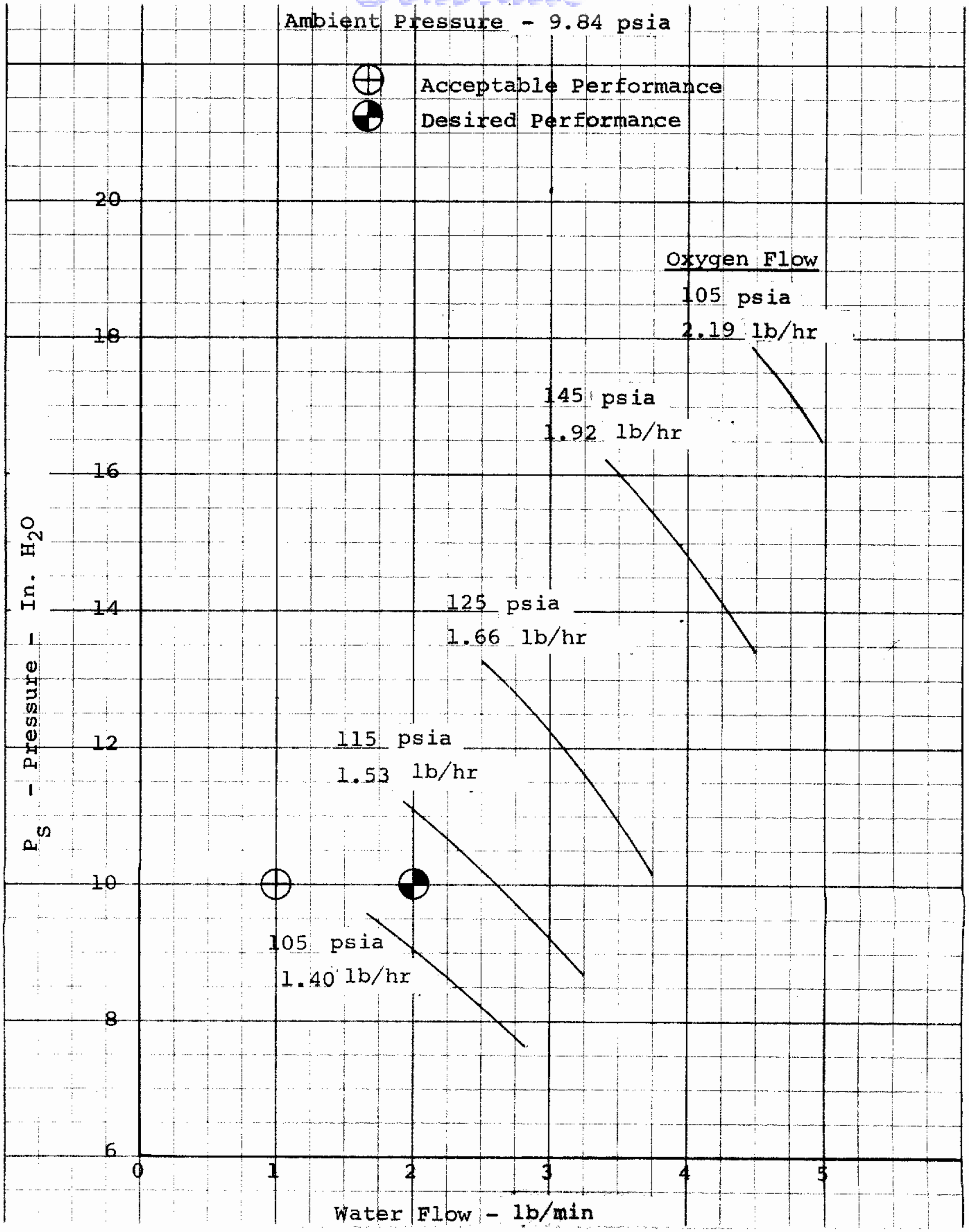

Figure 16

Type II Device Performance - 9.84 PSIA

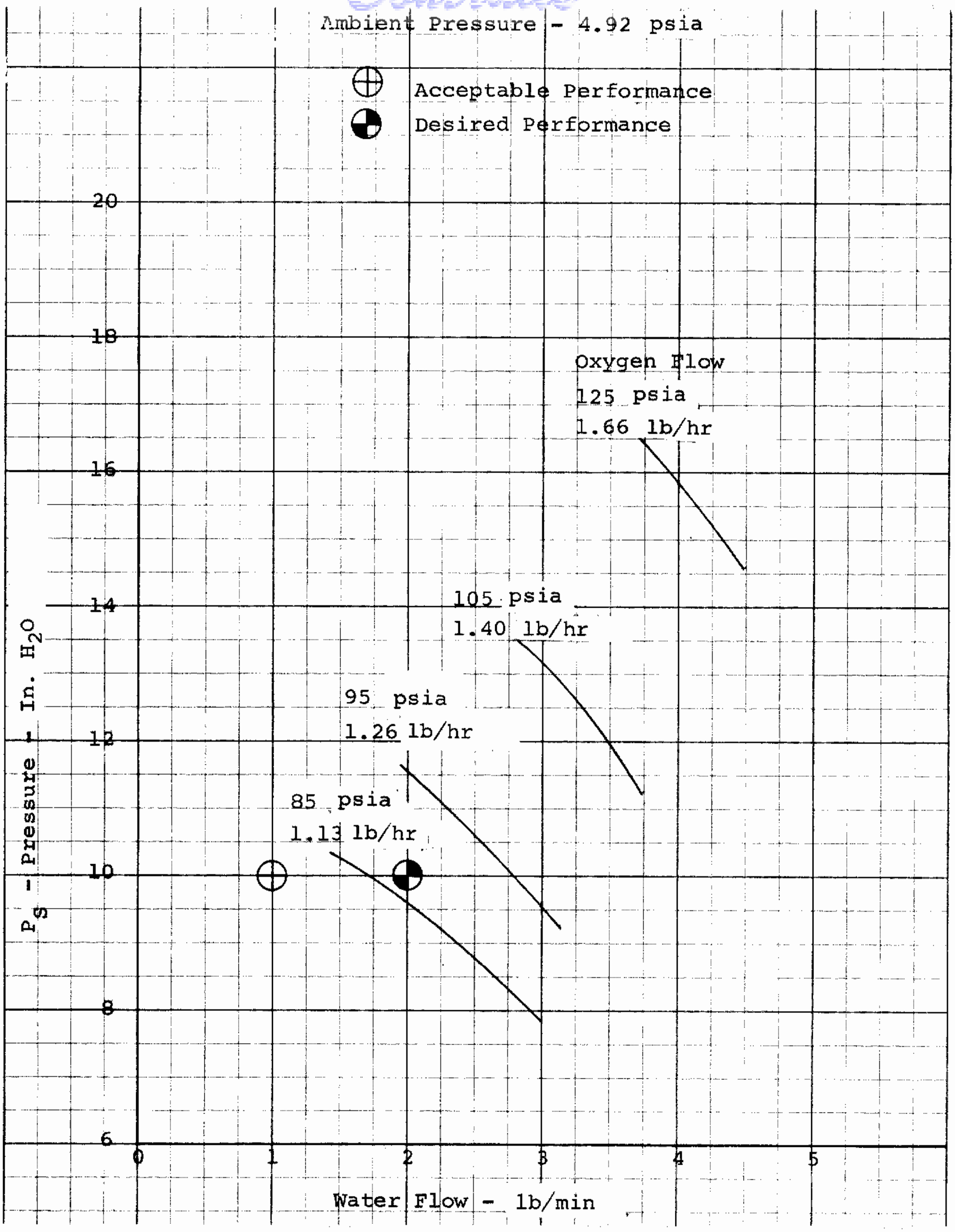

Figure 17

Type II Device Performance - 4.92 PSIA

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13 ABSTRACT

An analytical and experimental investigation was conducted to determine the feasibility of utilizing the energy available in breathing oxygen, when expanded from storage pressure to breathing pressure, to power two types of pumping devices; (1) an oxygen-circulating blower, and (2) a water-circulating pump. With an oxygen weight flow of 2.65 lb/hr and an ambient environmental pressure of 5 psia, the required performance of 5 cfm at 10 inches H_2O static pressure for the circulating blower and 1 lb/min at a back pressure of 10 inches H_2O for the water pump were met. Performance considerably in excess of the requirement was obtained for the water pump. Both units were powered by single-stage impulse turbines and weighed less than the 1-pound requirement. Both units operated reliably without mechanical malfunction and appear to be suited for manned aerospace enclosures.

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Life support						
Oxygen equipment						
Respiration						
Pressure breathing						
Impulse turbines						
Aeronautics						

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