AN INVESTIGATION OF THE EFFECTS OF THE AREA RATIO ON A LOW AREA RATIO CENTRAL TYPE AIR EJECTOR EMPLOYING MEDIUM TEMPERATURE LOW PRESSURE AIR AS THE DRIVING FLUID

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ACKNOWLEDGMENT

I wish to thank Professor W. A. Hinton for his endless cooperation and patience in the preparation of this thesis.

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TABLE OF SYMBOLS

A = Area, square inches

b, c = Calculated values in Keenan, Nuemann and Lustwerk equations

C_p = Specific heat at constant pressure, lb.-ft. per lb. per degree Rankine

C_v = Specific heat at constant volume, lb.-ft. per lb. per degree Rankine

g = Gravitational constant, ft. per sec.^{2}

h = Enthalpy, B.T.U. per lb.

k = $\frac{C_p}{C_v}$, dimensionless

p = pressure, lb. per square inch absolute

R = Gas constant

T = Temperature, degrees Rankine

V = Velocity, feet per second

v = Specific volume, cubic feet per pound

w = Mass flow rate, pounds per second

α = Ratio of area of secondary nozzle to area of primary nozzle, dimensionless

ω = Ratio of secondary flow to primary flow, dimensionless

Subscripts

0 Refers to initial conditions of secondary fluid (ambient conditions)

1 Refers to fluid conditions at entrance to mixing tube

TABLE OF SYMBOLS (Continued)

2	Refers to fluid conditions at end of mixing tube
i	Refers to initial conditions of primary fluid, i.e., at entrance to primary nozzle
x	Refers to fluid conditions at exit of primary nozzle

Superscripts

'	Refers to primary fluid
"	Refers to secondary fluid

SUMMARY

This investigation grew out of the need for information on the effects of the area ratio in the lower area ratio range for air ejectors. A great deal of work has been done on large area ratio ejectors, particularly those employing steam, but the ratios employed were of the order of 100:1. The ratios studied in this investigation are much lower, and ranged from 1.0:1 to 1.39:1. The data obtained include the maximum operating pressure, maximum flow ratio obtainable at each area ratio, and a comparison of the flow ratios at a predetermined constant pressure. In addition, the measured flows were compared to the calculated flows as obtained from the little literature available in similar investigations.

Results of the investigation point out several interesting trends. The first of these is that as the area ratio is increased, the predicted flow by calculation comes closer to that obtained by measurement. The principal weakness of the apparatus was found to be that the ejector would not operate in a satisfactory manner at high nozzle pressures and low area ratios. The maximum operating pressure rises rapidly with increasing area ratio. The apparatus which was employed yielded relatively low diffuser recovery pressures. It is felt that the primary and secondary nozzles and the diffuser cone exert a large influence on the recovery pressure, and that the surfaces of these components should be highly polished.

In the event of future investigations, the area ratio should be varied by changing the diameter of the primary nozzle. The diameter of the secondary nozzle was varied in these tests, but this practice results in changing very slightly the length of the mixing tube and the diffuser, and could result in changing the diffuser efficiency if the diameter of the mixing tube were considerably changed during an investigation.

CHAPTER I

INTRODUCTION

Much work on air ejectors has been done in the past by such men as Stodola, Bosnjakovic, and Flugel. Their achievements are being extended today by Goff and Coogan, Keenan, Nuemann, and Lustwerk, and other engineers and scientists. Attempts to analyze ejector performance have been made by applying the equations of continuity and momentum, the energy equation, and then applying simplifying assumptions in order to solve these equations.

Most of the experimental work done thus far has been performed using ejectors of relatively high area ratio, i.e., the ratio of the area of the secondary nozzle area to the area of the primary nozzle has been high. Low temperatures and pressures of the working fluid have also been employed in most investigations.

Data on ejectors of low area ratio are extremely scarce. The purpose of this investigation was to study the effect of the area ratio on the performance of low area ratio central type air ejectors.

The design of the test apparatus is similar to that used previously in central type ejectors. Assumptions of constant pressure mixing and one dimensional flow were made in order to limit the number of variables involved and thereby facilitate an approximate mathematical solution. The experimental work consisted of observing the effects of varying the area ratio while holding the inlet pressures and temperatures constant. The nozzle pressures employed were low and ranged from 5 to 75 inches mercury gage. The temperature of the primary fluid was of the order of 280 degrees centigrade.

CHAPTER II

EQUIPMENT AND INSTRUMENTATION

The test equipment was designed to admit air under moderate pressure and at a moderately high temperature to the ejector. A schematic diagram of the apparatus is shown in Figure 1.

The primary air was supplied from a reciprocating compressor through a line containing an oil trap to the inlet pressure regulating valve. The air then flowed through five heating tubes containing a total of ten 1000 watt electric heating elements which were connected in series with a Powerstat. The Powerstat is a commercial variable rheostat. The temperature of the primary air was controlled by manipulation of the rheostat. The primary air temperature was maintained at approximately 1000° Rankine by means of the equipment just described. After leaving the heaters, the air flowed through the primary nozzle, the secondary nozzle, past the discharge valve, and finally out to the atmosphere.

The secondary air entered a length of straight $1\frac{1}{4}$ inch pipe, flowed through a metering orifice to the mixing chamber and then into the secondary nozzle.

The ejector consisted of the primary nozzle, secondary nozzle, and the mixing chamber. The primary nozzle was fixed to the mixing chamber, but the secondary nozzle could be moved axially relative to the primary nozzle. Figure 2 shows the primary and secondary nozzles.

The primary air was metered by means of a standard flat plate orifice placed in the line before the heaters. The static pressure was measured with a mercury manometer or a calibrated bourdon gage when pressures exceeded the manometer range. The temperature of the air at the orifice was measured with a mercury thermometer. After the air passed through the heaters, a thermocouple measured the temperature before the air reached the primary nozzle. The pressure within the induction, or mixing, chamber and the pressure differential across the secondary metering orifice were measured with a water manometer.

Another mercury manometer and thermocouple measured the secondary discharge pressure and temperature. An aneroid barometer was employed to obtain the atmospheric pressure. The ambient temperature was observed using a standard mercury thermometer.

The orifices used conformed with A. S. M. E. standards. The heating tubes were well insulated to minimize heat loss and excessive power consumption. Increased turbulence of the primary air in the heaters was obtained to improve heat transfer by inserting wire screen and a network of 1" x 1" x 1/8" steel angles into the three inch diameter heating tubes.

The orifice which was designed to measure the secondary air flow was installed in a 1¼ inch pipe. This pipe size is rather small for an orifice installation, but should not produce an inaccuracy of greater than five per cent.[1] The flow computed from this orifice was compared

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[1]Stearns, R. F., R. M. Jackson, R. R. Johnson, and C. A. Larson; Flow Measurement with Orifice Meters, D. Van Nostrand Co., Inc., N. Y., 1951, p. 191.

with the flow measured with a Porter Flowrator. The Flowrator range was inadequate, but it was found that the results obtained by using the readings of the orifice were satisfactory.

The area ratio was varied by reaming the secondary nozzle to a larger diameter between runs. The true diameter of the nozzle is estimated to be within 0.002 inch of the recorded value.

CHAPTER III

PROCEDURE

The procedure employed in the test runs was as follows:

The primary air pressure regulating valve was set to the desired pressure after the compressor had built up full pressure in the air reservoir. The discharge valve was opened fully and the electric heaters turned on. The Powerstat was adjusted to the setting required to hold the primary air at the established test temperature. After the pressure and temperature had been regulated, the distance from the secondary nozzle to the primary nozzle was adjusted to obtain the maximum induced secondary air flow. This distance varied slightly from run to run, but it was not recorded since its effect was not being studied. Other investigators have supplied a large amount of data on the effect of distance between nozzles.

The static pressures and the temperatures of the primary and secondary air were recorded, as well as the orifice readings. The secondary nozzle discharge pressure was then increased by closing the discharge valve (See Fig. 1), and the readings listed above were again recorded. This procedure was continued until the discharge pressure reached the maximum value. At this pressure, the secondary flow was zero. The series of readings obtained by following this procedure constituted one run.

Before the next run, the secondary nozzle was removed, reamed to a larger diameter, reinstalled, and the next run was started.

A total of twelve runs were made, and the area ratio was varied from a minimum of 1:1 to a maximum of 1.39:1.

CHAPTER IV

DISCUSSION

The primary nozzle was designed to operate at the critical pressure assuming complete expansion and using a simple one-dimensional analysis. The nozzle diameter was limited by the compressor capacity and the heating capacity of the electric heating elements. The apparatus had a design capacity of five pounds of air per minute at 475° F.

The secondary nozzle mixing length was kept within the limits found advisable by Keenan, Nuemann, and Lustwerk.[1] The diffuser half-angle was 4.1°.

When the equipment was first tested, it was found that if the primary nozzle was operated at the critical pressure, air was forced out of the induction chamber rather than being drawn into the chamber. Accordingly, the primary pressure was reduced considerably in order to obtain an induced flow over a wide range of secondary nozzle exhaust pressures. Kastner and Spooner[2] found that this same condition existed in the smallest area-ratio ejector which they tested, which had an area

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[1]Keenan, J. H., E. P. Nuemann, and F. Lustwerk; "An Investigation of Ejector Design by Analysis and Experiment", Journal of Applied Mechanics, Trans. A. S. M. E., Vol. 17, 1950, pp. A-299-A-309.

[2]Kastner, L. J., and J. R. Spooner; "An Investigation of the Performance and Design of the Air Ejector Employing Low-Pressure Air as the Driving Fluid"; Proceedings, The Institution of Mech. Engr., Vol. 162, pp. 149-159.

ratio of 1.44:1. Wang[3], using an ejector with a ratio of 1.90:1 experienced the same difficulty.

Analysis.--It is assumed that the ideal nozzle relations hold, i.e.

$$V_x = 224\sqrt{h_i - h_1} \quad \text{(See Fig. 3)} \tag{1}$$

If we further assume that there is no external heat transfer, that all fluid characteristics are uniform across any cross section (one-dimensional flow assumption), and that all mixing is complete at section 1 of the secondary nozzle, the theoretical flow ratio, ω, the ratio of secondary mass flow to primary mass flow, may be computed.

The following equations may be written:

Continuity

$$\left.\frac{VA}{v}\right)_x = \left.\frac{VA}{v}\right)_1 \tag{2}$$

Energy

$$W\left(h + \frac{V^2}{2g}\right)_x = W\left(h + \frac{V^2}{2g}\right)_1 \tag{3}$$

Momentum

$$\left(\frac{W}{g}V\right)_x + P_xA_x = \left(\frac{W}{g}V\right)_1 + P_1A_1 \tag{4}$$

Perfect gas

$$pv = RT \tag{5}$$

$$h = C_p T \tag{6}$$

[3]Wang, D. I. J.; The Effects of Primary Temperature and Discharge Pressure on the Performance of an Air Ejector, Unpublished Master's Thesis, Georgia Institute of Technology, 1952.

Keenan, Nuemann, and Lustwerk[4] combined the above equations and assumed that processes i-1 and 0-1 were reversible adiabatic expansions to solve for the mixed stream velocity, V_1.

$$V_1 = -\frac{b}{2}\left(1 + \sqrt{1 - \frac{4c}{b^2}}\right) \tag{7}$$

where

$$b = \frac{-2g\ C_p\ (T_i - T_o) \quad \frac{2gk}{k-1}\ \frac{p_x A_1}{W'}\ V_x''}{V_x' - V_x'' + \frac{2gk}{k-1}\ \frac{p_x A_1}{W'}}$$

and

$$C = \frac{2g\ C_p\ (T_i\ V_x'' - T_o V_x')}{V_x' - V_x'' + \frac{2gk}{k-1}\ \frac{p_x A_1}{W'}}$$

By assuming that processes 0-1 and i-1 are reversible adiabatic, we may solve for V_x'' and V_x'. Using equation (1), we have

$$V_x' = 224\sqrt{h_1 - h_1} \tag{1}$$

which may be shown to be

$$V_x' = 224\sqrt{C_p\ T_i\left[1 - \left(\frac{p_x}{p_i}\right)^{\frac{k-1}{k}}\right]}$$

Also

$$V_x'' = 224\sqrt{C_p\ T_o\left[1 - \left(\frac{p_1}{p_o}\right)^{\frac{k-1}{k}}\right]}$$

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[4]Keenan, Loc. cit.

Having solved for V_1, we may now compute the flow ratio, ω, by applying the momentum equation between section x and section 1.

$$\frac{w'}{g} V_x' + p_x A_x + \frac{w''}{g} V_x'' = p_1 A_1 + \frac{(w' + w'')}{g} V_1$$

The difference of the pA terms is small in comparison to the other terms, so we drop them to obtain the approximation

$$V_x' + \omega V_x'' = (1 + \omega) V_1$$

or

$$\omega = \frac{V_1 - V_x'}{V_x'' - V_1}$$

The only unknown in the equation is now ω, so we may solve for it to obtain the approximation to the theoretical flow ratio under the previously prescribed conditions.

Results.--The results obtained in this investigation qualitatively follow those obtained by investigators who employed higher area ratio ejectors. Kastner,[5] *et al* found that the maximum flow ratio obtainable in the ejectors used in their investigation was given approximately by the equation

$$\omega = \sqrt{\alpha} - 1 \tag{8}$$

The flow ratios obtained in this investigation were considerably

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[5]Kastner, *op. cit.*, p. 154.

higher than those of Kastner. However, Wang[6] found that the primary temperature plays an important part in the resulting flow ratio. Kastner and Spooner employed primary and secondary air at the same temperatures, whereas in this investigation the primary air temperature was several hundred degrees above the secondary air temperature. Figure 4 shows the comparison of the maximum measured flow with the flow predicted by the Kastner and Spooner equation. The difference in the measured flow and the predicted flow is too large to be of any value.

Figure 5 shows curves of the measured flow at constant primary pressure and temperature versus the computed values using the Keenan method. The differences in the calculated and measured flow ratios become less significant as the area ratio increases when viewed on a percentage basis.

Figures 6, 7, and 8 show the relation between the flow ratio, and the diffuser gage pressure, p_D. These curves were plotted by using the area ratio, α , as the parameter. The family of curves show the same geometric shape and therefore the same qualitative tendencies, regardless of the area ratio. The maximum induced flow for each area ratio occurred at virtually zero diffuser pressure and ranged from about 6 per cent for the 1:1 ratio to about 37 per cent for the 1.39:1 area ratio.

Figure 9 shows the minimum pressure at which maximum flow could be obtained. Figure 9 also shows the maximum pressure at which the

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[6]Wang, op. cit., pp. 24-26.

diffuser could be operated. This maximum pressure is the nozzle pressure at which no flow took place through the primary metering orifice. At pressures above the curve, the ejector forces air out of the secondary orifice. At pressures below the curve, a flow into the ejector through the secondary orifice is induced. Figure 9 therefore gives the range of pressures at which the ejector can be operated for a given area ratio. Note that the maximum nozzle pressure increases very rapidly as the area ratio increases.

Figure 10 was derived from Figures 6, 7, and 8 and shows the flow ratio obtained at three different diffuser pressures for each of the area ratios used. Again, the curves are similar and show an increase in the induced flow as the diffuser pressure is decreased.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Conclusions.--It was pointed out in Chapter IV that the calculated performance of the ejector using the Kastner and Spooner equation does not agree closely with the measured quantities. The Kastner and Spooner equation is an empirical one which applied at slightly higher area ratios, and the primary and secondary fluids were at very nearly the same temperature. The fact that a large temperature difference existed in this investigation makes the Kastner and Spooner equation almost valueless for purposes of comparison.

Figure 5 shows that the flow ratio calculated by the Keenan equation is about seven per cent below the measured flow ratio. This value would be quite acceptable for rough design calculations. However, the ejector would not operate satisfactorily except at very low primary nozzle pressures. This fault seems to be the weakest link in the calculated design of the ejector. The maximum operating pressure of the ejector rises very rapidly with increasing area ratio, as Figure 9 shows.

The recovery pressure in the diffuser was low and would seriously limit the application of similar ejectors. Figures 6 through 8 show how severely the diffuser pressure affects the flow ratio.

Recommendations.--The principal difficulty experienced in these tests was with the instrumentation, particularly with the measurement of the secondary flow. Moreover, when an orifice is placed in the secondary line, the secondary flow is changed due to the increased losses at the orifice. This flow should be measured with more accurate means than was available for this investigation.

In this thesis, the area ratio was varied by varying the diameter of the secondary nozzle. Changing the secondary nozzle diameter changed the diffuser efficiency slightly and changed the length of the mixing tube of the secondary nozzle. If more tests are made on a similar design, I recommend that the primary nozzle diameter be varied while the secondary diameter remains constant.

The surfaces of the primary nozzle, secondary nozzle, and the diffuser section should be finished with the greatest care. It is felt that this factor is of particular importance in cases where the diameters are small, and that a greater recovery pressure could be obtained in the equipment used with a polished diffuser cone.

APPENDIX I

FIGURES

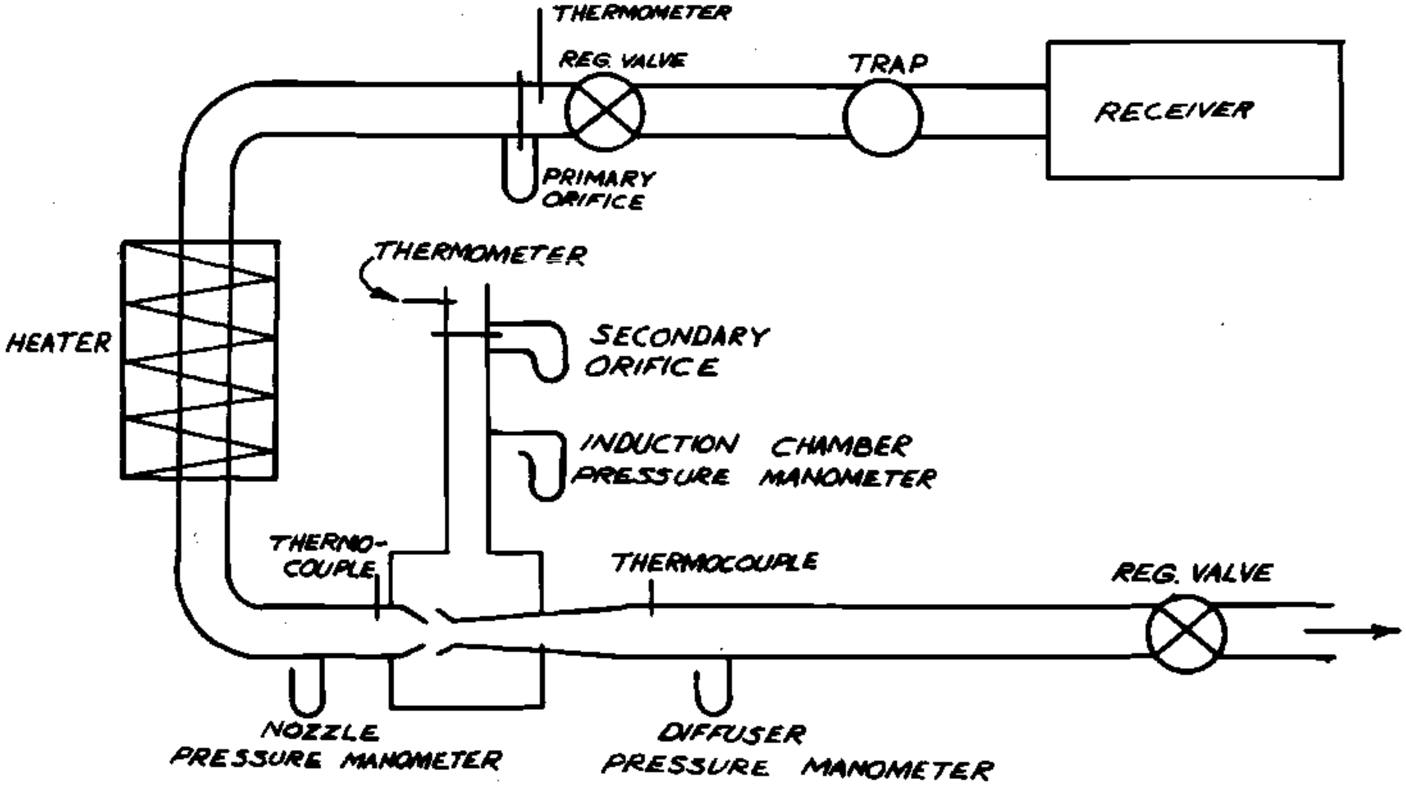

Figure 1. Schematic Diagram of Apparatus

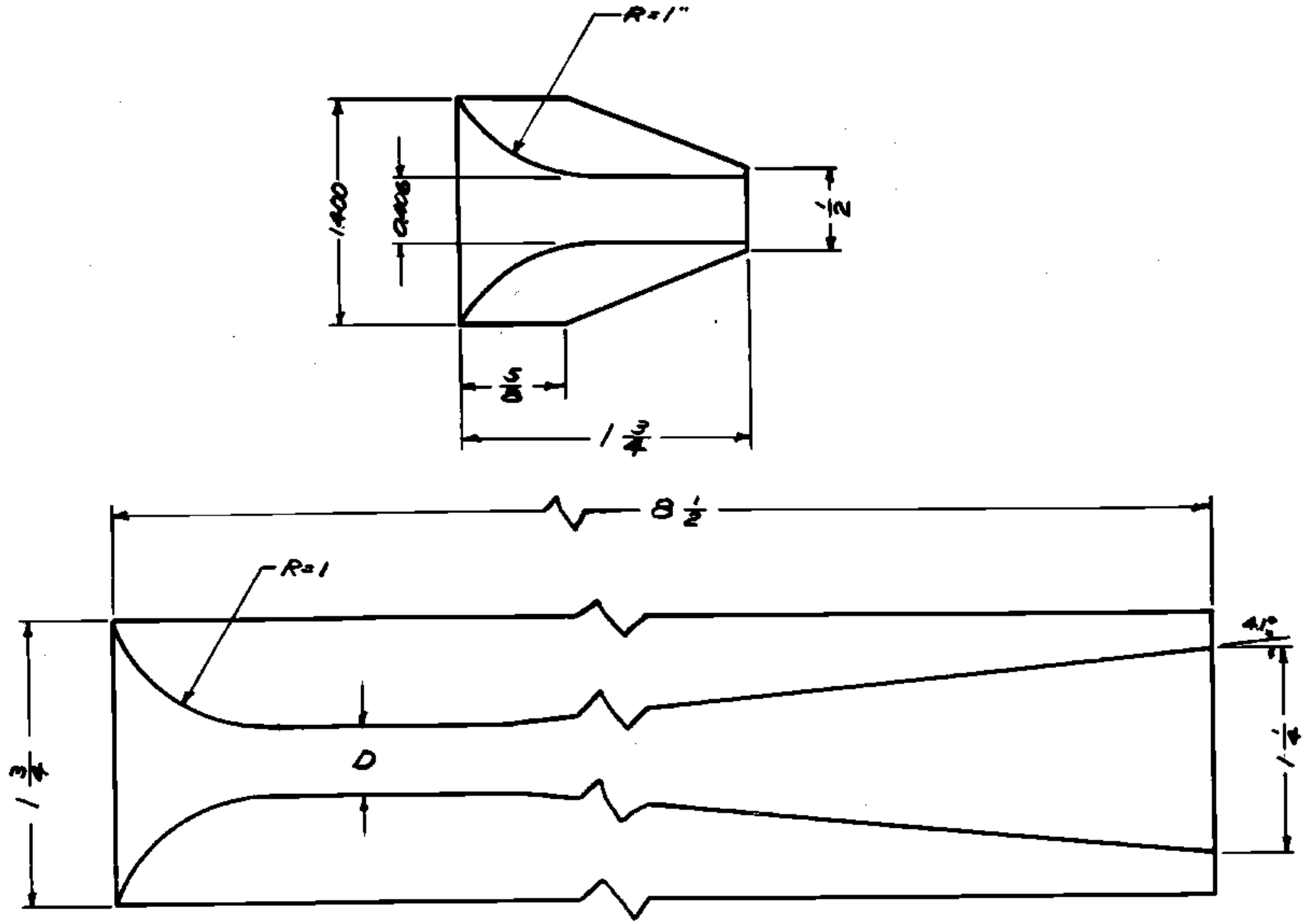

Figure 2. Nozzles and Diffuser

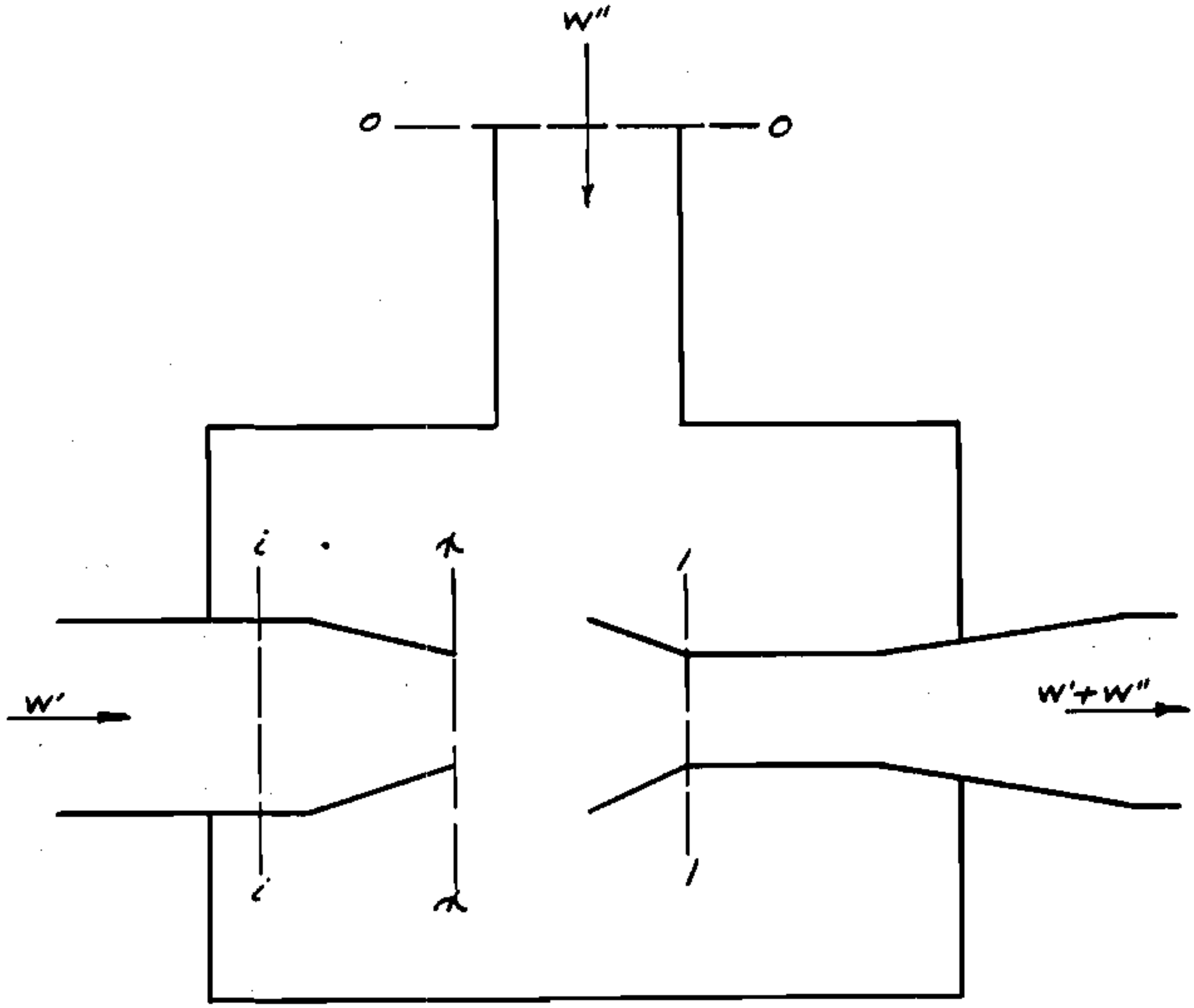

Figure 3. Induction Chamber

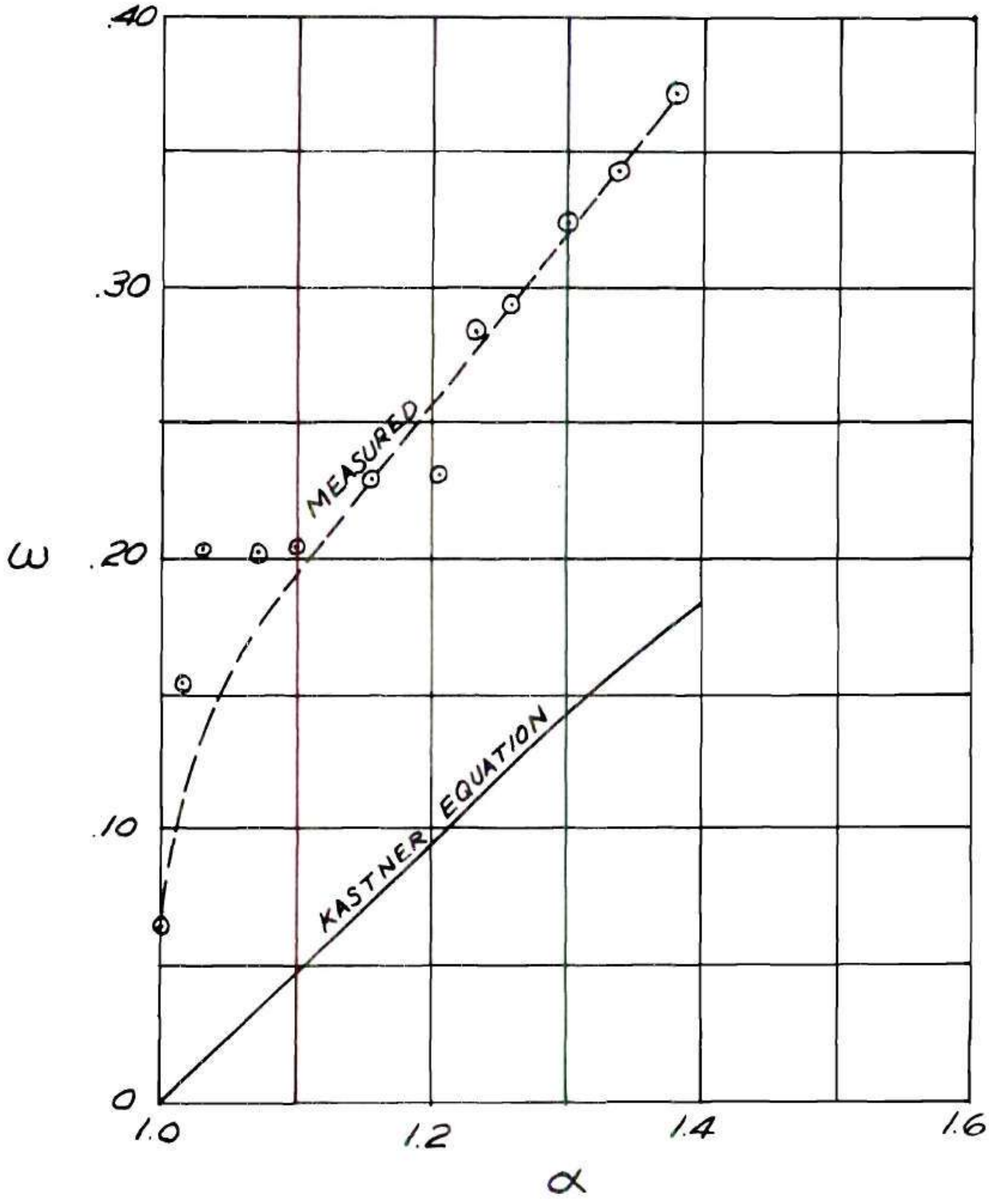

Figure 4. Comparison of Maximum Flow with Flow Calculated by Kastner Equation

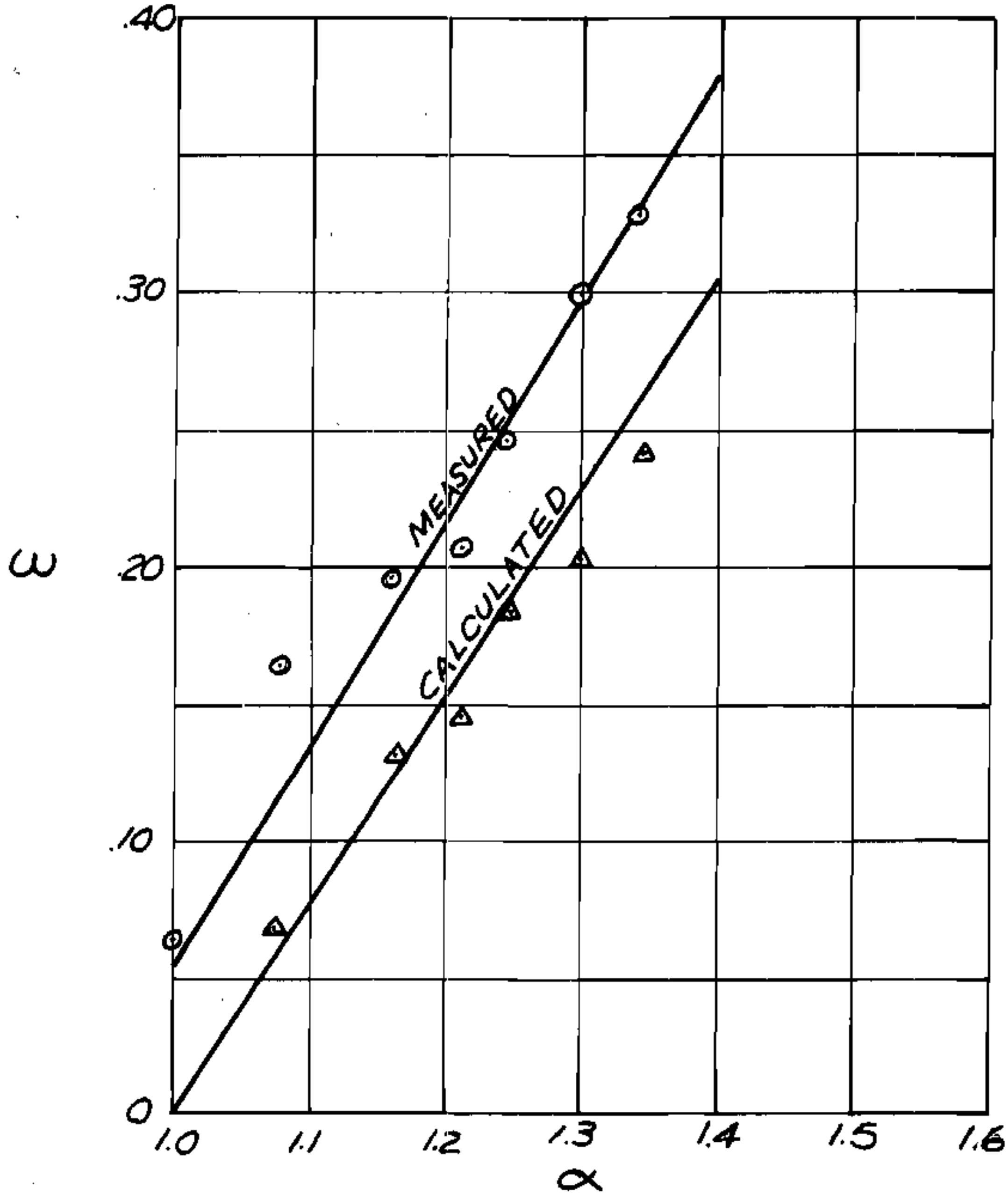

Figure 5. Comparison of Measured Flow with Flow Calculated by Modified Keenan Method

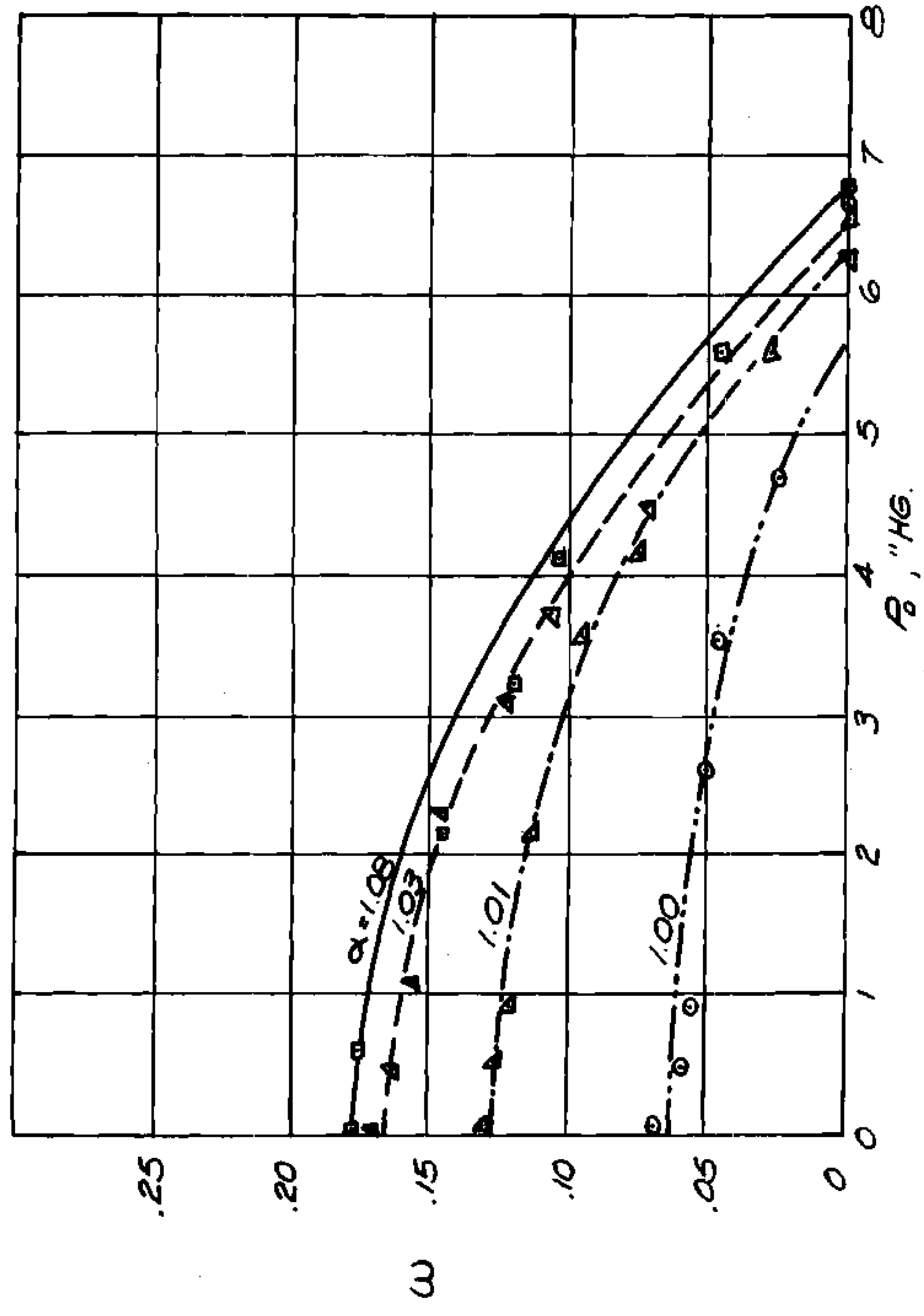

Figure 6. Performance at Constant Area Ratios

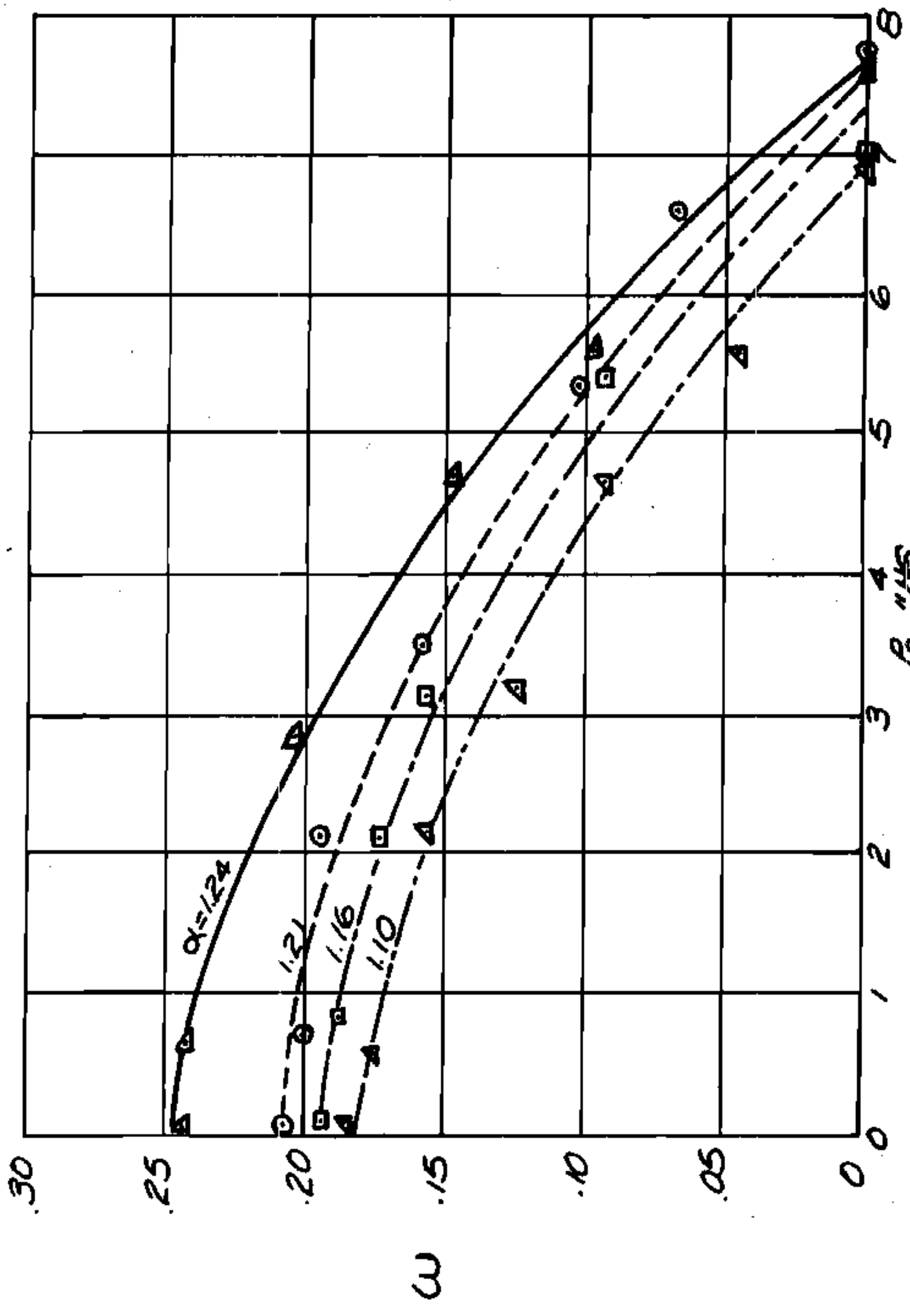

Figure 7. Performance at Constant Area Ratios

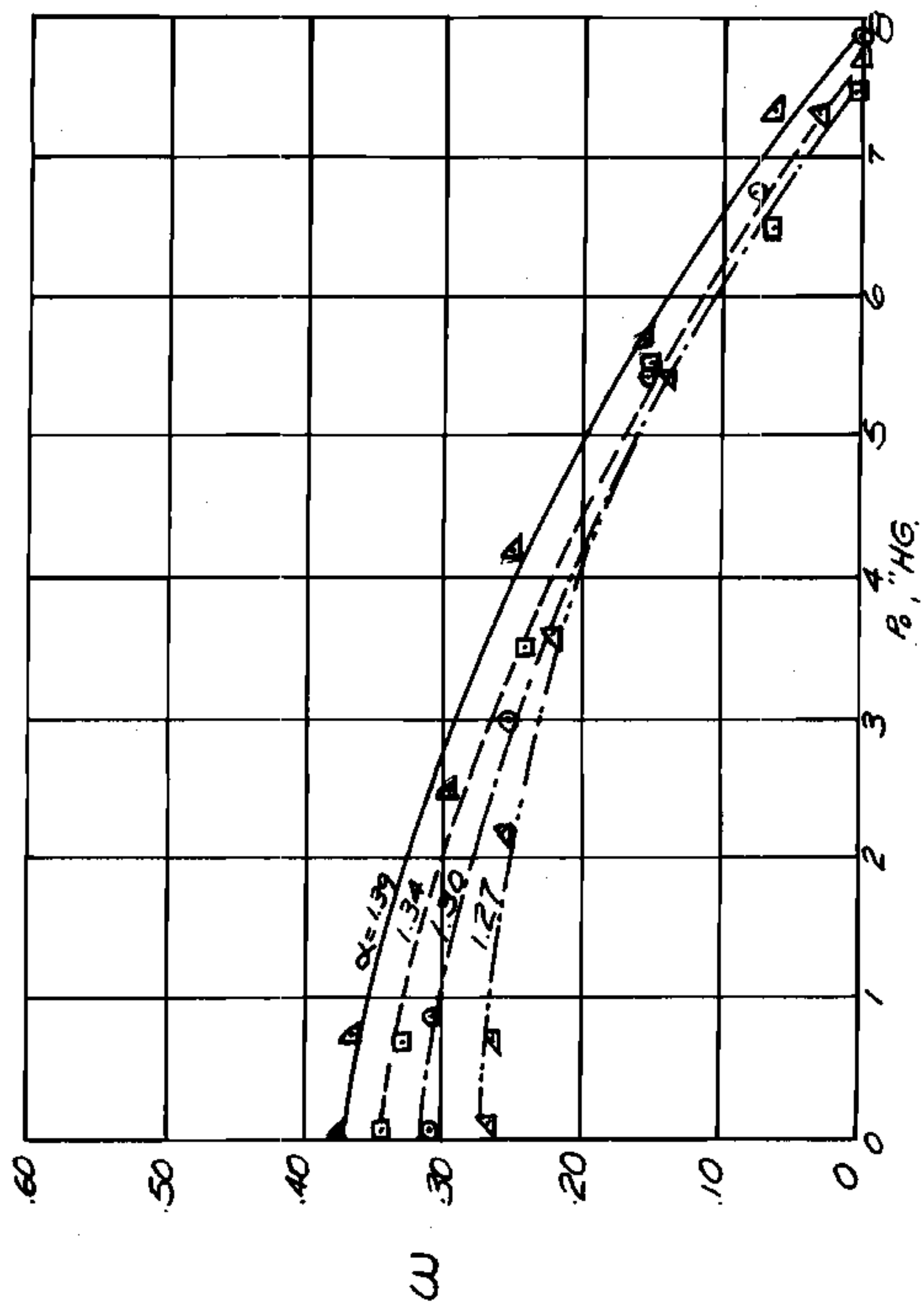

Figure 8. Performance at Constant Area Ratios

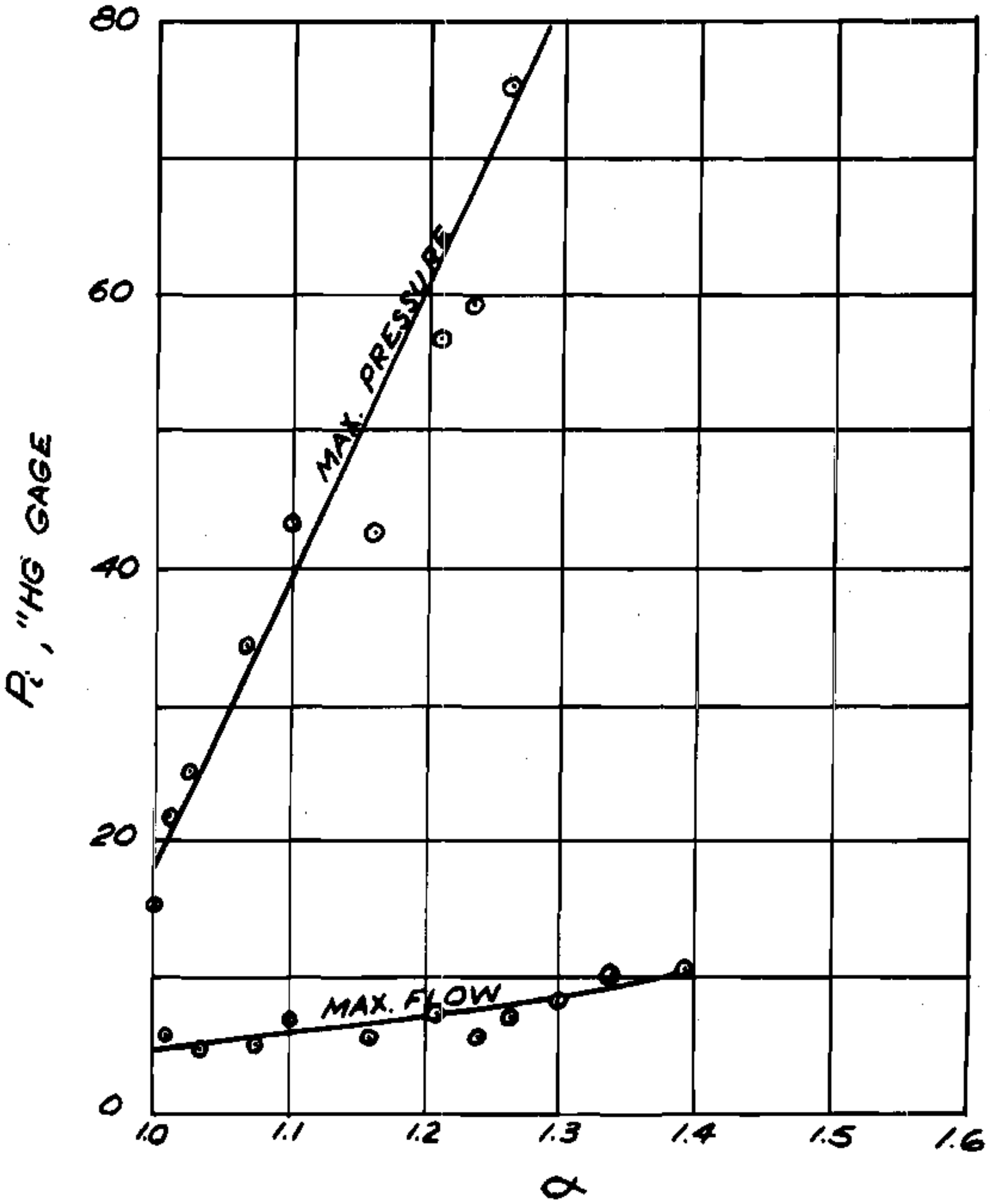

Figure 9. Maximum Operating Pressures and Maximum Flow Pressures

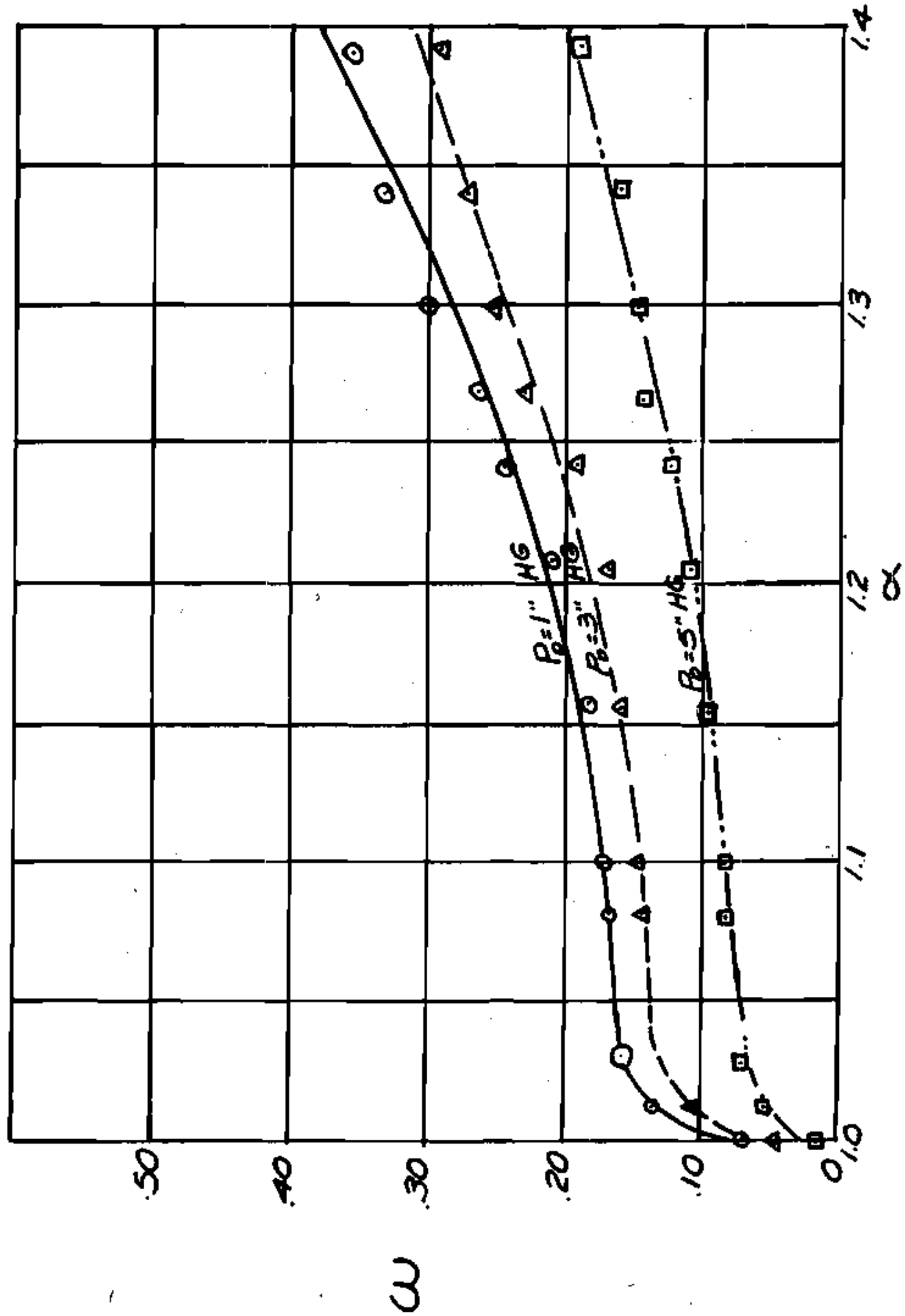

Figure 10. Performance at Constant Diffuser Pressures

APPENDIX II

DATA

Table 1. Data, α = 1.00

Primary Diameter = 0.405"
Secondary Diameter = 0.405"
Primary Air Flow = 2.28 Pounds per Minute

Barometer = 28.96" Hg
Ambient Temp. = 82° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.152	10.1	277	0.1	269	0.7
0.135	11.1	278	0.5	269	0.5
0.133	10.2	278	0.9	270	0.5
0.114	10.9	281	2.6	273	0.4
0.103	10.9	281	3.6	278	0.3
0.060	11.4	287	4.6	285	0.1
0	11.4	287	5.6	287	0
0	15.0	282	0.1	281	0

Table 2. Data, $\alpha = 1.01$

Primary Diameter = 0.405"
Secondary Diameter = 0.410"
Primary Air Flow = 2.28 Pounds Per Minute

Barometer = 28.96" Hg
Ambient Temp. = 84° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.291	10.8	282	0.1	264	2.4
0.286	10.5	282	0.5	264	2.3
0.273	10.9	283	0.9	264	2.1
0.272	11.2	283	2.2	264	2.1
0.207	10.6	283	3.6	268	1.2
0.172	10.4	283	4.1	273	0.8
0.059	11.2	283	5.7	280	0.1
0	11.2	284	6.5	284	0
0.354	5.9	280	0.2	252	4.5
	51.0	282	0.2	282	0

Table 3. Data, $\alpha = 1.03$

Primary Diameter = 0.405"
Secondary Diameter = 0.415"
Primary Air Flow = 2.26 Pounds Per Minute

Barometer = 29.20
Ambient Temp. = 78° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.359	9.6	275	0.1	250	3.6
0.347	10.3	277	0.5	252	3.4
0.338	10.0	279	1.1	255	3.2
0.310	9.8	279	2.3	259	2.7
0.261	10.3	281	3.1	264	1.9
0.233	10.2	282	3.7	268	1.5
0.148	10.0	284	4.4	273	0.6
0	10.6	285	6.2	283	0
0.447	4.3	284	0.1	246	5.6
0	25.2	285	0.1	283	0

Table 4. Data, $\alpha = 1.08$

Primary Diameter = 0.405"
Secondary Diameter = 0.420"
Primary Air Flow = 2.27 Pounds Per Minute

Barometer = 29.20
Ambient Temp. = 82° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.385	11.0	280	0.1	258	4.2
0.359	11.0	280	0.7	257	3.9
0.331	10.4	280	2.1	258	3.1
0.279	10.8	283	3.2	261	2.2
0.232	11.0	283	4.2	267	1.5
0.087	11.0	282	5.6	277	0.2
0	11.0	282	6.8	281	0
0	34.1	282	0.2	281	0
0.466	4.6	280	0.1	245	6.2

Table 5. Data, $\alpha = 1.10$

Primary Diameter = 0.405"
Secondary Diameter = 0.425"
Primary Air Flow = 2.27 Pounds Per Minute

Barometer = 29.20
Ambient Temp. = 83° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.422	10.8	280	0.1	250	5.1
0.401	10.9	280	0.7	254	4.6
0.356	11.0	283	2.2	259	3.6
0.286	11.0	284	3.2	264	2.3
0.199	11.5	284	4.7	269	1.1
0.105	10.4	285	5.7	280	0.3
0	11.5	286	6.9	286	0
0	42.9	270	0.3	268	0
0.467	7.6	267	0.1	237	6.2

Table 6. Data, $\alpha = 1.16$

Primary Diameter = 0.405
Secondary Diameter = 0.437
Primary Air Flow = 2.28 Pounds Per Minute

Barometer = 29.2
Ambient Temp. = 82° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.439	11.5	272	0.1	242	5.5
0.427	11.4	275	0.9	247	5.2
0.389	12.2	276	2.2	253	4.3
0.357	11.9	277	3.2	256	3.6
0.208	11.9	279	5.4	268	1.2
0	11.7	280	7.0	279	0
0	42.4	280	0.2	276	0
0.525	5.4	276	0.1	240	7.9

Table 7. Data, $\alpha = 1.21$

Primary Diameter = 0.405"
Secondary Diameter = 0.445"
Primary Air Flow = 2.38 Pounds Per Minute

Barometer = 29.32" Hg
Ambient Temp. = 80° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.493	11.4	284	0.1	252	6.9
0.476	11.4	284	0.8	253	6.4
0.433	11.0	284	2.1	255	5.3
0.376	12.0	283	3.5	258	4.0
0.247	11.4	283	5.3	265	1.7
0.159	12.0	284	6.7	273	0.7
0	11.8	284	7.7	282	0
0	57.3	283	0.5	282	0
0.543	8.4	275	0.2	240	8.3

Table 8. Data, $\alpha = 1.24$

Primary Diameter = 0.405"
Secondary Diameter = 0.451"
Primary Air Flow = 2.31 Pounds Per Minute

Barometer = 29.32" Hg
Ambient Temp. = 79

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.573	11.6	276	0.1	243	9.3
0.564	11.4	280	0.7	244	9.0
0.469	11.8	280	2.9	248	6.2
0.343	11.3	278	4.6	255	3.3
0.217	11.3	278	5.6	268	1.3
0	11.8	280	7.6	277	0
0.643	6.1	280	0.1	243	11.0
0	58.8	282	0.5	281	0

Table 9. Data, $\alpha = 1.27$

Primary Diameter = 0.405"
Secondary Diameter = 0.457"
Primary Air Flow = 2.29 Pounds Per Minute

Barometer = 29.32" Hg
Ambient Temp. = 80° F.

Secondary Air Flow, Lb./Min.	Nozzle		Diffuser		Chamber
	Press "Hg	Temp. °C	Press "Hg	Temp. °C	Vacuum "H_2O
0.623	11.2	282	0.1	236	11.0
0.620	11.2	282	0.8	240	10.9
0.591	11.6	282	2.2	243	9.1
0.504	11.6	283	3.6	249	7.2
0.332	11.6	283	5.4	260	3.1
0.062	11.6	285	7.4	278	0.1
0	12.2	285	9.2	283	0
0.610	7.8	287	0.1	247	12.7
0	75.0	272	0.6	263	0

Table 10. Data, $\alpha = 1.30$

Primary Diameter = 0.405"
Secondary Diameter = 0.461"
Primary Air Flow = 2.30 Pounds Per Minute

Barometer = 29.28" Hg
Ambient Temp. = 81° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.701	11.3	277	0.1	232	14.0
0.701	11.3	278	0.9	230	14.0
0.575	11.1	277	3.0	235	9.4
0.358	11.8	277	5.4	247	3.6
0.191	11.7	280	6.8	265	1.0
0	11.1	280	7.9	274	0
0.747	8.5	280	0.1	240	15.8
0	7.4	250	0.6	248	0

Table 11. Data, $\alpha = 1.34$

Primary Diameter = 0.405"
Secondary Diameter = 0.469"
Primary Air Flow = 2.27 Pounds Per Minute

Barometer = 29.28" Hg
Ambient Temp. = 81° F.

Secondary Air Flow, Lb./Min.	Nozzle Press "Hg	Nozzle Temp. °C	Diffuser Press "Hg	Diffuser Temp. °C	Chamber Vacuum "H_2O
0.763	10.2	278	0.1	227	16.6
0.734	10.8	279	0.8	229	15.4
0.530	11.1	279	3.5	240	8.0
0.341	11.2	280	5.5	253	3.3
0.160	10.9	280	6.4	268	0.7
0	10.9	283	7.4	280	0

Table 12. Data, α = 1.39

Primary Diameter = 0.405"
Secondary Diameter = 0.478"
Primary Air Flow = 2.28 Pounds Per Minute

Barometer = 29.28" Hg
Ambient Temp. = 84° F.

Secondary Air Flow, Lb./Min.	Nozzle		Diffuser		Chamber
	Press "Hg	Temp. °C	Press "Hg	Temp. °C	Vacuum "H_2O
0.852	10.7	282	0.2	226	20.7
0.817	11.1	283	0.8	228	19.1
0.678	10.6	285	2.5	233	13.1
0.558	11.1	285	4.2	244	8.9
0.331	10.7	285	5.7	262	3.1
0.136	11.7	285	7.4	274	0.5
0	11.9	285	7.8	284	0

APPENDIX III

SAMPLE CALCULATIONS

SAMPLE CALCULATIONS

Data from Table 8, Run 1

In order to calculate the flow ratio using the Keenan method, the following data must be known:

T_i, the primary air temperature just before entering the primary nozzle.

T_o, the secondary air inlet temperature, in this case the ambient air temperature.

P_o, the atmospheric pressure.

P_i, the primary air pressure.

A_1, the mixing tube area in the secondary nozzle.

P_x, the pressure at the exit of the primary nozzle, ($P_x = P_1$).

W', the mass flow rate of primary air.

$T_i = 276° \text{ C} = 990° \text{ R}$

$T_o = 539° \text{ R}$

$P_o = 29.32" \text{ Hg} = 14.40 \text{ lb/in}^2$

$P_i = 29.32" \text{ Hg} \quad 11.6" \text{ Hg} = 20.10 \text{ lb/in}^2$

$P_x = P_1 = 29.32" \text{ Hg} - 9.3 \text{ "}H_2O = 14.06 \text{ lb/in}^2$

$A_1 = (\pi/4)(.451)^2 = 0.160 \text{ in}^2$

$W' = 0.0383 \text{ lb/sec}$

$$V_x' = 224 \sqrt{C_p \; T_i \left[1 - \left(\frac{P_x}{P_i}\right)^{\frac{k-1}{k}}\right]}$$

$$= 224 \sqrt{(0.24)(990) \left[1 - \left(\frac{14.06}{20.10}\right)^{\frac{1.4-1}{1.4}}\right]} = 1072 \text{ ft/sec}$$

$$b = \frac{-2g\,C_p\,(T_i - T_o) + \frac{2gk}{k-1}\frac{p_x A_1}{W'} V''_x}{V'_x - V''_x + \frac{2gk}{k-1}\,\frac{p_x A_1}{W'}}$$

$$b = -\frac{(2)(32.2)(187)(990-539) + 0}{1072-0 + \frac{(2)(32.2)(1.4)}{0.4}\,\frac{(14.06)(0.16)}{0.0383}}$$

V''_x, the velocity of the secondary air in the axial direction is zero at section x. (See Figure 3)

$$b = -\frac{(12{,}030)(451)}{1072 + (225)(58.8)} = -379$$

$$c = \frac{2g\;C_p(T_i V''_x - T_o V'_x)}{V'_x - V''_x + \frac{2gk}{k-1}\quad\frac{P_x A_1}{W'}}$$

$$c = \frac{(2)(32.2)(187)\;\left\{(990)(0)-(539)(1072)\right\}}{1072-0 + \frac{(2)(32.2)(1.4)}{0.4}\,\frac{(14.06)(0.160)}{0.0383}}$$

$$c = -488{,}000$$

$$V_1 = -\frac{b}{2}\left(1 + \sqrt{1 - \frac{4c}{b^2}}\right)$$

$$= 189.5\left(1 + \sqrt{1 + \frac{4(488{,}000)}{(379)^2}}\right) = 910 \text{ ft/sec}$$

Applying the momentum equation, we have

$$1072 = 910\;(1 + \omega\;)$$

$$\omega = \frac{162}{910} = 17.8\%$$

The measured flow for this run was found to be approximately 24%.

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