AN EXPERIMENTAL INVESTIGATION OF AIR FLOW RATES AT ELEVATED TEMPERATURE THROUGH FITTINGS THAT ARE TYPICAL OF THE PLUMBING IN BARO-INSTRUMENTATION

A THESIS

Presented to

the Faculty of the Graduate Division

Georgia Institute of Technology

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Aeronautical Engineering

By

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July 1956

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Date Approved by Chairman: July 27, 1956

ACKNOWLEDGEMENTS

The author is indebted to Doctor Arnold L. Ducoffe for his suggestion of the subject and for his valuable aid during the preparation of this thesis. Gratitude is extended to Professor Donnell W. Dutton and Professor William A. Hinton for their critical reading of the topic.

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LIST OF SYMBOLS

A_2	area of fitting, in.^{2}
D_1	inside diameter of pipe, in.
D_2	inside diameter of fitting, in.
g	acceleration of gravity, 32.17 ft per sec per sec
K	coefficient of discharge
P_1	pressure upstream of test element, lbs per sq in
P_2	pressure downstream of test element, lbs per sq in
ΔP	$P_1 - P_2$, lbs per sq in
R	gas constant
μ_1	absolute viscosity upstream of test element, lbs per ft sec
w	mass flow rate, lbs per sec
Rn	Reynolds number, $48w/\pi D_2\mu_1$
r	pressure ratio, P_2/P_1
T_1	temperature upstream of test element, deg R
V	velocity, ft per sec
β	diameter ratio, D_2/D_1
ρ	density, lbs per cu ft
Ω	flow factor, $w\sqrt{T_1}/A_2P_1$, $\sqrt{^\circ R}$/sec
$1/\sqrt{1-\beta^4}$	velocity of approach factor
Y	compressibility factor
E	thermal expansion factor for fitting
$\bigvee$	thermocouple

SUMMARY

This report is the result of an experimental investigation conducted to determine, with particular attention given to the effect of elevated temperature, the flow rate of air through fittings that are representative of those found in the plumbing of airplane and missile pressure instrumentation. The analysis was made in a manner similar to that used on standard ASME flow measuring devices such as nozzles and orifices by placing the fittings between the flanges of a commercial grade of 3.15 in. ID pipe and regulating the upstream pressure, the upstream temperature and the downstream pressure. The permanent pressure loss resulting essentially from free expansion and turning losses was measured by using pipe taps placed upstream and downstream of the test element. The fittings tested were ASME orifices having small orifice to pipe diameter ratios and AN standard tee, elbow, and straight-through fittings. The study was limited to (a) maximum head pressures of 60 psi, (b) maximum head temperatures of 500° F, (c) negligible approach velocities (small fitting to pipe diameter ratios), and (d) Reynolds numbers between 10,000 and 380,000.

It was determined that for fittings of similar geometry under the restrictions stated above the flow factor Ω can be expressed as a pure function of the pressure ratio r and curves to this effect are presented for the fittings tested. From this data it is concluded that the mass flow rate of air varies, at least within the limitations of the

experiment, as the square root of the temperature upstream of the test element as predicted by a theoretical analysis of the flow.

CHAPTER I

INTRODUCTION

The high velocities now attained by missiles and airplanes and the subsequent elevated surface temperatures on these vehicles have indicated the need for determining at high temperatures the flow characteristics of air through fittings used in the barometric instrumentation of such aircraft. Reported errors in instrument accuracy have made it seem probable that high temperature phenomena can cause non-negligible departures from the fluid characteristics that are predicted by normal temperature investigations.

In the analysis of actual flow through sharp-edged orifices and nozzles (1, 2, 3) the procedure has been to assume that the mass flow rate varies inversely as the square root of the fluid temperature upstream of the element as is indicated by a theoretical development of the flow equation. The purpose of this research is to investigate the validity of this assumption at elevated temperatures and to predict the flow properties of air through plumbing fittings that are representative of pressure instrumentation. This is done by presenting information, obtained experimentally, expressing implicitly the mass flow rate as a function of the pressure ratio r at room and at elevated temperature through sharp-edged orifices and AN standard tee, straight-through and elbow fittings. This investigation will be limited to small diameter ratios (negligible velocities of approach) and maximum head temperatures of approximately 500° F.

CHAPTER II

APPARATUS

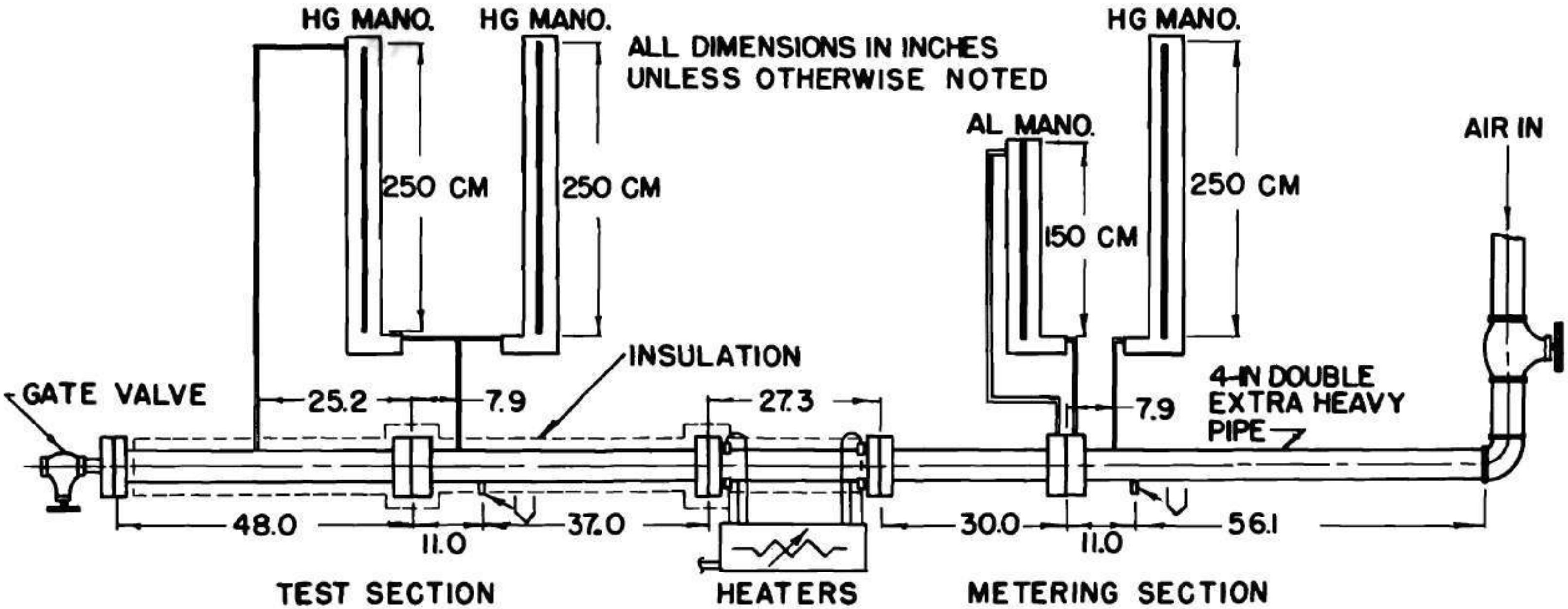
A general layout of the equipment used for the tests is shown in Fig. 1. ASME recommendations (2, 4) were followed where applicable to determine the configuration.

An electrically powered reciprocating compressor of approximately 350 cfm capacity was used as an air supply in combination with a storage and a surge tank and a pressure regulator. This capacity was great enough to allow constant pressure runs to be made.

Double extra heavy, 4 inch pipe of standard steel (inside diameter equal to 3.15 inch) was used for the metering section and all sections downstream of the metering section. Pipe of this diameter was chosen to achieve a negligible approach velocity at the test section. The extra thickness provided an adequate factor of safety at the working pressures and temperatures.

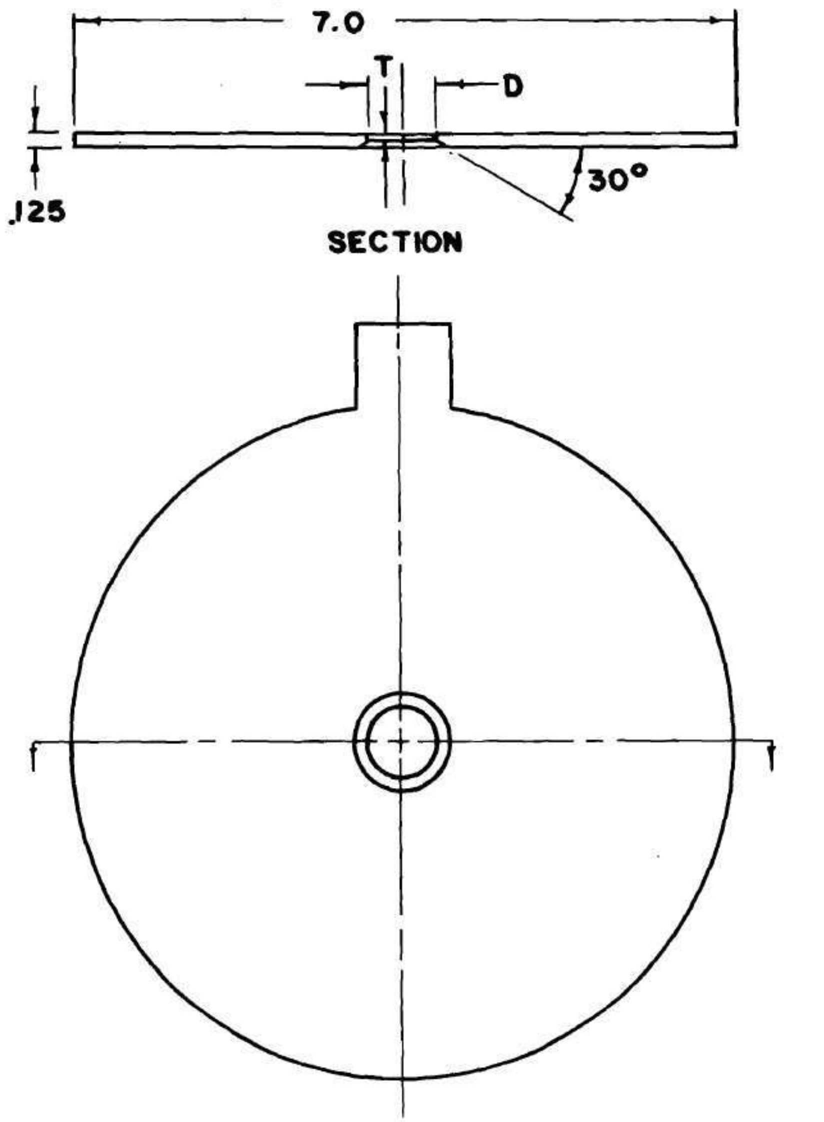
A standard ASME metering orifice as seen in Fig. 2, employing flange taps (4) and having a β ratio of 0.2, was used to determine the mass flow rate. Both the metering and the test orifice plates were made to conform to ASME standards and were constructed from stainless steel plate.

Pipe taps (4) were chosen to determine the pressure loss through the test element. By using pipe taps the permanent pressure loss can be read and the data may therefore be reasonably applied to two large cham-



TEST APPARATUS

FIG. 1



NOTE: T=.031, D=.250 AND .187 FOR TEST ORIFICES
T=.063, D=.630 FOR METERING ORIFICE

ORIFICES
FIG. 2

bers between which the minimum restriction is a plumbing fitting similar to that used in the tests.

It was decided that manometers would yield sufficient accuracy over the large pressure range investigated (30-60 psi). It was found that the rapid dissipation of heat from the copper tubing used as leads to the manometers made unnecessary a consideration of inaccuracy due to heating. A 150 cm cistern-type alcohol manometer was used to determine the pressure drop across the measuring orifice while a 250 cm cistern-type mercury manometer was used to measure the differential pressure across the test section. Two 250 cm cistern-type mercury manometers measured the static pressure in front of the metering and test sections.

The orifices and AN standard fittings tested are given in Table 1. The AN fittings were installed between the flanges of the test section

Table 1

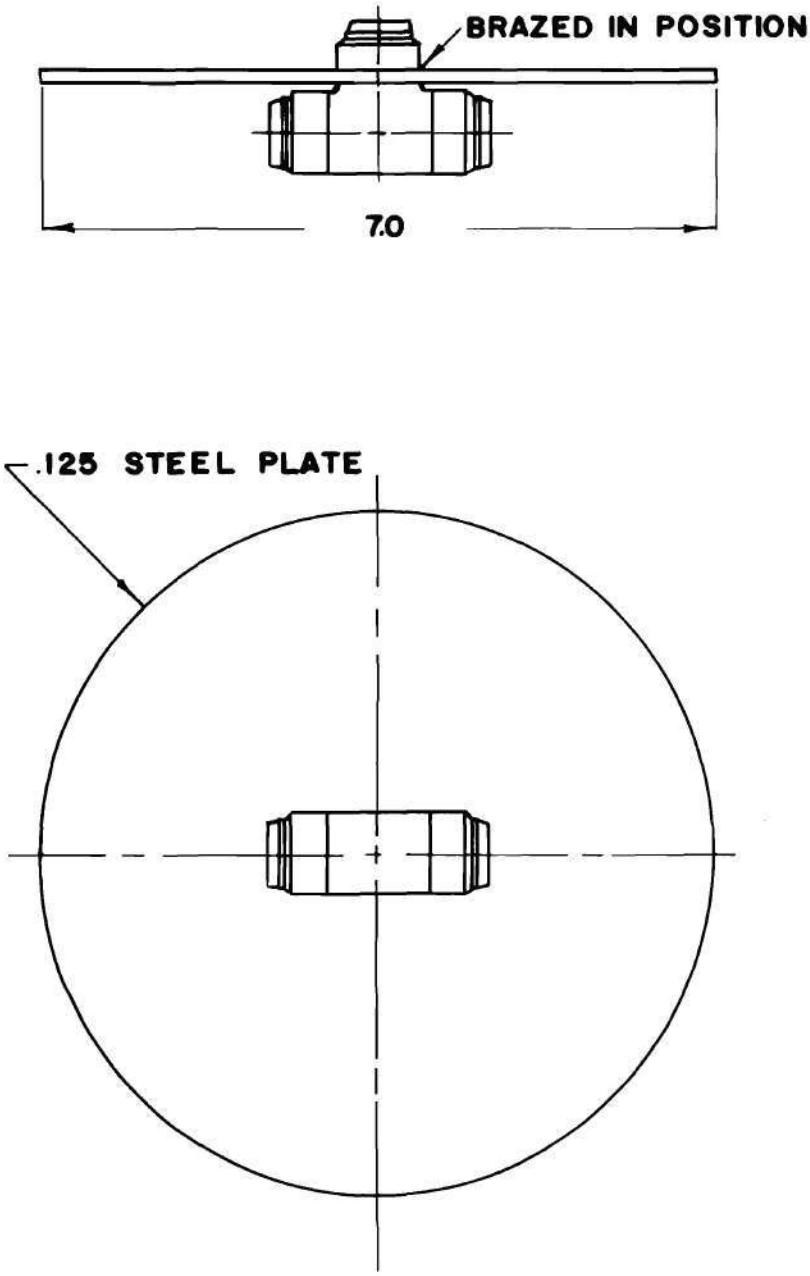
Fittings Tested

Identification	Diameter, D_2	Diameter Ratio, β
	Orifices	
A	0.187	0.0593
B	0.250	0.0793
	AN Fittings	
AN-815-6	0.296	0.0939
AN-815-4	0.169	0.0536
AN-821-6	0.294	0.0933
AN-821-4	0.169	0.0536
AN-824-6	0.295	0.0936
AN-824-4	0.169	0.0536

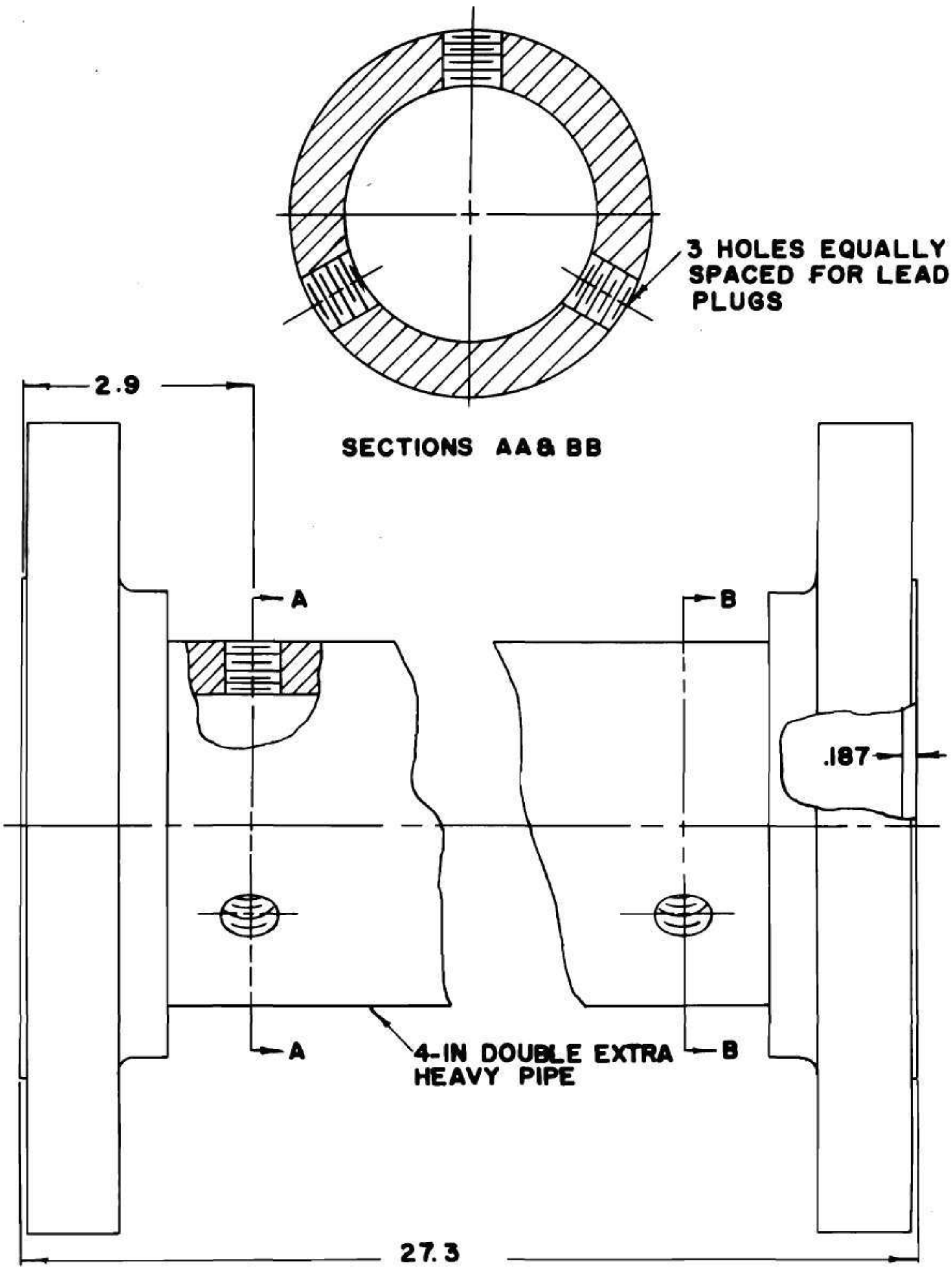
by brazing them in steel plates as seen in Fig. 3. A drawing of the orifices tested appears in Fig. 2.

Drawings of the heater section and its component parts are in Figs. 4 and 5. Heat was supplied by eight Ferrod strip heaters with a rating of 1200 watts each at 220 volts. The heaters were placed in a circular arrangement and mounted inside the pipe on sheet steel end plates as shown in Fig. 5. The downstream end plate was free to move to allow for expansion while the upstream plate was fixed between flanges. The heaters were wired in parallel in banks of two and three. Power was supplied inside the pipe by positioning six Champion igniter plugs as leads as seen in Fig. 4. Copper leads were then run from the electrodes of the plugs to the heater terminals. Current to each bank of heaters was controlled by a rheostat having a continuous range from zero to one-hundred per cent power. The heater section and all sections downstream of this point were insulated with high temperature Kaylo insulation externally wrapped around the pipe.

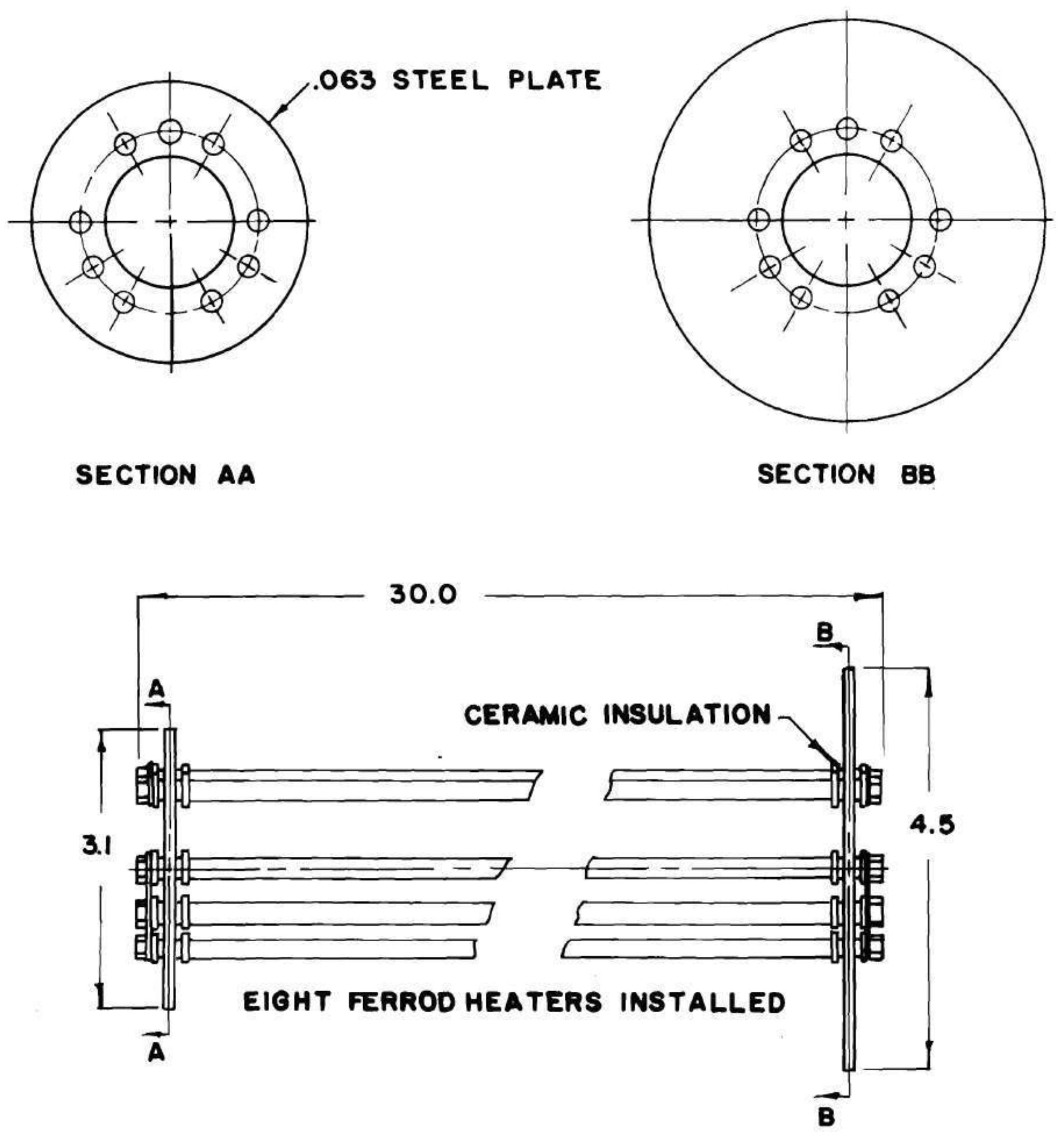
The temperature upstream of both the metering and the test section was measured by thermocouples placed three and one-half pipe diameters from the element. The thermocouples were constructed from .032 diameter chromel-alumel wire and they were not shielded.



TYPICAL AN FITTING INSTALLATION
FIG. 3



HEATER CASE SECTION
FIG. 4



HEATERS

FIG. 5

CHAPTER III

PROCEDURE

The test procedure was as follows: The metering orifice and the test element were first positioned as shown in Fig. 1. Then P_1, the pressure in front of the test section, was obtained by regulating the surge tank pressure while T_1, the temperature in front of the test section, was controlled by regulating the amount of current flowing to the heaters. When stable temperature conditions were reached, ΔP was varied in approximately equal increments (by adjustment of the back pressure P_2 using the downstream valve) for maximum to minimum mass flow as P_1 and T_1 were held constant to within plus or minus 0.2 psi and 10° F respectively. The values of P_1, T_1, the static pressure in front of the metering orifice, the temperature in front of the metering orifice and the pressure difference across the metering orifice were recorded at each value of ΔP.

To investigate the effect of pipe wall proximity on the pressure losses through the tee and elbow fittings all runs for these elements were repeated for identical conditions but in a position with the original upstream and downstream faces reversed.

Each fitting as heretofore discribed was tested at constant P_1 pressures of 30 and 60 psi. At each of these pressures, runs were made while the temperature T_1 was held constant at 500° F and at the storage tank discharge temperature which ranged between 80 and 100° F. The temperature at the metering orifice was not appreciably effected by the nearness of the heaters since the temperature at the metering section was at all times below 100° F.

The choice of a metering orifice with a diameter ratio equal to 0.2 resulted in Reynolds numbers such that standard ASME methods and discharge coefficients (2) could be used to evaluate all flow rate values except a negligible number which were below a Reynolds number of 10,000. The mass rate for these low Reynolds numbers was evaluated by using coefficients determined by Ambrosius and Spink (5). The flow rates thus determined were then applied to the test element.

CHAPTER IV

RESULTS

A development of the basic flow equation as outlined by the ASME (4) will be helpful in justifying the simplifying assumptions made later. Consider sections in front of and at the element illustrated in Fig. 6. Assume that the flow is steady, incompressible and obeys the perfect gas relations; that there is no loss of energy from friction; and that no transfer of heat takes place between the fluid and the surrounding walls. The incompressible Bernoulli equation then gives

$$P_1 + \frac{\rho_1 V_1^2}{2g} = P_2 + \frac{\rho_1 V_2^2}{2g} \tag{1}$$

or

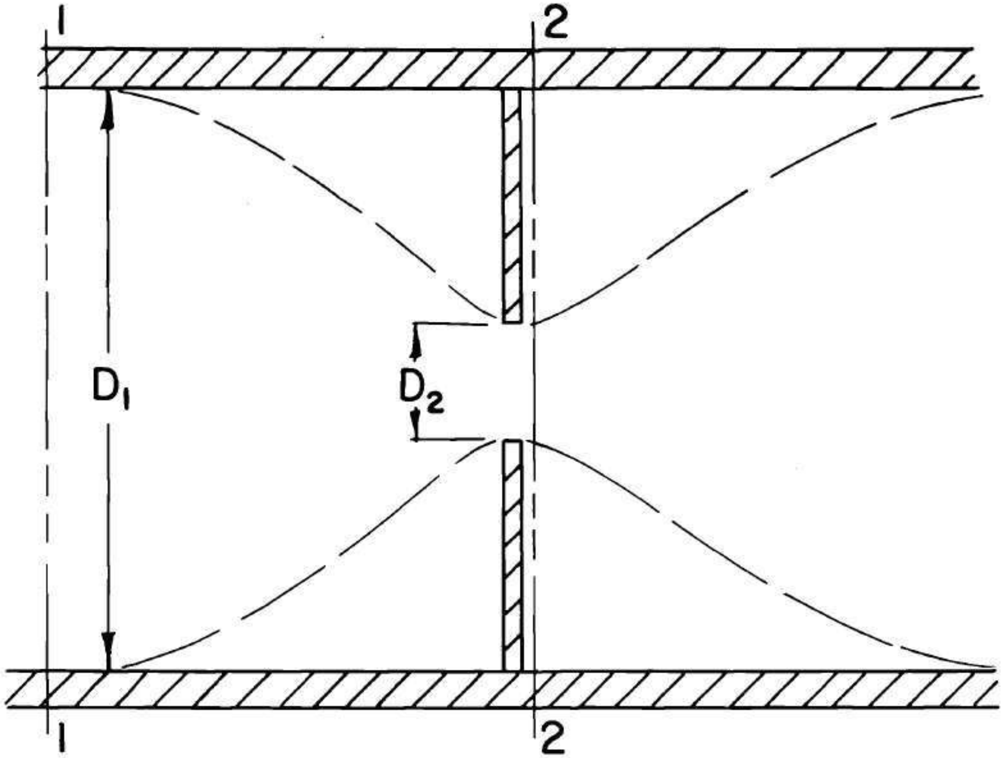
$$\frac{\rho_1}{2g} (V_2^2 - V_1^2) = P_1 - P_2 = \Delta P \tag{2}$$

and from continuity considerations when analyzing a circular cross section

$$w = \rho_1 V_1 \frac{\pi D_1^2}{4} = \rho_1 V_2 \frac{\pi D_2^2}{4} \tag{3}$$

or

$$V_1 = V_2 \left(\frac{D_2}{D_1}\right)^2 = V_2 \beta^2 \tag{4}$$



ELEMENT SECTION

FIG. 6

So that substitution of equation (4) in equation (2) gives

$$V_2 = \frac{1}{\sqrt{1 - \beta^4}} \sqrt{\frac{2\ g\ \Delta P}{\rho_1}}$$

or from equation (3) and the equation of state ($P_1 = \rho_1\ RT_1$) we get

$$w = \frac{A_2}{\sqrt{1 - \beta^4}} \sqrt{\frac{2\ g\ P_1\ \Delta P}{RT_1}} \tag{5}$$

Since the actual flow will vary from the theoretical it is necessary to introduce the empirical coefficients K, Y, and E (4). Where

$$K = C / \sqrt{1 - \beta^4}$$

an empirical discharge coefficient with the velocity of approach factor included; the term Y is a compressibility factor that is a function of the β ratio and the pressure ratio r; and the coefficient E corrects for thermal expansion of the element. So that finally for air

$$w = 1.10\ A_2 KYE \sqrt{P_1 \Delta P / T_1} \tag{6}$$

It is now possible to apply the conditions of the test to the preceding relation. The discharge coefficient

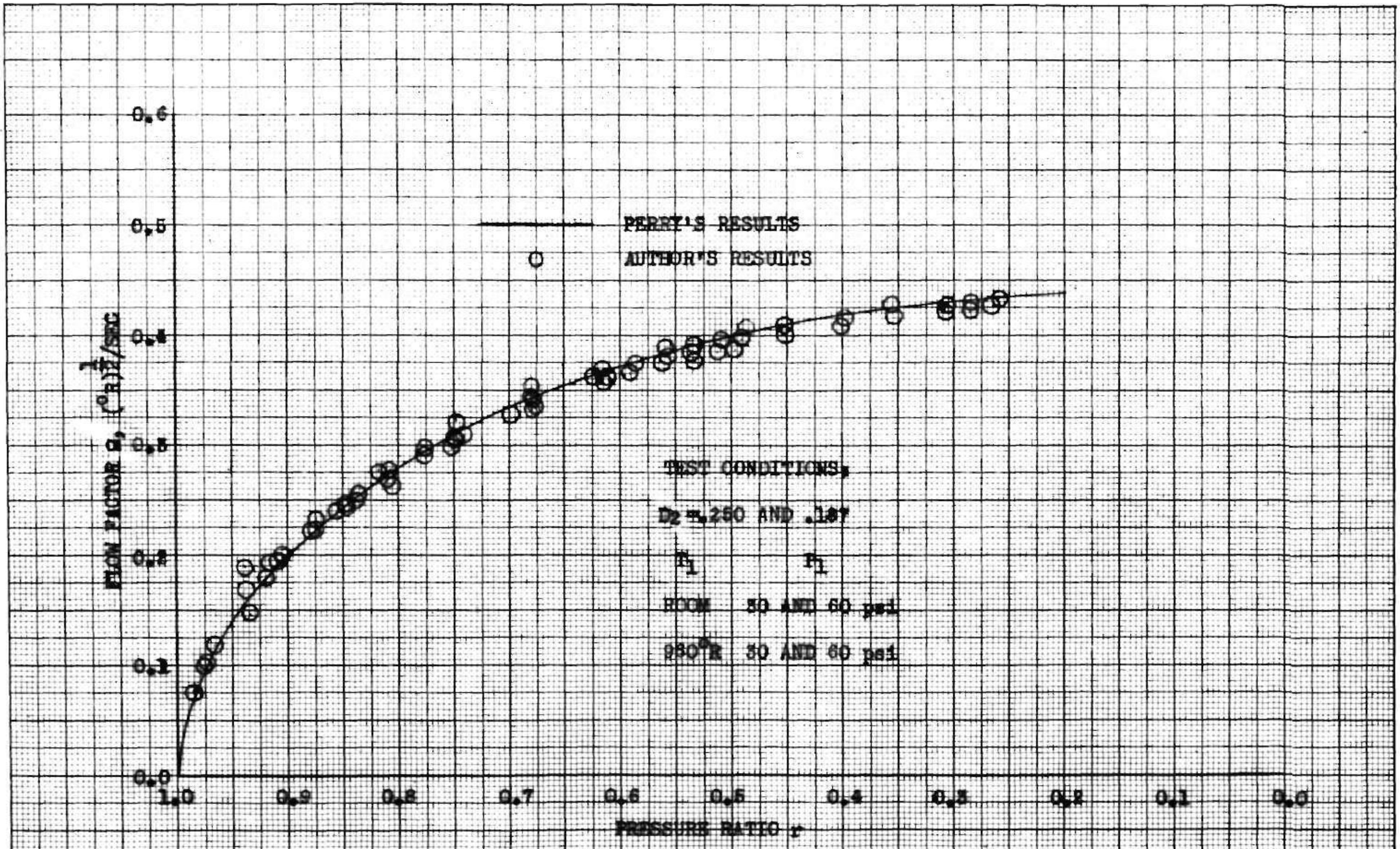
$$K = C / \sqrt{1 - \beta^4} = f(Rn, \beta, D_1, \text{shape})$$

However, for small diameter ratios and geometrically similar fittings the effect of pipe size, β and shape tend to become negligible. In addition K is essentially a constant over a wide range of Reynolds numbers so that the discharge coefficient will be assumed constant for purposes of the test under consideration (where Rn varied from approximately 10,000 to 380,000). For very small values of β the expansion factor Y is significantly a function of r alone. It will be shown that the mass flow varies directly as the area of the element. Also it will be shown that the flow rate varies inversely as the square root T_1 within the limits of the analysis. In addition it is assumed that the effect of E is negligible over the temperature range investigated. So that following Perry (1) and rearranging equation (6) and dividing both sides of the relation by P_1 it is seen that

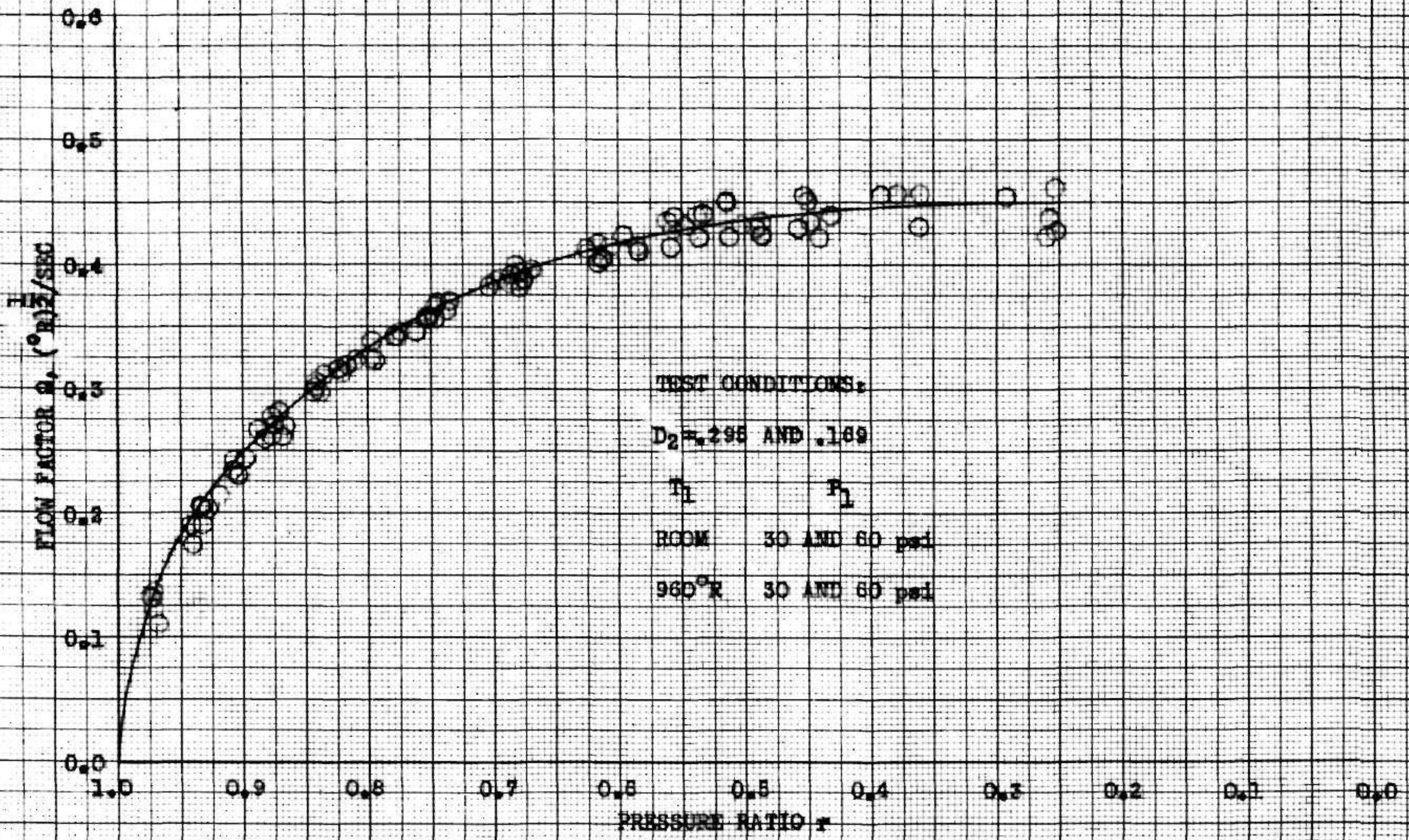
$$\frac{w\sqrt{T_1}}{P_1 A_2} = \Omega = f(r) \tag{7}$$

The values of Ω were calculated for each test run made. Plots of Ω vs r for the orifices and the fittings tested appear in Fig. 7 and Figs. 8, 9, and 10 respectively.

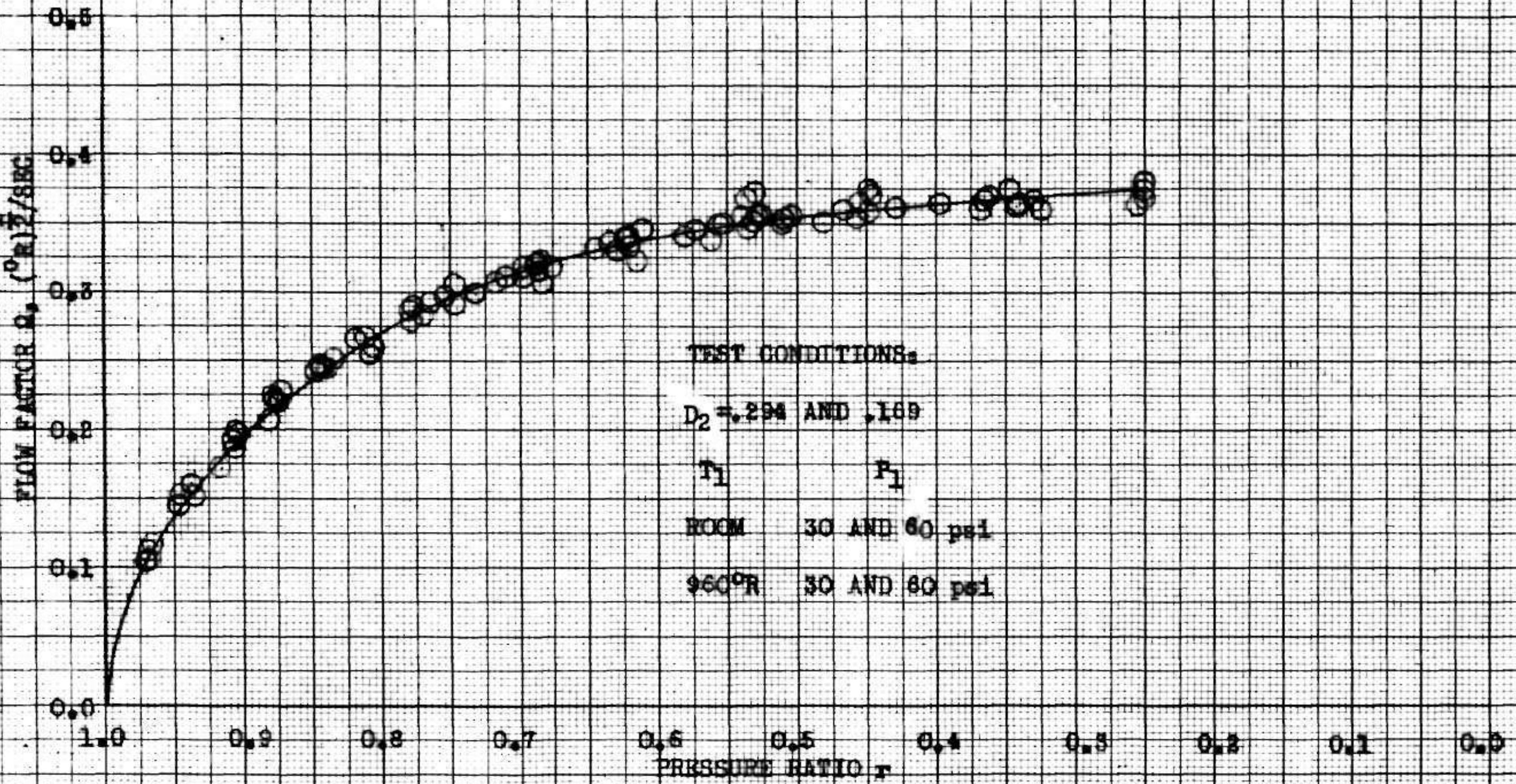
In Fig. 7 the author's test results for orifice flow at room temperature and at 500° F are compared with those of Perry (1) which were obtained under similar conditions and restrictions but entirely at room temperature. It can be seen that the agreement is excellent over the entire pressure ratio range investigated. It is evident from the data presented that the assumptions that mass flow rate varies inversely as the square root of T_1 and that Ω is a pure function of r are valid for the flow conditions stipulated.



FLOW THROUGH SHARP-EDGED ORIFICES
FIG. 7

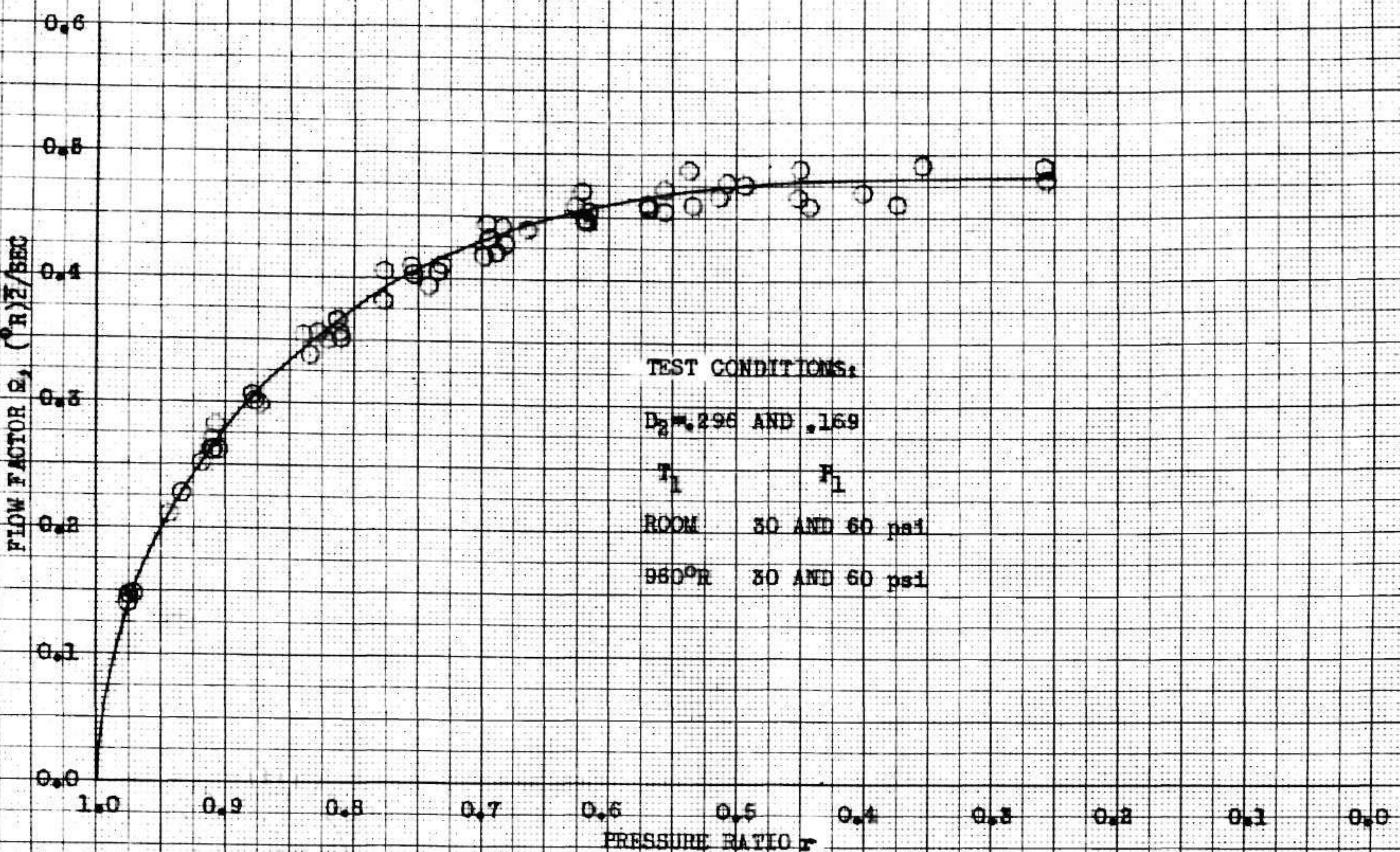


FLOW THROUGH AN-824 TEE FITTINGS
FIG. 8



FLOW THROUGH AN-821 ELBOW FITTINGS

FIG. 9



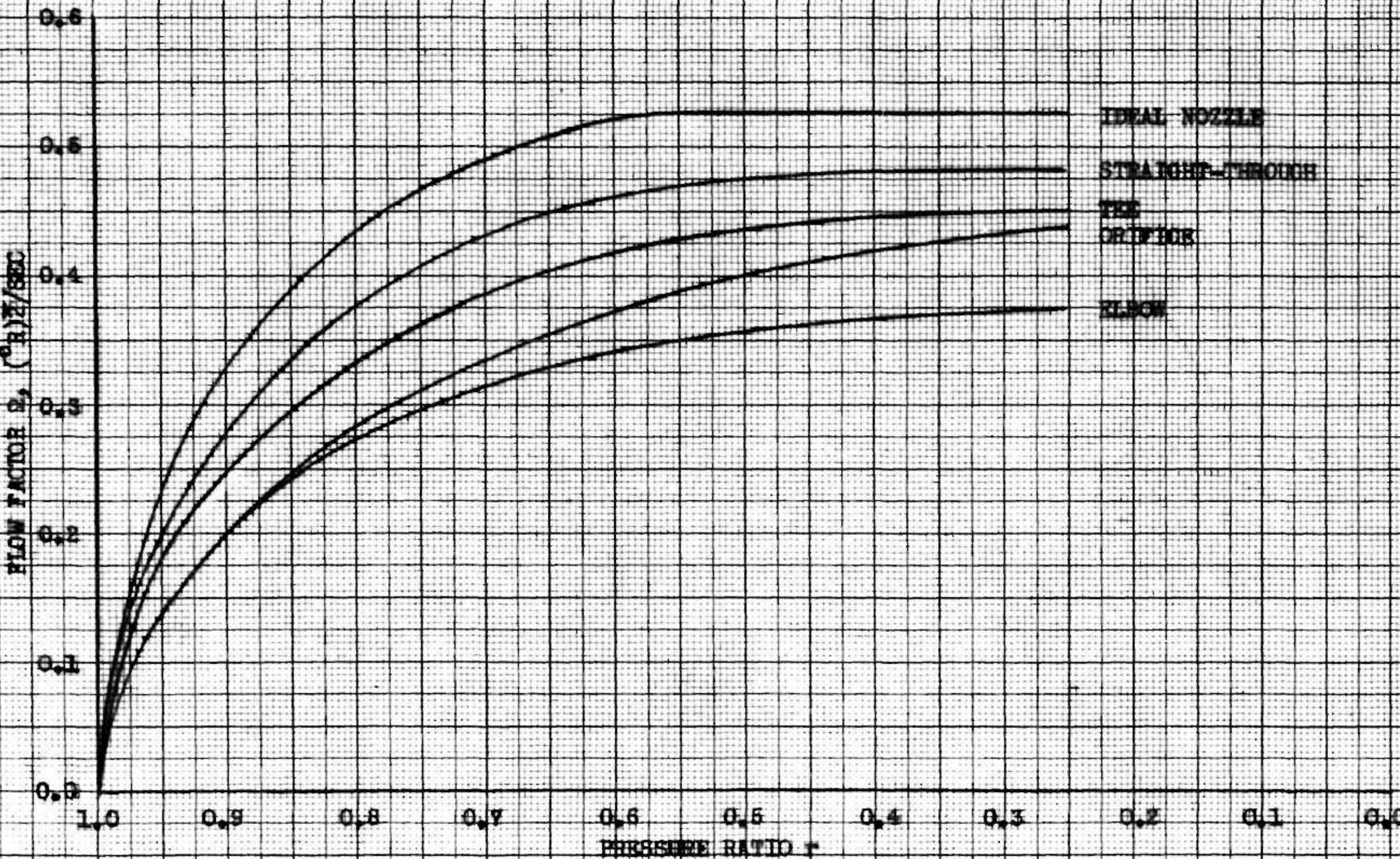
FLOW THROUGH AN-816 STRAIGHT-THROUGH FITTINGS
FIG. 10

In Figs. 8, 9 and 10 plots of Q vs r for the tee, elbow and straight-through fittings respectively are presented. Here again it is evident that the stated assumptions are valid for the flow conditions imposed on the test fittings. When applying this data the conditions of the test must be remembered; that is, a permanent pressure drop due principally to free expansion and turning (in the case of the tees and elbows) losses has been obtained. As previously stated, however, this information could logically be applied to two large chambers between which the minimum restriction is a similar plumbing fitting. In addition a fair estimate of the turning losses through the tee and elbow fittings may be had by subtracting the total loss for the straight-through fitting from the total loss through these fittings.

The maximum deviation of the test points from the average curve occurred in the critical flow range and was of the order of 5.0 per cent for the tee fittings. However, the average deviation in the subcritical range was considerably less than this for all fittings tested.

In Fig. 11 composite plots of the test results are compared with ideal nozzle flow so that the relative variation from ideal flow for each fitting may be seen.

Discharge coefficients as a function of pressure ratio have been evaluated by Bennett (6) for the AN tee, elbow, and straight-through fittings.



FITTING FLOW COMPOSITE

FIG. 11

CHAPTER V

CONCLUSIONS

From the data obtained in this investigation the following conclusions have been reached:

1. The assumption that the mass flow rate of air through an orifice or fitting varies inversely as the square root of the temperature upstream of the element is valid up to 500° F.

2. When analyzing flow through AN-824, AN-821 and AN-815 fittings, the flow factor Ω may be expressed as a pure function of r as shown in Figs. 8, 9, and 10 for the conditions of negligible approach velocity, head temperature below 500° F and Reynolds number above 10,000.

CHAPTER VI

RECOMMENDATIONS

1. It is recommended that this study be extended to include higher flow temperatures. This can best be done by constructing a heater section that is more efficient and has a greater overall power rating than that used in the author's test. One method would be to use a series of heater sections similar to the one used by the author, but each with a power rating that is somewhat less in order to insure a more efficient transfer of heat.

2. An investigation should be made to determine the flow characteristics through fittings when the approach velocities are not negligible, such as would occur in the more realistic tubing and fitting combination.

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