EXPERIMENTAL INVESTIGATION OF VORTEX REFRIGERATION

by

Robert J. Corless

Robert L. Solnick

Submitted in Partial Fulfillment of the Requirements
for the Bachelor of Science Degree in
Mechanical Engineering
from the
Massachusetts Institute of Technology
September, 1947

Signature redacted

Signature redacted

Acceptance:

Instructor in Charge of Thesis: Signature redacted

Dr. Joseph Kaye

September 12, 1947

Massachusetts Institute of Technology
Cambridge 39, Massachusetts
September 12, 1947

Professor J. S. Newell
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Professor Newell:

We hereby submit the enclosed thesis entitled, "Experimental Investigation of Vortex Refrigeration," as partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering from the Massachusetts Institute of Technology.

Very truly yours,

Signature redacted

Robert J. Corless

Signature redacted

Robert L. Solnick

ACKNOWLEDGMENT

The authors wish to thank Dr. Joseph Kaye for his valuable assistance during the preparation and completion of this thesis.

Appreciation is also expressed to Dr. Samuel C. Collins for his many helpful comments and suggestions.

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PART I

Purpose of the Investigation

The purpose of our investigation was to obtain quantitative data concerning the operation of a large-scale centrifugal* refrigerator and also to obtain data that might aid in the analytical solution of this problem.

Reasons for the Selection of the Subject

We became interested in this subject after talking to students who had previously done work on this problem. Our interests were further motivated by reading two theses and several magazine articles about this topic.

Previous Contributions to the Subject

M. G. Ranque,[1] a French scientist, holds a patent (U. S. Patent Office, Washington, D. C., 1934) on a device using the centrifugal principle for refrigeration although R. Hilsch[2] of Erlangen, Germany, was the first to publish any information concerning the apparatus. An attempt has been made to locate Mr. Ranque; but, at the time of this writing, he had not been located. The idea was introduced into this country by R. N. Milton[3] of John Hopkins University, who discovered a workable

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(1) Ranque, M. G., Journal de Physique, 71, 4, 112; 1933.
(2) Hilsch, R., "The Use of the Expansion of Gases in a Centrifugal Field as a Cooling Process," Thesis, Erlangen, Germany.
(3) Industrial and Engineering Chemistry, 38, 5-5; 1946.

*The terms "centrifugal" and "vortex" are used interchangeably when referring to the apparatus tested.

model in Germany while making post-war investigations of German scientific work.

With Milton's paper came newly aroused interest, and several theses were performed in rapid succession[(1)]. These theses have, in general, followed Hilsch's form and have verified his observations and data. Some important conclusions drawn by these men are as follows:

1. The best ratio of diameter of inlet air nozzle to diameter of pipe is 1:4 and of diameter of orifice to diameter of pipe is 1:3.

2. Change in length of the cold pipe has no effect upon the temperature drop except to increase heat transfer losses with increased length of pipe. The throttling valve on the hot exit pipe should be a minimum of 50 diameters from the inlet air nozzle for most efficient operation.

3. A greater temperature drop will be obtained as the cold orifice is moved toward the center of the inlet air nozzle.

4. The center of the vortex is displaced from the center of the chamber.

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(1) See Bibliography for theses performed at M.I.T.

Inadequacy of Previous Data

Previous experimental work in the United States was not begun until 1946, when R. M. Milton discovered the apparatus in Germany. Since Milton brought the apparatus to the United States, several scholastic theses have been conducted on vortex refrigeration,and research has been initiated by various American companies. Due to the fact that the data from these sources have been rather general, this thesis was conducted to give information which might aid in the theoretical solution of the phenomena.

PART II

Summary of Results

1. The maximum temperature drop was observed when approximately 20 per cent of the total flow was flowing out the cold side of the apparatus.

2. With an increase in inlet pressure, the temperature difference of the cold air (Δt_c) continually rises, but at a decreasing rate.

3. The maximum temperature drop was observed using a .793 inch cold orifice opening, giving a ratio of chamber to cold orifice diameter of 2.5 to 1.

4. As the ratio of the cold air flow to the total air flow decreases, a change in the cold orifice diameter has less effect upon the temperature drop of the cold air.

5. The temperature rise of the hot exit air (Δt_H) appears to increase at a constant rate with a decrease in the percentage of air flowing through the hot pipe (W_H).

6. The pressure gradient inside the chamber increases with an increase in pressure.

7. A plot of the dimensionless ratio, temperature drop to inlet temperature, versus the dimensionless ratio, inlet pressure to hot exit pressure, produced a curve which can be used to predict the temperature drop for a 2 inch diameter chamber at a

given inlet air temperature and pressure and hot exit pressure.

8. The center of the vortex, at the point of cold exit, was displaced from the center of the model.

Overall Description of the System

The layout of the apparatus, as shown on page 36, and the photographs on the following pages will aid the reader in picturing the equipment used in this experiment.

An Ingersoll-Rand two-stage compressor, rated at 120 C.F.M. of free air, was used to supply the compressed air. The inlet air was supplied by a 1/2 inch standard pipe equipped with a thermometer, pressure gage, and valve. A bleeder valve was installed on the inlet air pipe to maintain constant pressure.

The apparatus itself was made of Plexiglas. The compressed air entered the chamber tangentially through a converging nozzle. (See following section entitled, "Detailed Description of Model").

A 11-1/2 inch length of Micarta tubing of 2.5 inch O.D., 2 inch I.D., served as the hot exit pipe. An automobile muffler of approximately 8 inch diameter was used as a stagnation tank. A thermometer well and pressure tap were installed in a standard 2 inch pipe which carried the hot air from the hot stagnation tank. Following 71 inches of this pipe was a 2 inch standard flange equipped with a flat-plate, sharp-edged orifice (1 inch diameter) and pressure taps for flow measurement purposes. The length of pipe and the orifice were designed according to A.S.M.E. specification for a Radius Orifice(1). Twenty-two inches of this same pipe followed with a valve at the end. After the valve, the hot air was exhausted to the atmosphere by means of an elbow and a length of pipe which reached the vicinity of the ceiling.

Fourteen inches of Micarta tubing, 1.25 inch O.D., .75 inch I.D., took the cold air from the cold orifice. The various size cold orifice plates were inserted between the end of this Micarta tubing and a space on the apparatus designed for holding plates in position. A goose-neck joint made of elbows and 1 inch standard nipples was attached to the end of the Micarta tubing to permit removal of the cold orifice plates and to reduce any strain on the various pipe fittings leading into the apparatus. This 1 inch pipe carried the air to a 10 inch standard cast iron pipe, 24 inches in length. This served as a cold stagnation tank. Thirty inches of 2 inch standard pipe carried the cold air to the cold flow orifice which was constructed in the same manner as the hot flow orifice. Following the cold flow orifice, the cold air was exhausted to the atmosphere.

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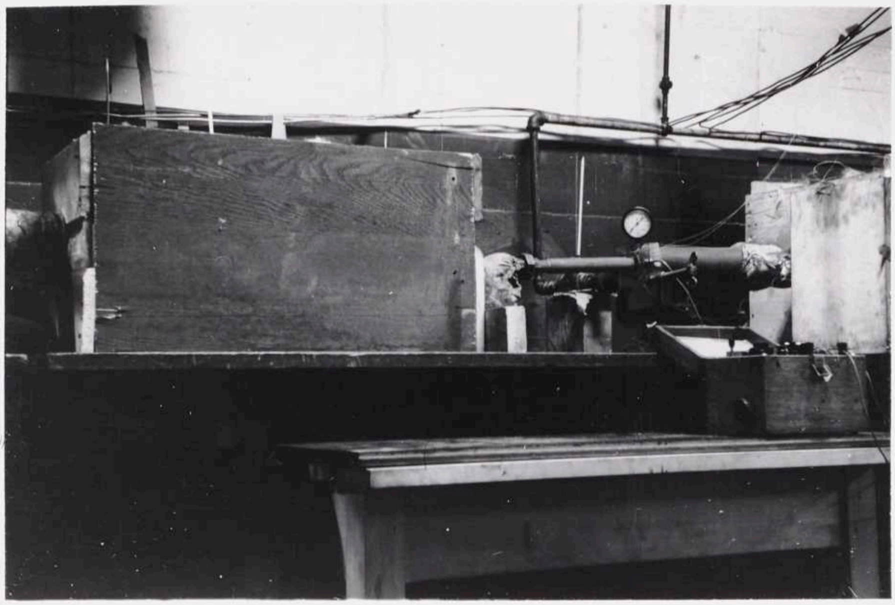
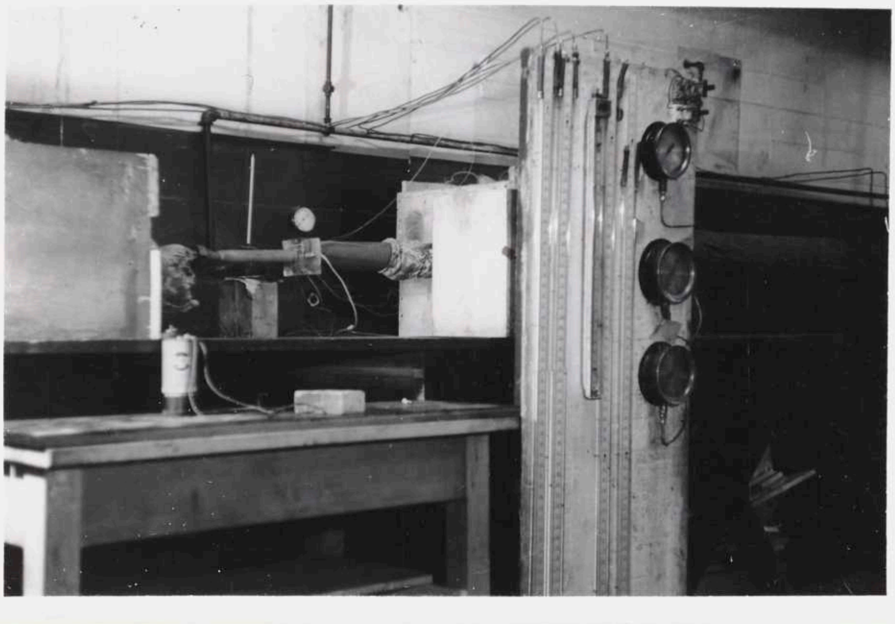
(1) <u>Quantitative Measurement</u>, A.S.M.E. Power Test Codes, 1940.



PHOTOGRAPHS OF APPARATUS



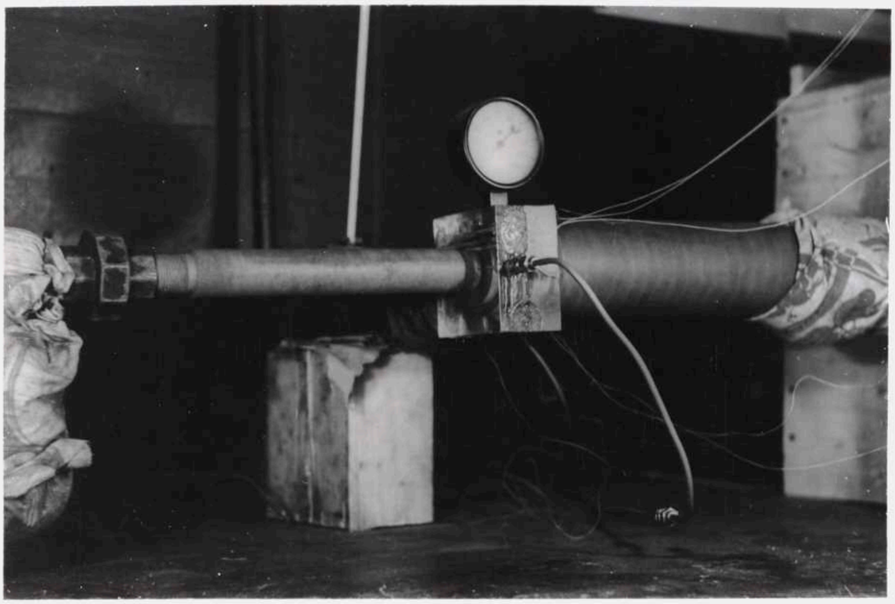
Detailed Description of Model

A working drawing of the model is shown on page 37.

The model was machined from a 2 inch sheet of commercial plastic, tradenamed "Plexiglas." The chamber diameter was 2 inches, and the nozzle diameter .207 inch. The nozzle was machined with a special tool, giving a conical shape with the sides of the conic being curves of 1 inch radius.

Three pressure taps were placed in the model to obtain a pressure gradient across the chamber. These pressure taps were located at .520, .755, and 1.005 inches radii. Three thermocouples were placed in the side of the model; but, due to difficulties later encountered, it was impossible to obtain data from them.

The size of the cold outlet was controlled by placing 3/64 inch stainless steel orifice plates over the outlet. The plates were held in position by the cold outlet pipe. Orifice sizes were .625, .50, .375, .25 inches and the cold outlet in the plastic formed a .793 inch orifice.



CLOSE-UP VIEW OF MODEL

Method of Testing

The first seven runs were made at constant pressure (160 p.s.i.g.), using five different cold orifices (.793, .625, .50, .375, and .25 inch diameter). For each orifice size, six settings of the hot exit valve were made. Next, a cold orifice plate of .5 inch diameter was used to make five more runs at 160, 140, 120, 100, and 80 p.s.i.g., varying the hot exit valve setting six times for each pressure.

In each case the runs were conducted as follows: Insert a specified cold orifice size, establish the desired pressure, and then vary the value of $\frac{W_c}{W_t}$ from a very low value (say, 5%) to a value of about 40 to 50%.

Discussion of Results

The results of the experiment indicate the general characteristics of the particular unit tested.

Figure 1 shows the effect of changing inlet pressure while holding the cold orifice diameter constant. This graph clearly indicates that greater temperature differences can be obtained by using higher inlet pressures, but the relative increase in t_c, for each increment of higher pressure, decreases. (See Figure 2, which is derived from Figure 1). Figure 1 also indicates that the greatest temperature differences are achieved with only 18 to 25% cold flow ($\frac{W_c}{W_t} \times 100$). From a practical standpoint, this is rather discouraging because any utilization of the cooling effect of the Vortex Refrigerator would require at least 75% of the flow to be wasted through the hot end of the unit. It can also be noted from this curve that the point of maximum temperature difference is located at increasing values of percentage cold flow as the inlet pressure increases. This might indicate that at very high pressures, say 1000 p.s.i., a much greater cold flow per cent could be obtained at the optimum cooling point.

Figure 3 shows the effect of changing the orifice diameter while holding the pressure constant. The results of Figure 3 are more clearly shown in Figure 4. Figure 4 rearranges the variables plotted in Figure 3 to show the effect of changing orifice diameter for various values of $\frac{W_c}{W_t}$, holding the inlet pressure constant. From Figure 4 it can be seen that the effect of cold orifice diameter on cooling effect is related to the per cent of cold air $(\frac{W_c}{W_t})$. At higher percentages of $\frac{W_c}{W_t} \times 100$, the cold orifice diameter has a very definite effect on Δt_c while at lower values of $\frac{W_c}{W_t}$, the curves tend to approach a straight line. These curves show the point of maximum cooling effect as being a .793 inch cold orifice, giving a ratio of chamber diameter to cold orifice diameter of 2.5 to 1. The inconsistency of the .250 inch orifice curve in Figure 3 and the $\frac{W_c}{W_t} = .20$ curve in Figure 4 is due to the inaccuracy of measurement in these ranges caused by ice formation in the chamber.

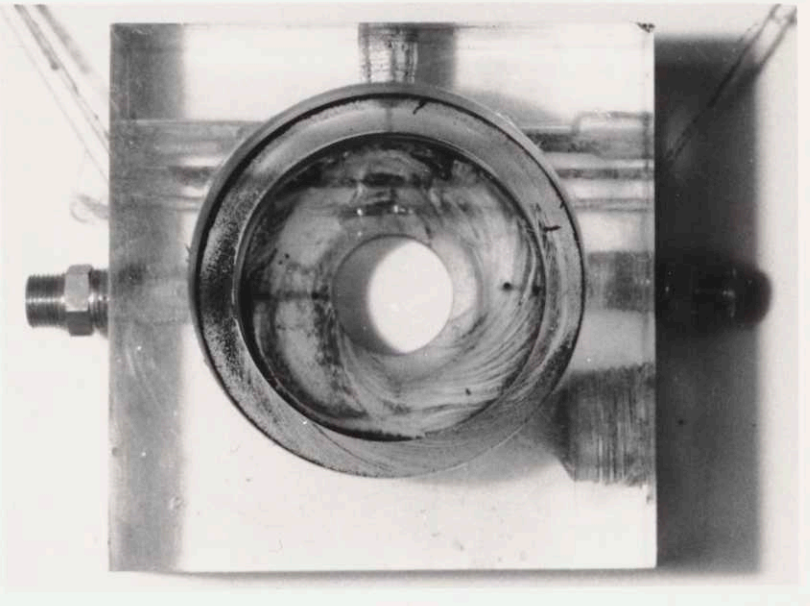
Figure 5 shows the effect of similar variables for the hot exit side that Figure 1 shows for the cold exit side. Although points were plotted for pressures of 140 and 160 p.s.i.g., it was not possible to draw consistent curves through these points. It is felt that, at these higher pressures, an unexplainable phenomenon occurs to cause these inconsistencies. The graph shows a linear distribution of Δt_H with per cent hot flow $(\frac{W_H}{W_t})$ for the range of data taken. It is highly probable that the linear distribution changes to curve back toward the zero t_H point on each end of this plot, due to the fact that there would be no t_H for either zero $\frac{W_H}{W_t}$ or $\frac{W_H}{W_t}$ equal to 1. The cooling noted at higher values of $\frac{W_H}{W_t}$ is due to the Joule-Thomson effect caused by the hot stagnation tank.

Figures No. 6 and 7 show that an increase in the inlet pressure causes an increase in the pressure gradient inside the chamber. Since the pressure at the center of the chamber remains essentially constant at a pressure just above atmospheric, an increase in the inlet should result in an increased pressure gradient. This then is in accordance with the theory.

Very little difference in pressure gradient can be noted for various values of the ratio $\frac{W_c}{W_t}$ (ratio of cold air flow to total air flow), and there seems to be no obvious reason why the value of $\frac{W_c}{W_t}$ should have any effect on the pressure gradient. Of course, with an increase in the value of $\frac{W_c}{W_t}$ comes an increase in the overall pressure in the chamber, as is evidenced by the higher positions of the members of the family of curves.

Figure 8 shows the result of plotting the dimensionless ratio, temperature drop to inlet temperature, versus the dimensionless ratio, inlet pressure to hot exit pressure. Since this plot includes points from all runs, it will be useful in predicting the temperature drop that may be expected from a 2 inch diameter chamber. With a given inlet air pressure and temperature, the temperature in the cold pipe can be read from this plot for any value of hot exit pressure.

The path of the air prior to leaving the cold orifice, as defined by the sediment, is shown in the photograph on the following page. This photograph indicates that the center of the vortex was displaced due to the effect of the inlet air. It has been suggested that this could be remedied by equally placing nozzles around the periphery of the chamber. This picture also indicates that there are very few rotations by a unit mass of air about the circumference of the chamber before the cold air leaves the chamber.



VIEW OF MODEL FROM HOT EXIT END

SHOWING SEDIMENT PATTERN

Accuracy of Results

The chief difficulty encountered was due to the formation of ice at the cold exit of the chamber. When this happened, the system seemed to go into a transition state as noted by the rapid and continuous changes in the pressure and temperature readings. Thus there is a good possibility of percentage errors up to 30 per cent for readings taken when very little cold air was being removed from the apparatus and the pressure high (conditions conducive to very low temperatures at the cold exit).

The stagnation tanks were installed in the system to reduce the rotational velocity of the air leaving the chamber and so give true temperature readings. These temperature measurements were used to compute an overall heat balance which for several runs showed an average error of about 6 per cent. This error could be contributed to the fact that it was impossible to reduce heat losses to a negligible amount because of the problem of insulation presented by such large tanks.

A check was made on the flow rate measurements by comparing the theoretical flow to the measured flow. Figure 9 shows the results of that comparison. As should be expected, the theoretical flow rate was always greater than the actual and a greater difference occurred at higher pressures (because of leakage) than at low.

When very little air was being passed through the hot pipe, the resulting temperatures were comparatively high (around 120°F.) and steady state conditions difficult to obtain. It is estimated that a period of two hours would be required to obtain such a steady state. Due to lack of time, this was impossible, and consequently some error was introduced.

Suggestions for Future Investigation

1. A series of seven or eight pressure readings could be taken to determine very accurately the pressure variation inside the chamber.

2. Maintain a vacuum at the cold exit so that greater temperature drops could be attained without an increase in the inlet pressure. This would also probably eliminate a good deal of the mixing which occurs when the cold air is being forced out of the chamber.

3. Maintain the inlet air at such a temperature that no ice formation would take place at any point in the apparatus. An alternate method of eliminating "icing" difficulties would be to use air completely free of moisture.

4. Make a photographic study of the flow patterns inside the chamber and also in both the hot and cold exit pipes.

5. Place a series of seven or eight thermocouples inside the chamber at various distances from the center of the chamber so that a temperature gradient of some sort might be established.

6. Place valves on both the hot and cold exit pipes and note the effect on the various variables.

7. Construct an oval-shaped chamber so that the air will have more difficulty in moving longitudinally and, therefore,

will allow more time for heat transfer from the center of the vortex to the periphery to take place.

8. Build a model and accompanying system through which water or some other liquid could be passed and take data similar to that taken here. Such data might prove whether or not all or most of the cooling effect is due to the expansion of the air in the chamber.

PART III

APPENDICES

APPENDICES
(continued)

APPENDIX A

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BIBLIOGRAPHY

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3. A.S.M.E. Power Test Codes, 1940, Information on Instruments and Apparatus, Part 5, Quantitative Measurement.

4. Hunter, John W., and Bruce H. Mayer, "Centrifugal Refrigeration," Thesis, M.I.T., January, 1947.

5. Industrial and Engineering Chemistry, 38, 12-5-16, 1946.

6. Haddox, Roy, Jr., Hunter, John W., and Plunkett, William H., Jr., "Experimental Investigation of Centrifugal Refrigeration," Thesis, M.I.T., June, 1947.

APPENDIX B

SYMBOLS

SAMPLE CALCULATIONS

<u>SYMBOLS</u>

A - Area of orifice (sq. in.)

D_o - Diameter of orifice (in.)

D_p - Diameter of pipe (in.)

h_i - Enthalphy of air at entrance to inlet nozzle (BTU/lb.)

h_2 - Enthalphy of air at exit of inlet nozzle, assuming isentropic expansion (BTU/lb.)

h_c - Enthalphy of air in cold stagnation tank (BTU/lb.)

h_h - Enthalphy of air in hot stagnation tank (BTU/lb.)

K - Discharge coefficient of orifice

P_1' - Pressure of air at entrance of inlet nozzle (p.s.i.g.)

P_2' - Pressure of air at exit of inlet nozzle (p.s.i.g.)

P_1 - Pressure in chamber at a point .520 inch from center of chamber (p.s.i.g.)

P_2 - Pressure in chamber at a point .755 in. from center of chamber (p.s.i.g.)

P_3 - Pressure in chamber at a point 1.005 inches from center of chamber

P_c - Pressure in cold pipe measured just past the cold stagnation chamber (p.s.i.a.)

P_H - Pressure drop across flat plate orifice in hot pipe (p.s.i.a.)

P_c - Pressure drop across flat plate orifice in cold pipe (p.s.i.a.)

SYMBOLS
(continued)

R_D - Reynolds number

T_c - Temperature of air in cold stagnation tank (°F.)

T_h - Temperature of air in hot stagnation tank (°F.)

T_{in} - Temperature of air entering the chamber measured at a distance of 8 inches from the inlet nozzle

T_2 - Temperature of air in chamber measured by means of a thermocouple embedded in the wall of the chamber at a distance of .725 inch from its center (°F.)

T_3 - Temperature of air in chamber measured as above at a distance of 1.005 inches from the center of the chamber.

v_1 - Specific volume of air at entrance of inlet nozzle (ft.^{3}/lb.)

v_2 - Specific volume of air at exit of inlet nozzle (ft.^{3}/lb.)

V_2 - Velocity of air at exit from inlet nozzle assuming isentropic expansion (ft./sec.)

W - Theoretical flow rate of incoming air assuming isentropic expansion (lb./sec.)

W_c - Flow rate through cold pipe (lb./sec.)

W_H - Flow rate through hot pipe (lb./sec.)

W_t - ($W_H + W_c$) Total flow rate of gas leaving apparatus (lb./sec.)

Y - Expansion factor

SYMBOLS
(continued)

ρ_c - Density of air at P_c and T_c (lb./ft.^{3})

ρ_H - Density of air at P_H and T_H (lb./ft.^{3})

ϕ_1 - Entropy of air before inlet nozzle (BTU/lb.)

ϕ_2 - Entropy of air at exit of inlet nozzle (BTU/lb.)

SAMPLE CALCULATIONS

Rate of Flow (Calculation made for each run)

The method used for calculating the rates of flow is in accordance with that set forth in the A.S.M.E. Power Test Codes, 1940, Information on Instruments and Apparatus, Part 5, Quantitative Measurement. A sharp-edged flat-plate orifice 1 inch in diameter and radius taps were used. The pipe used for both the hot and cold flow measurements was standard 2-inch pipe.

Run No. 1

$W_c = 0.668 \ \underline{Y} \ A \ \underline{K} \sqrt{\rho \Delta P}$

$\underline{Y}$ from plot of $\frac{D_o}{D_p}$ versus $\underline{Y}$ = .999

$$A = \frac{\pi D_o^2}{4}$$

$\underline{K}$ remained constant at .620, so that there was no necessity for estimating a Reynolds Number and then resolving for a new Reynolds Number with the flow rate thus attained.

$$\bar{\underline{K}} = 2.702 \frac{P_c}{T_c} = .0821$$

$W_c = 0.668 \times .999 \times .785 \times .620 \times .0571$

$W_c = .0185$ lb./sec.

Average Actual Rate of Flow ($W_t = W_H + W_c$) for One Inlet Pressure

Runs No. 29, 30, 31, 32, 33, 34

Run No.	W_H	W_c	W_t
29	.0836	.0144	.0980
30	.0800	.0181	.0981
31	.0717	.0250	.0967
32	.0622	.0314	.0936
33	.0590	.0371	.0961
34	.0538	.0436	.0974
			.5799

$W_t = \frac{.5799}{6}$ = .0967 lb./sec. (Arithmetic Average for $P_1' = 120$ p.s.i.g.)

Average Theoretical Rate of Flow (W) for One Inlet Pressure

Runs No. 29, 30, 31, 32, 33, 34

$T_1 = 100.5°F. = 560.5°R$

$P_1' = 120$ p.s.i.g. $= 134.7$ p.s.i.a.

$h_1 = 38.50$ BTU/lb.

$\phi_1 = .08091$ BTU/lb.

$T_2 = 467°R$

$P_2' = .528 \times 134.7 = 71$ p.s.i.a.

$h_2 = 16.06$ BTU/lb.

$v_2 = \dfrac{53.3 \times 467}{71 \times 144} = 2.43$ ft.^{3}/lb.

$$\phi_2 = \phi_1 - R\ln \frac{P}{P_2}$$

$$= .08091 - .04381$$

$$= .03710 \text{ BTU/lb.}$$

$V_2 = 223.8 \quad 38.50-16.06 \quad = 1060$ ft./sec.

Velocity at entrance to inlet nozzle estimated to be 100 ft./sec.

$V_2 = V_2 + 100 = 1160$ ft./sec.

$$W = \frac{A_2V_2}{v_2} = \frac{3.14 \times .0166 \times 1160}{4 \times 2.43}$$

$W = .103$ lb./sec.

Heat Balance

Run No. 13

$$h_1 W_t = h_c W_c + h_h W_H$$

$$.1268 \times 37.9 = 23.73 \times .0518 + 41.51 \times .0750$$

$$4.80 = 1.23 + 3.12$$

$$4.80 = 4.35$$

$$\% \text{ Error} = \frac{4.80-4.35}{4.80} = 9.39\%$$

APPENDIX C

PLOTS

LEGEND:
P₁' = INLET PRESSURE (PSIG)
COLD ORIFICE DIAM. KEPT CONSTANT AT 0.50 IN.

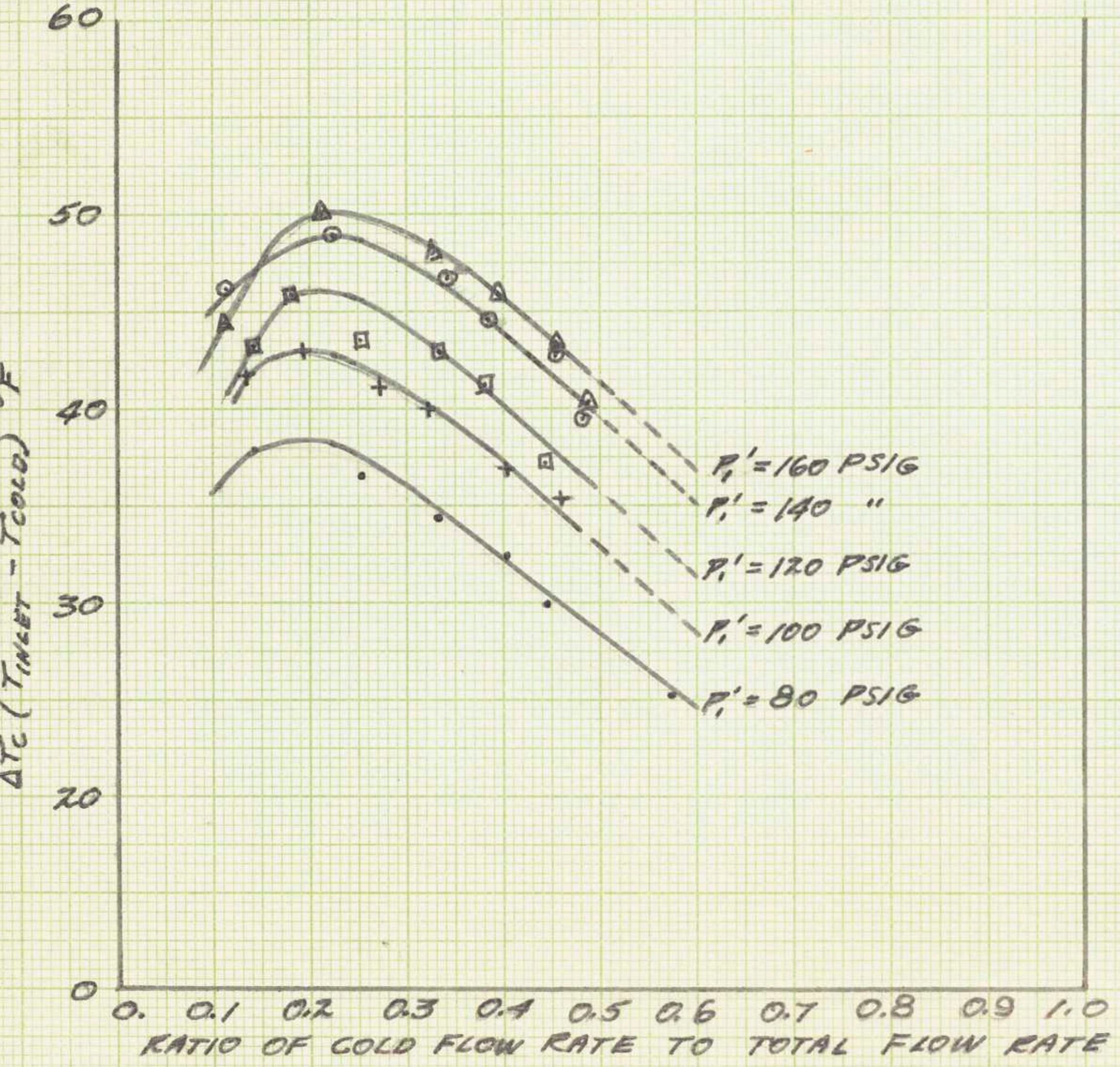

FIGURE NO. 1

EFFECT OF VARYING INLET PRESS. AND THROTTLE SETTING AT HOT END ON TEMP. DROP OF COLD AIR

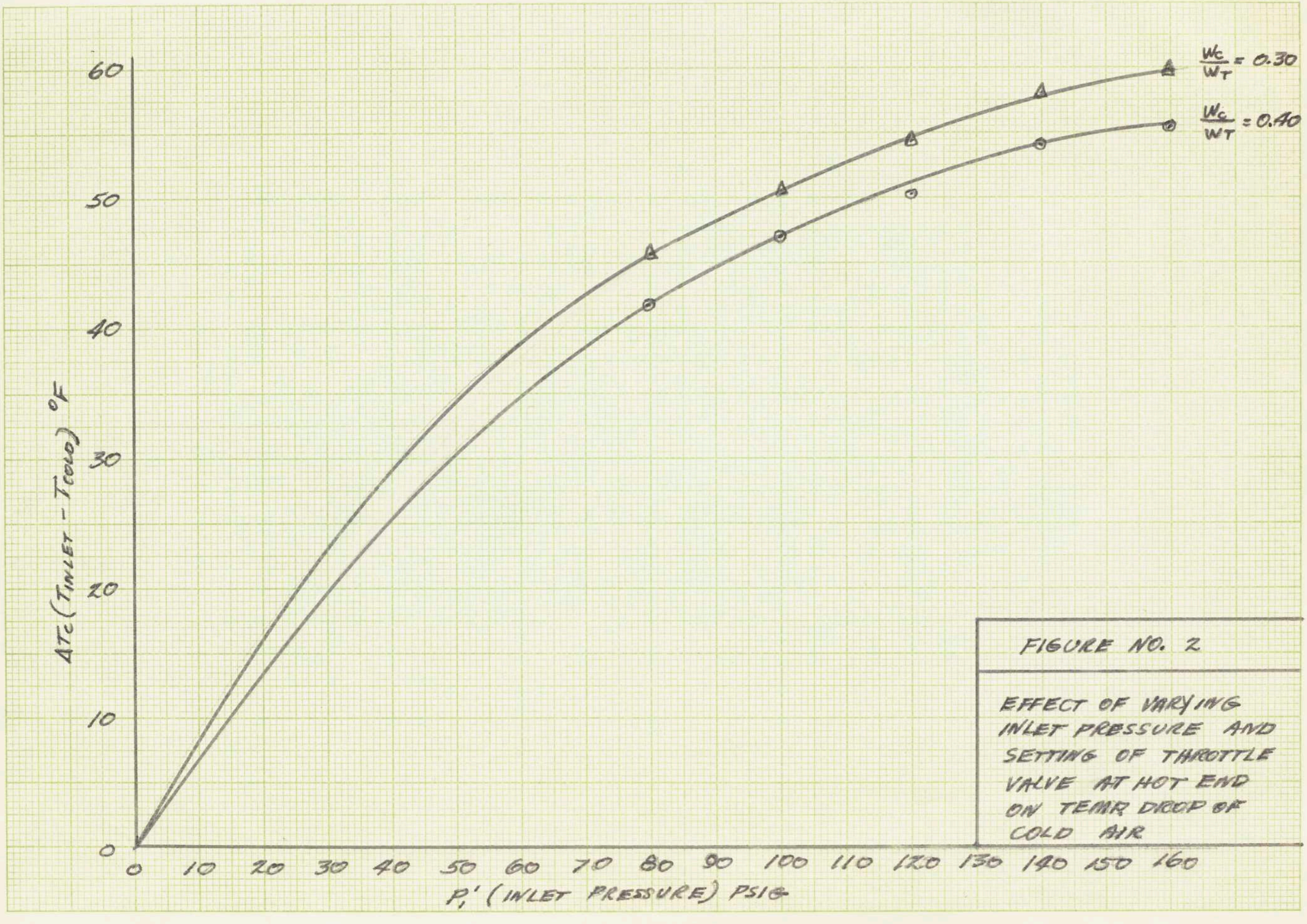

FIGURE NO. 2

EFFECT OF VARYING INLET PRESSURE AND SETTING OF THROTTLE VALVE AT HOT END ON TEMP. DROP OF COLD AIR

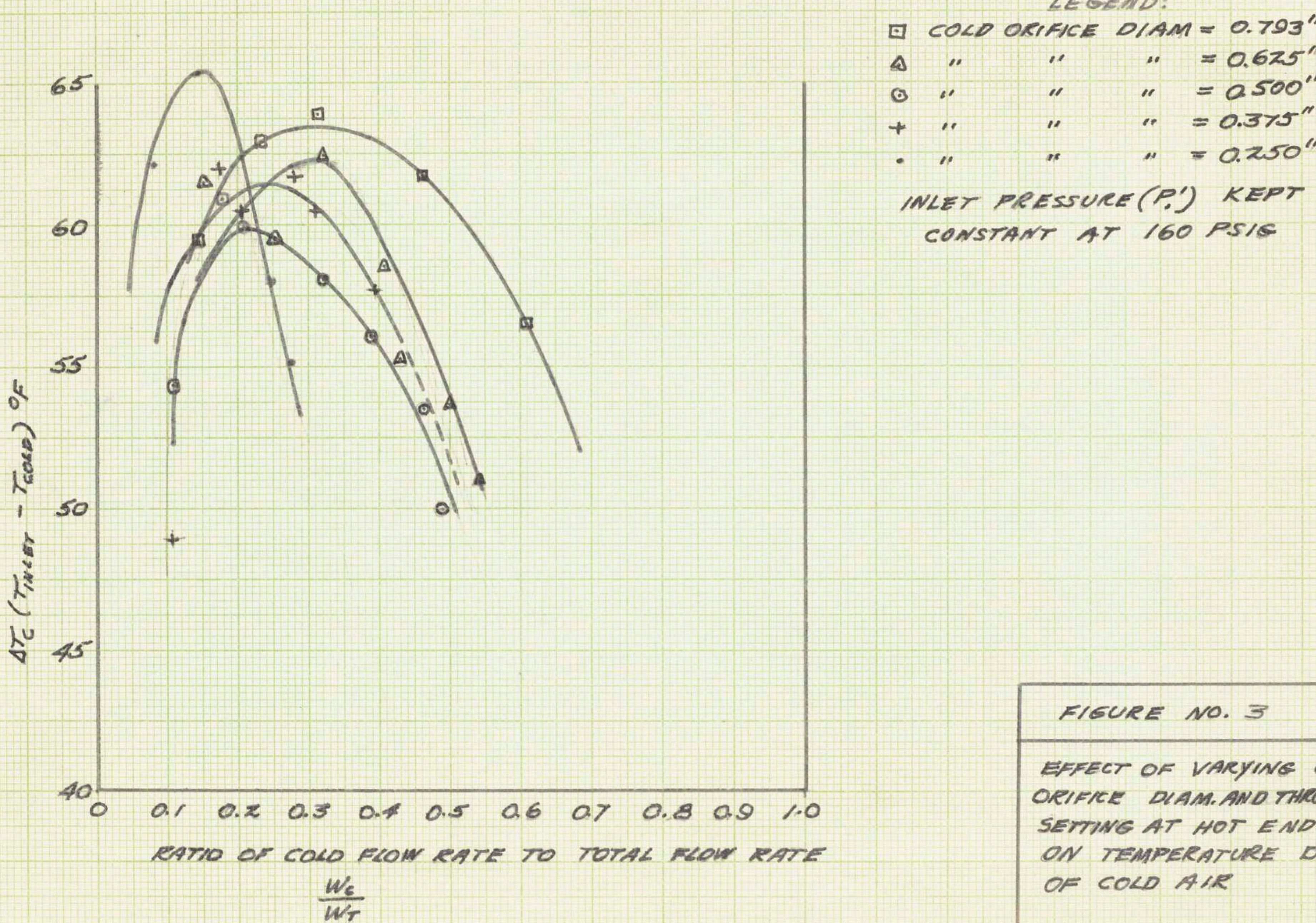

FIGURE NO. 3

EFFECT OF VARYING COLD ORIFICE DIAM. AND THROTTLE SETTING AT HOT END ON TEMPERATURE DROP OF COLD AIR

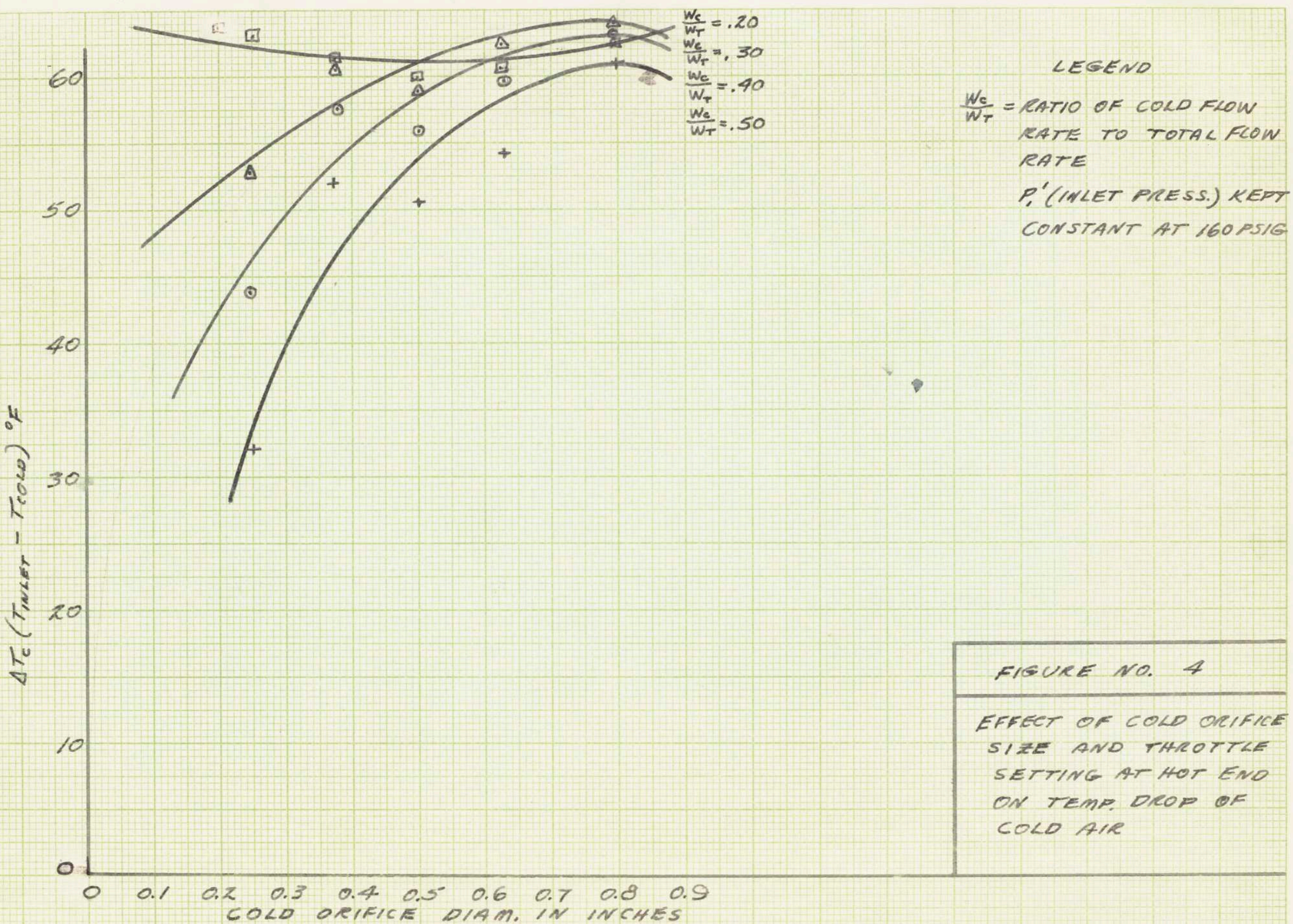

FIGURE NO. 4

EFFECT OF COLD ORIFICE SIZE AND THROTTLE SETTING AT HOT END ON TEMP. DROP OF COLD AIR

LEGEND:

- ⊡ $P_1' = 80$ PSIG
- ⊙ $P_1' = 100$ "
- • $P_1' = 120$ "
- + $P_1' = 140$ "
- △ $P_1' = 160$ "

COLD ORIFICE DIAM. KEPT CONSTANT AT 0.5 INCHES

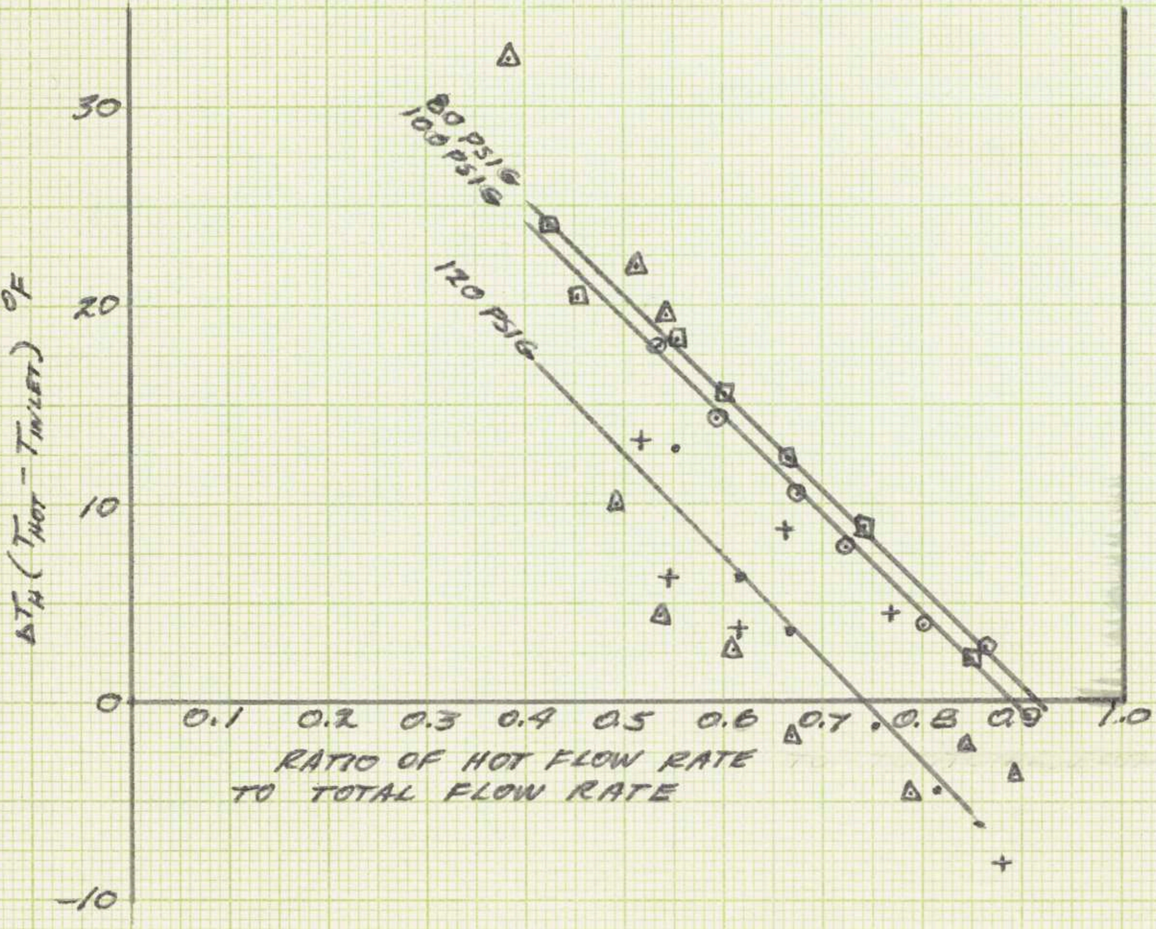

FIGURE NO. 5

EFFECT OF VARYING INLET PRESSURE AND THROTTLE SETTING AT HOT END ON TEMPERATURE RISE OF HOT GASES

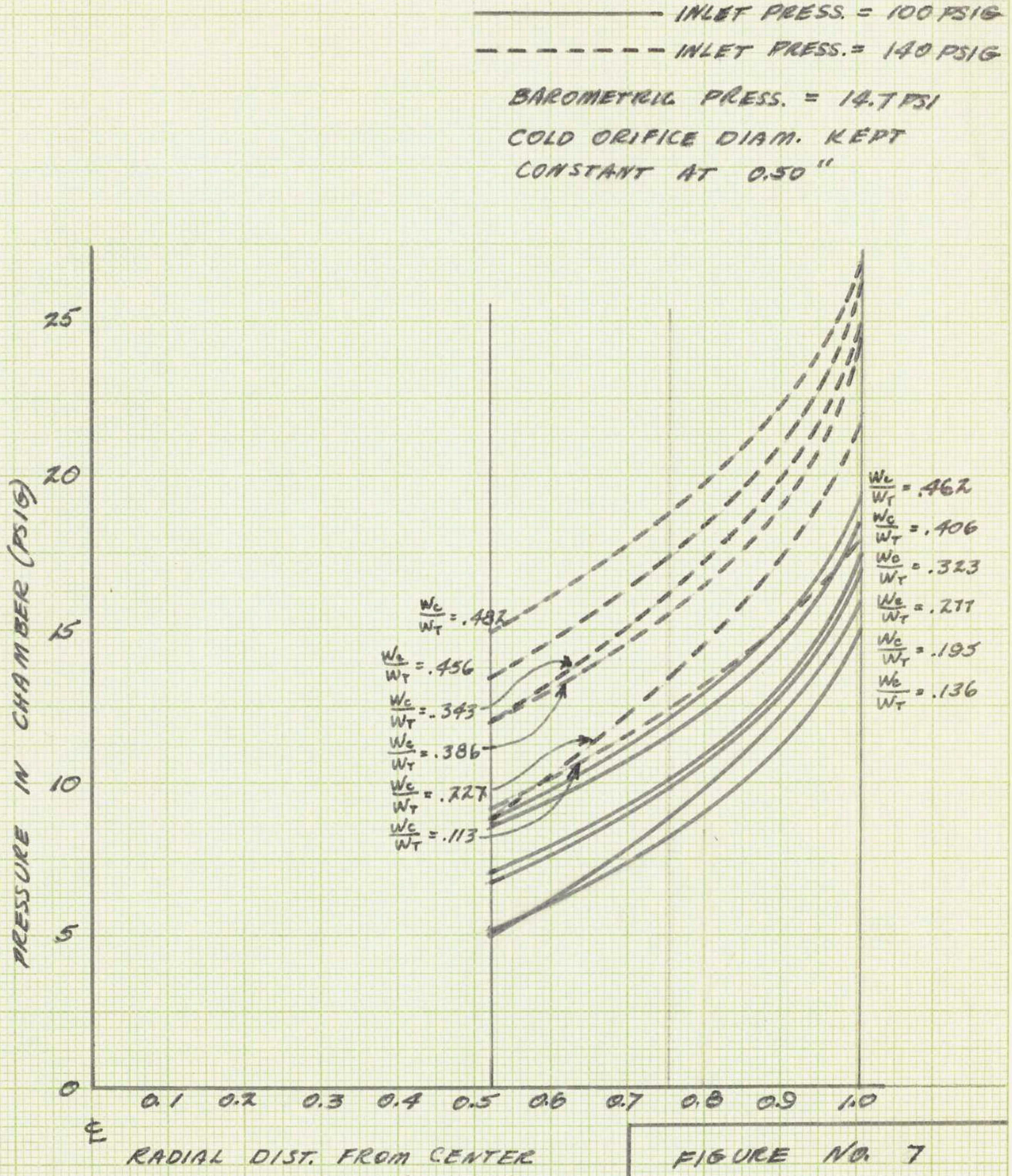

FIGURE NO. 7

PRESSURE VARIATION IN CHAMBER OF APPARATUS

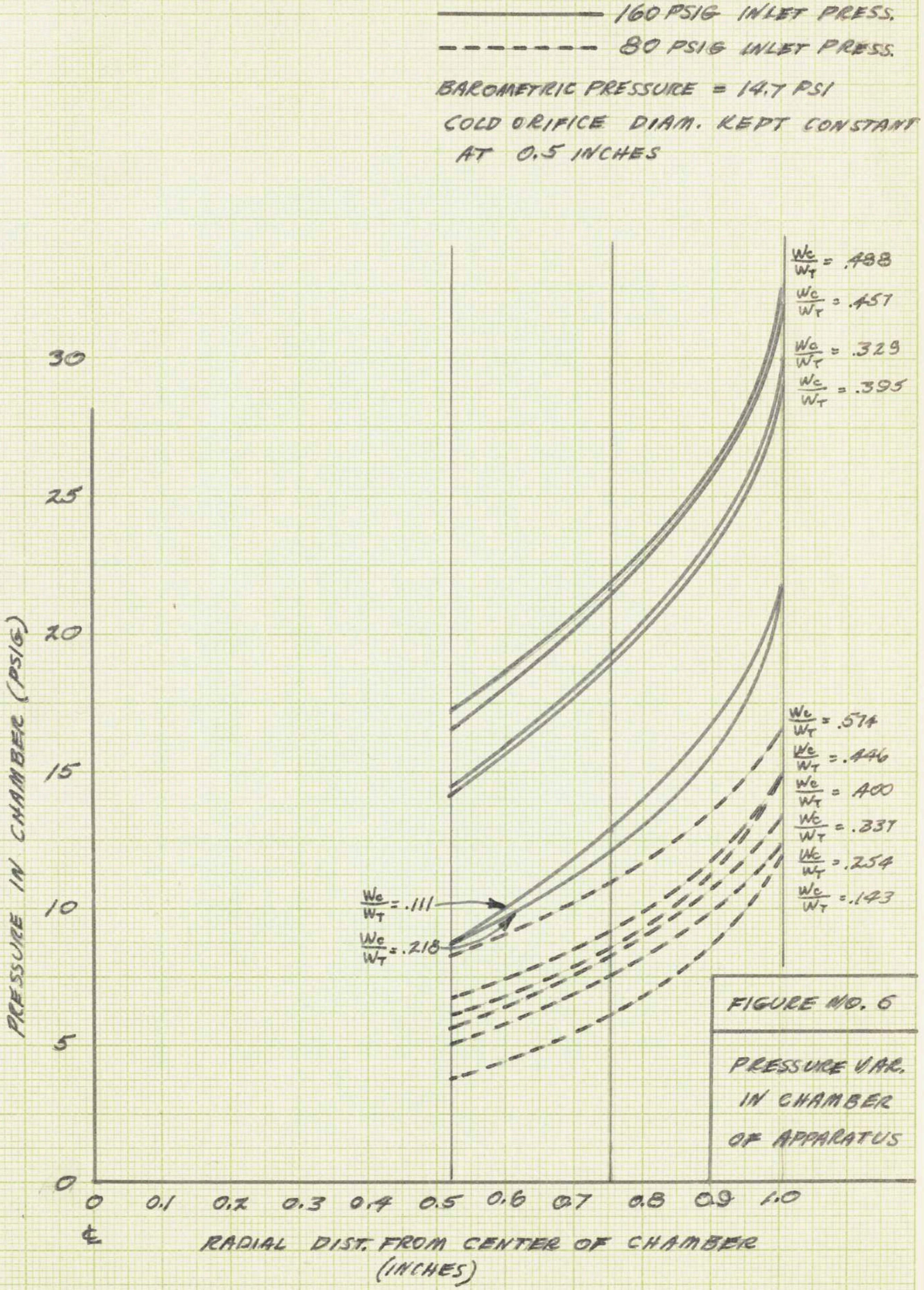

FIGURE NO. 6

PRESSURE VAR. IN CHAMBER OF APPARATUS

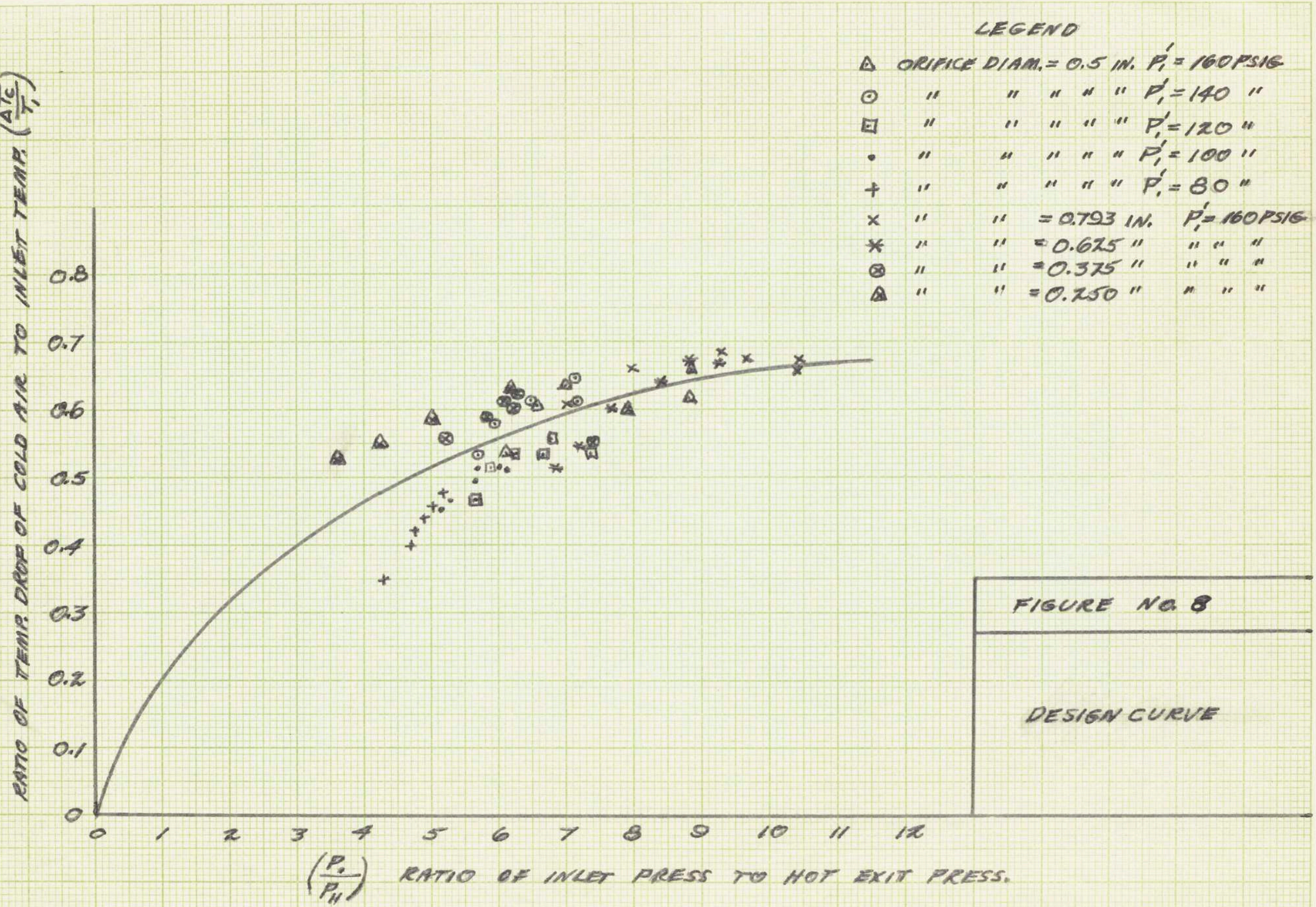

FIGURE NO. 8

DESIGN CURVE

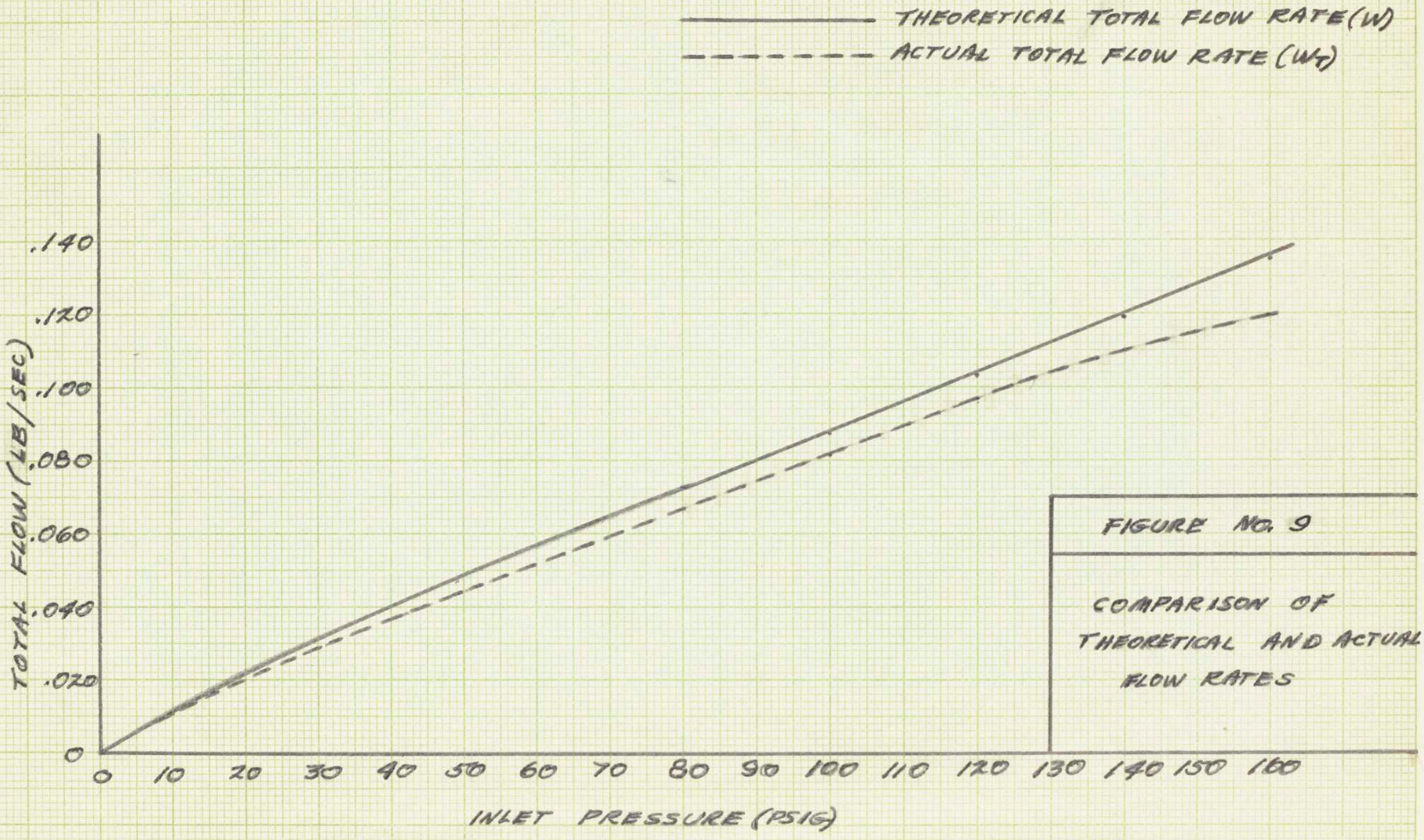

FIGURE No. 9

COMPARISON OF THEORETICAL AND ACTUAL FLOW RATES

APPENDIX D

DIAGRAMS

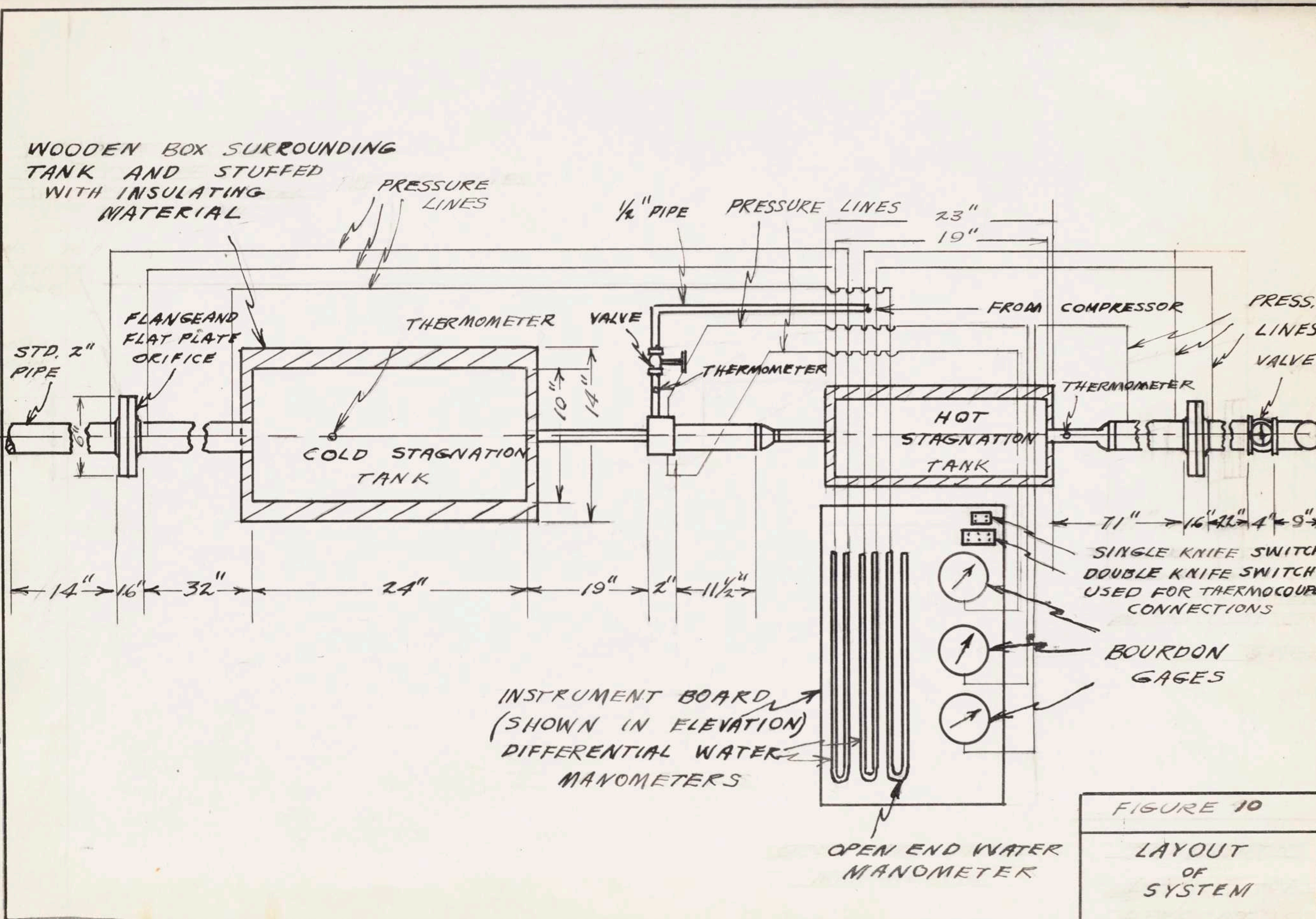

FIGURE 10

LAYOUT OF SYSTEM

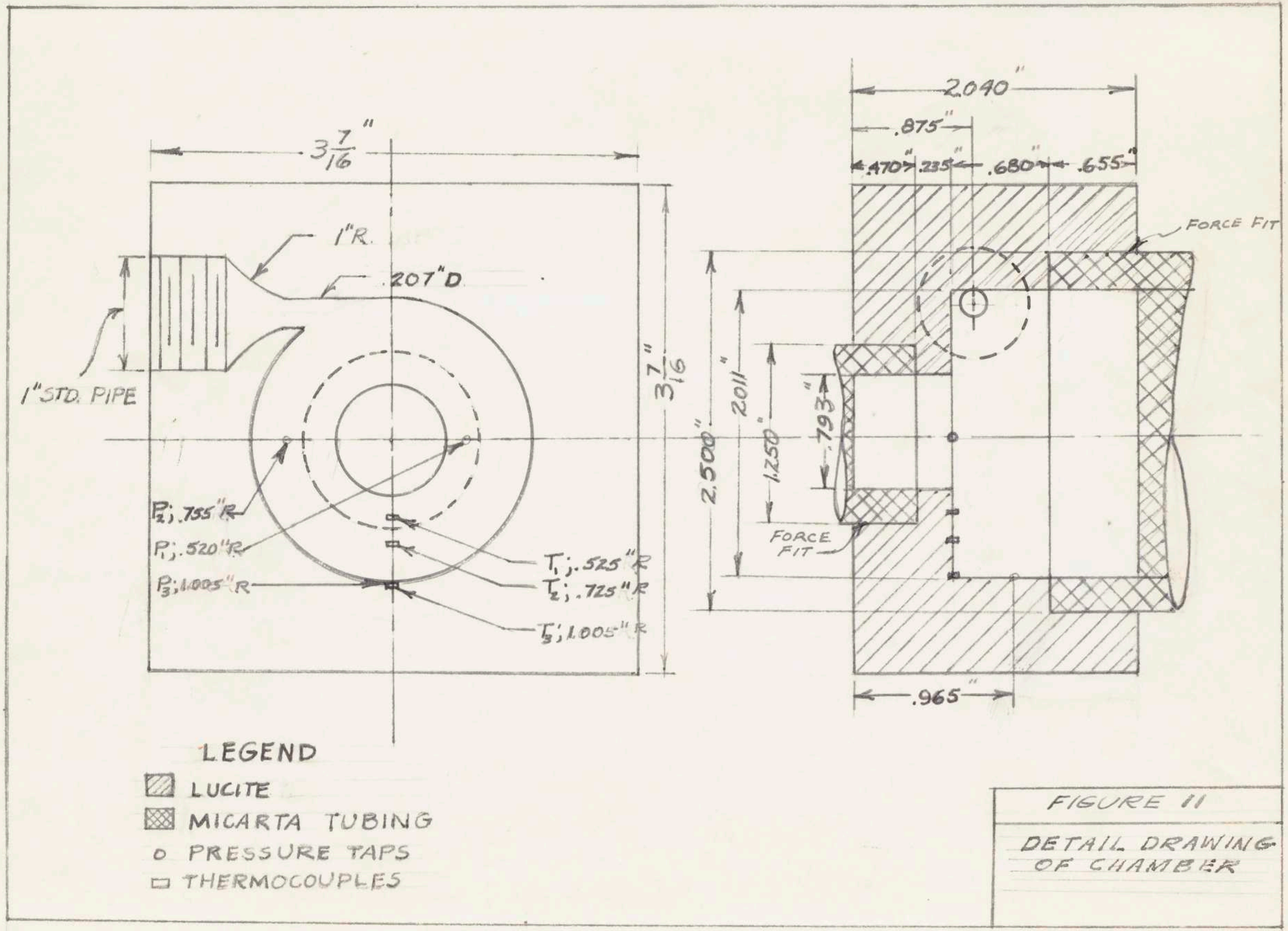

FIGURE 11

DETAIL DRAWING OF CHAMBER

APPENDIX E

DATA AND CALCULATED RESULTS

DATA WORKSHEET FOR MAKING CURVES

Run No.	Ori-fice in.	P_1' psig	T_{in} °F.	T_c °F.	T_h °F.	P_1 psig	P_2 psig	P_3 psig	ΔP_H in H_2O gage	ΔP_c in H_2O gage	P_c in H_2O gage	P_H psig	ΔT_c [x] °F.	ΔT_h [x] °F.	W_c [x] lb./sec.	W_H [x] lb./sec.	W_t [x] lb./sec.	$\frac{W_c}{W_t}$x100 [x] %	$\frac{W_H}{W_t}$x100 [x] %
1	.793	175	91	31.5	77	5.4	8.8	18	37.9	1.1	6.5	2	59.5	-14	.0185	.108	.1265	14.6	85.4
2	.793	159	92	31.2	81.3	6.8	9.6	19	35.8	1.3	.92	2.6	60.8	-10.7	.0199	.107	.1269	15.7	84.3
3	.793	159	92	31	82.5	6	10	18	34.1	1.85	1.3	2	61	- 9.5	.0238	.105	.1288	18.5	31.5
4	.793	159	92	30.5	84	7.2	10	19	30.3	2.5	1.85	2.5	61.5	- 8	.0277	.0982	.1259	22.0	78.0
5	.793	159	93	30	88	7.1	10	19	29.0	3	2.1	3.2	63	- 5	.0304	.0957	.1261	23.8	76.2
6	.793	159	93	29.5	96.5	7.5	11	20	25.9	4	4	4	63.5	3.5	.0358	.0932	.1290	27.8	72.2
7	.793	159	93	29	93	8.6	12.8	22	20.6	5.1	3.8	5	64	0	.0398	.0850	.1248	31.9	68.1
8	.793	159	93.5	31.8	98	10	14.6	23	11.9	10.7	8	7	61.7	+ 4.5	.0599	.0678	.1277	46.8	53.2
9	.793	159	95.5	39	128	12.8	17	27	5.4	19.5	14.6	9.9	56.5	32.5	.0771	.0480	.1251	61.4	38.6
10	.625	159	89	29.5	102.5	9.5	13	23	24.7	3.4	2.4	5	59.5	+13.5	.0323	.0929	.1252	25.8	74.2
11	.625	164	92.5	30.7	40.5	8	11.6	21	38.2	1.2	.8	4	61.8	- 2.0	.0191	.113	.1221	15.6	84.4
12	.625	162	97.3	34.7	112	9.2	14.2	24	19.85	5.6	4.1	6	62.6	+14.7	.0415	.0845	.1260	33	67
13	.625	162	97.5	39	113	10	15.8	25	14.1	8.8	6.6	8	58.5	15.5	.0518	.0750	.1268	41.0	59
14	.625	160	98	42.7	107.5	11.3	15.7	25	11.1	11.0	8.2	9.2	55.3	9.5	.0580	.0755	.1335	43.3	56.7
15	.625	159	99	45.2	109.2	11.9	16.2	26	9.0	12.9	9.6	9.4	53.8	10.2	.0629	.0618	.1247	50.5	49.5
16	.625	159	99.5	48.5	120	13.1	17.2	27.5	7.65	15.5	11.6	10.6	51.0	20.5	.0690	.0576	.1266	54.5	45.5
17	.50	160	88.5	34	85	8.5	12	22	37.2	.6	.5	5	54.5	- 3.5	.0135	.108	.1215	11.1	88.9
18	.50	160	90.5	30.5	86	8.6	12.8	22	27.7	2.2	1.4	5	60	- 4.5	.0259	.0926	.1185	21.8	78.2
19	.50	159	91.5	33.5	90.5	14.5	19.1	30	16.1	5.1	4.78	10.2	58.0	- 1.5	.0390	.0798	.1188	32.9	67.1
20	.50	149	92.5	36.5	95.5	14.2	19.0	29	12.7	7.33	5.6	11.6	56	+ 3.0	.0471	.0727	.1189	39.5	60.5

x = Derived Results

DATA WORKSHEET FOR MAKING CURVES
(continued)

Run No.	Orifice in.	P_1' psig	T_{in} °F.	T_c °F.	T_h °F.	P_1 psig	P_2 psig	P_3 psig	ΔP_H in H_2O gage	ΔP_c in H_2O gage	P_c in H_2O gage	P_H psig	ΔT_c °F. x	ΔT_h °F. x	W_c lb./sec. x	W_H lb./sec. x	W_t lb./sec. x	$\frac{W_c}{W_t}$x100 % x	$\frac{W_c}{W_t}$x100 % x
21	.50	160	92	38.6	111.5	16.5	21.3	32	10.05	10.7	7.9	13.2	53.4	19.5	.0571	.0676	.1247	45.7	54.3
22	.50	160	92.7	42.6	114.5	17.1	21.7	32.5	8.95	11.85	8.85	13.8	50.1	21.8	.0600	.0630	.1230	48.8	51.2
23	.50	140	94.5	40	98.5	12.0	15.5	24	11.35	5.7	4.3	9.4	54.5	4.0	.0413	.0657	.1070	38.6	61.4
24	.50	140	93	37	85	9.1	12.5	18	31.0	.6	.10	7.0	56	- 8.0	.0134	.1055	.1189	11.3	88.7
25	.50	140	92	33	96.5	9.0	13.3	22	21.6	2.05	1.0	6.8	59	4.5	.0250	.0855	.1105	22.6	77.4
26	.50	140	92	35.2	101	12.0	16.1	25	14.1	4.4	3.05	9.0	56.8	9	.0365	.0700	.1065	34.3	65.7
27	.50	140	92.5	39.5	99	13.4	17.4	26.5	8.5	7.9	5.80	11.0	53.0	6.5	.0490	.0585	.1075	45.6	54.5
28	.50	140	93	43.5	106.5	15.0	18.8	27	6.55	10.3	7.7	12.3	49.5	13.5	.0558	.0519	.1077	48.2	51.8
29	.50	120	98	44.8	92	6.7	10	17.5	20.8	.7	.43	4.3	53.2	- 6	.0144	.0836	.0980	14.2	85.8
30	.50	120	100	44	95.5	7.0	10.5	18.0	17.65	1.1	.8	4.9	56	- 4.5	.0181	.0800	.0981	18.4	81.6
31	.50	120	100.5	45	99.3	7.9	11.6	20.0	14.2	2.12	1.58	5.5	53.5	- 7	.0250	.0717	.0967	25.8	74.2
32	.50	120	101	48	104.8	8.9	12.7	21	11.05	3.35	2.45	6.9	53	3.8	.0314	.0622	.0936	33.6	66.4
33	.50	120	101.5	51	108	10.1	13.6	22	9.1	4.7	3.5	8	51.5	6.5	.0371	.0590	.0961	38.6	61.4
34	.50	120	102	54.7	115	10.9	14.1	22	6.95	6.5	4.55	9	47.3	13	.0436	.0538	.0974	44.7	55.3
35	.50	100	102	50	104.8	5.1	8.1	15	15.2	.42	.35	3.5	52	2.8	.0110	.0700	.0810	13.6	86.5
36	.50	100	102	49	106	4.9	8.9	16	13.4	.89	.55	4.2	53	4	.0161	.0669	.0830	19.5	80.5
37	.50	100	101.2	49.8	109	6.6	9.8	17	10.27	1.70	1.10	5.0	51.4	7.8	.0223	.0587	.0810	27.7	72.3
38	.50	100	101	50.8	111.5	7.1	10.2	17.5	8.75	2.39	1.65	5.2	50.2	10.5	.0265	.0554	.0819	32.3	67.7
39	.50	100	100.8	53.8	115	8.5	11.6	18	6.55	3.80	2.57	6.9	47.0	14.2	.0337	.0494	.0831	40.6	59.4
40	.50	100	101	55.5	119	8.9	12.2	19.5	5.13	4.69	3.52	7.2	45.5	18	.0374	.0435	.0809	46.2	53.8

x = Derived Results

DATA WORKSHEET FOR MAKING CURVES
(continued)

Run No.	Orifice in.	P_1' psig	T_{in} °F.	T_c °F.	T_h °F.	P_1 psig	P_2 psig	P_3 psig	ΔP_H in H_2O gage	ΔP_c in H_2O gage	P_c in H_2O gage	P_H psig	ΔT_c x °F.	ΔT_h x °F.	W_c x lb./sec.	W_H x lb./sec.	W_t x lb./sec.	$\frac{W_c}{W_t}$x100 x %	$\frac{W_c}{W_t}$x100 x %
41	.50	80	100	52	102.3	3.7	6.0	12	12.15	.37	.10	3.1	48	2.3	.0104	.0620	.0724	14.3	85.7
42	.50	80	99.5	53	108.5	4.9	7.5	12.5	7.29	.93	.79	3.8	46.5	9	.0165	.0485	.0650	25.4	74.6
43	.50	80	99.5	55	111.5	5.6	8.9	13.5	5.8	1.72	1.30	4.4	44.5	12	.0223	.0439	.0662	33.7	66.3
44	.50	80	99.2	56.5	114.5	6.1	8.6	15	4.62	2.40	1.60	5.0	42.7	15.3	.0263	.0397	.0660	40	60.0
45	.50	80	99.2	59	117.5	6.7	9.1	15	3.98	3.14	2.30	5.3	40.2	18.3	.0301	.0373	.0674	44.6	55.4
46	.50	80	99	63.8	123	8.3	10.8	16.5	2.29	5.37	3.85	7.1	35.2	24	.0394	.0292	.0686	57.4	42.6
47	.375	160	89	50	79	7.4	15.2	24	37.3	.7	.05	8.6	49	-10.0	.0143	.127	.1413	10.1	89.4
48	.375	160	99.5	37.5	88.5	14.0	18.2	29	22.3	2.0	1.4	13.2	62.0	-11.0	.0245	.1065	.1310	18.65	81.35
49	.375	160	78.5	38	90.5	16.3	20.3	30	18.0	3.0	2.45	13	60.5	- 8.0	.0300	.0931	.1231	20.43	79.47
50	.375	160	100	38.3	93	16.4	20.6	31	16.5	4.1	2.4	13.6	61.7	- 7	.0352	.0914	.1267	27.8	72.2
51	.375	160	102	41.7	98	18.0	22.1	32.5	14.4	5.7	4.2	15.0	60.3	- 4.0	.0413	.0884	.1297	31.9	68.1
52	.375	160	103	45.4	101	21.1	24.4	35	10.1	8.6	5.7	18.2	57.6	- 2.0	.0510	.0767	.1277	40.0	60.0
53	.25	160	103.5	41.2	88.3	10.9	14.4	25	36.8	.45	.13	7.5	62.3	-15.2	.0116	.122	.1336	8.68	91.32
54	.25	160	103.5	38	91.5	16.1	19.8	30	23.4	1.20	.70	13.3	65.5	-12.0	.0190	.105	.1245	15.3	84.7
55	.25	160	104	42	96.0	23.2	26.0	37	15.95	2.05	1.64	20.1	62	- 8	.0247	.0995	.1242	19.9	80.1
56	.25	160	104	46	98.5	29.3	32.0	43	12.14	3.27	2.48	25.8	58	- 5.5	.0312	.0935	.1247	25.0	75.0
57	.25	160	104	48.8	99.4	34.0	35.5	47.5	10.1	4.8	3.35	30.0	55.2	- 4.6	.0370	.0945	.1315	28.1	71.9

x = Derived Results