



Stack design for small air flow
by Nelson Alejandro Quintero

A thesis submitted to the Graduate Faculty in partial fulfillment of the the requirements for the degree
of MASTER OF SCIENCE in Mechanical Engineering

Montana State University

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Abstract:

This paper is based on the study of stack design to induce small air inflow rates. The data for the study were obtained by testing six and eight inch diameter double walled stacks of five, ten, and fifteen foot lengths. The testing was conducted in conjunction with the study of the feasibility of ventilating a family size fall-out shelter by means of natural draft ventilation. This made it necessary to expand the testing to include stacks containing elbows, horizontal lengths and stack caps.

The inclusion of these fittings made it possible to investigate the effect of adding restrictions to the system. Such restrictions will provide the designer of a family fallout shelter with a more feasible stack design.

An empirical equation was developed to solve for the amount of available draft provided by the system. The values obtained by solving this equation were in close agreement, throughout the testing, with the measured values. This established the validity of this empirical equation for the conditions existing in the laboratory. After establishing the validity of the equation a computer program was written to solve the equation for stacks of five, ten, fifteen, twenty, twenty-five, and thirty feet height and four, six, eight, and ten inch diameter. The results of the computer program are given in the form of graphs in Chapter 13. The maximum value of draft obtained was 400 ACFM.

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Abstract

This paper is based on the study of stack design to induce small air inflow rates. The data for the study were obtained by testing six and eight inch diameter double walled stacks of five, ten, and fifteen foot lengths. The testing was conducted in conjunction with the study of the feasibility of ventilating a family size fall-out shelter by means of natural draft ventilation. This made it necessary to expand the testing to include stacks containing elbows, horizontal lengths and stack caps. The inclusion of these fittings made it possible to investigate the effect of adding restrictions to the system. Such restrictions will provide the designer of a family fallout shelter with a more feasible stack design. An empirical equation was developed to solve for the amount of available draft provided by the system. The values obtained by solving this equation were in close agreement, throughout the testing, with the measured values. This established the validity of this empirical equation for the conditions existing in the laboratory. After establishing the validity of the equation a computer program was written to solve the equation for stacks of five, ten, fifteen, twenty, twenty-five, and thirty feet height and four, six, eight, and ten inch diameter. The results of the computer program are given in the form of graphs in Chapter 13. The maximum value of draft obtained was 400 ACFM.

CHAPTER 1

INTRODUCTION

The purpose of this paper is to provide the reader with the basic theory of stack design and with a set of graphs and tables which will permit him to obtain the amount of air, in actual cubic feet per minute, available under a specific set of conditions.

Since tests could not be performed on every combination of stack length and diameter available, it was decided to develop an empirical equation which would solve the problem and to conduct testing on six and eight inch diameter stacks of different lengths in order to check the validity of the empirical equation. The values obtained by making measurements in the laboratory closely agreed with the values obtained by solving the empirical equation. This indicated that this empirical equation will give accurate results when the system used is the same as the one used in the experiments.

After establishing the validity of the empirical draft equation a computer program was written to solve the empirical equation for various sizes of stack lengths and diameters. The results of the computer program are given in the form of graphs in Chapter 13.

Because the sole presentation of the graphs would not satisfy the interested reader, as they would not provide him with the information of how these graphs were obtained, a detailed presentation is given of the experimental procedure followed by a brief discussion on the basic theory of stack design. This discussion will give the reader enough information to accurately interpret the results presented in this paper.

CHAPTER 2

BASIC EQUIPMENT

The experimental part of this study was conducted in an air-tight room situated underneath the Mechanical Engineering Power Laboratory in Ryon Laboratory, Montana State University. The inside dimensions of the room are twenty-eight feet long, six feet wide and six feet high.

The basic equipment used in performing the testing was:

1. Stack
2. Temperature measuring apparatus
3. Air measuring apparatus
4. Fuel measuring and burning apparatus
5. Pressure measuring apparatus

The chimney was built of double walled tube sections, the inside wall of these tubes is made of aluminum and the outside wall of galvanized steel; the tube sections are five feet long and of six and eight inch diameter. By joining the tube sections, stacks of different lengths can be easily built.

The temperature measuring equipment consists of thirty gage iron-constantan thermocouples connected to temperature recording and indicating potentiometers.

The air flow measurement was approached in various ways. The original method was to use thermistors. The results obtained by this method were so irregular, would change from day to day, that it was decided to abandon this method. It was then decided to use a windmill type anemometer; this method provided an easier and more accurate way of measuring the air inflow.

The fuel measuring apparatus consists of a balance scale and stop watch with which the time to burn a fixed amount of fuel was measured. The fuel rate was obtained by dividing the amount of fuel burned by the time necessary to burn it.

The pressure drop measurements were taken with a micromanometer, plastic hoses connected the different tap points to the micromanometer. This set-up provided a satisfactory method of measuring pressure drops as small as .001 inch of water.

One of the three following types of fuel burners were used during the testing: (1) Space heater, see Figure 2; (2) Kerosene burner, see Figure 6; (3) Natural gas heater. The type of burner used depended on the desired value of mean temperature in the stack.

CHAPTER 3

CHIMNEY CONSTRUCTION

It was decided to construct the stack using double walled tubes because of its good insulation properties, the double wall allows the existence of an air film between the walls and thus reduces the heat transfer to the surrounding medium. The reduction in heat transfer to the surrounding medium results in higher mean temperatures in the stack with the same fuel consumption.

Testing was conducted on stacks of two different configurations; straight vertical stacks and stacks containing two 90° elbows in the lower part. The straight stack was used to reduce variables to a minimum. Adding the two 90° elbows at the lower part allowed investigation of the performance of the stack under conditions simulating the probable installation where turns in the duct work must be present to minimize radiation streaming. Stacks of six and eight inch diameter were tested at vertical heights of five, ten, and fifteen feet.

Pressure taps and thermocouples were placed along the vertical height of the stack. See Figure 1. The exact position of the taps and thermocouples will be described in the sections on pressure and temperature measurements.

The entrance to the stack consists of a bottomless rectangular hood, see Figure 3. The dimensions of the hood are: height, 13 inches; width, 10 inches; and length, 17 inches.

The burners were positioned under the hood, see Figure 2, in such a way that the hot gases would go up the stack with a minimum loss of heat from the hot gases to the surrounding medium.

The chimney was situated on the end of the testing room opposite to the air entrance opening. See Figure 4. This allowed the air surrounding the chimney entrance to be undisturbed.

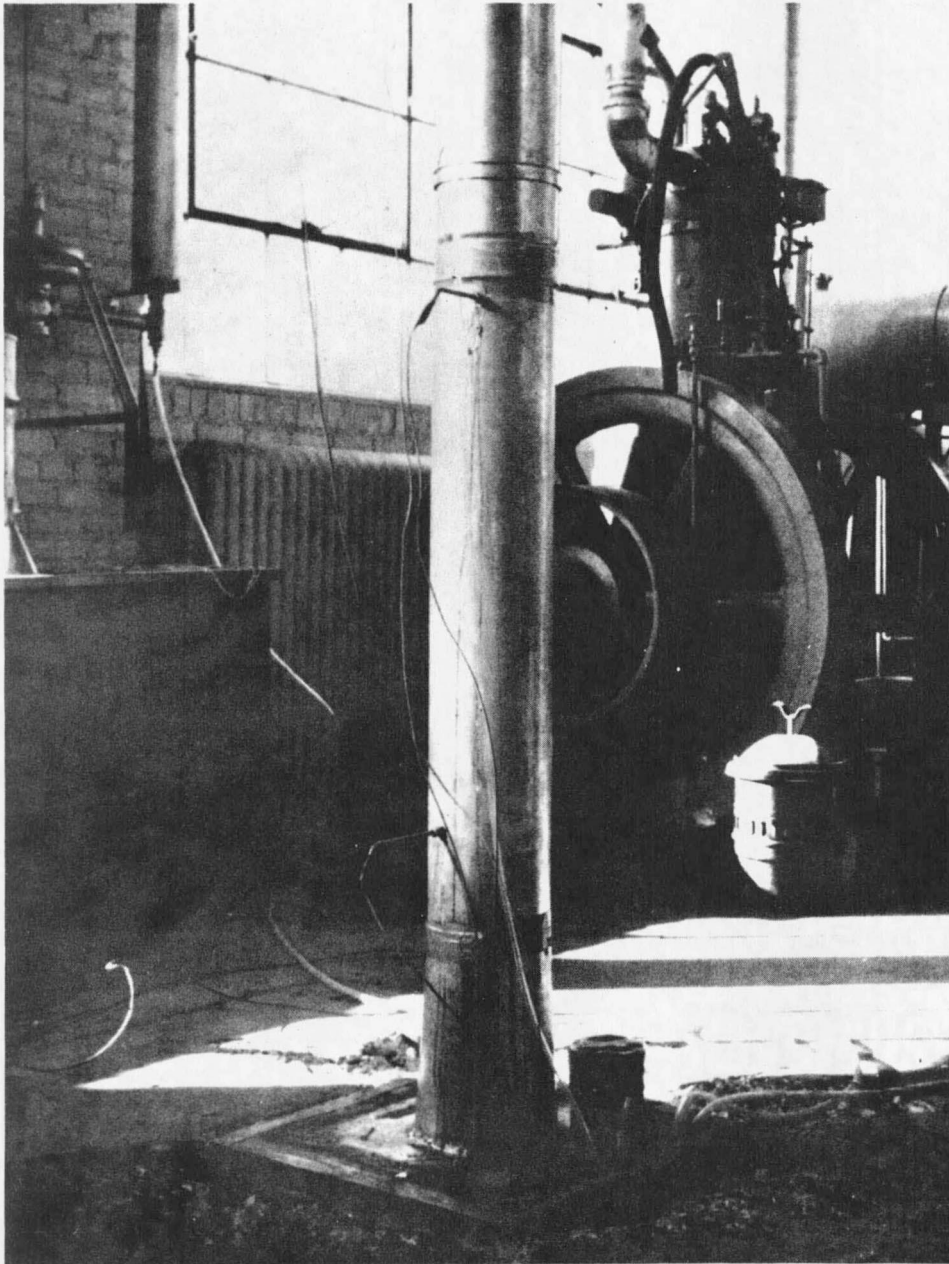


Figure 1 UPPER PORTION OF STACK

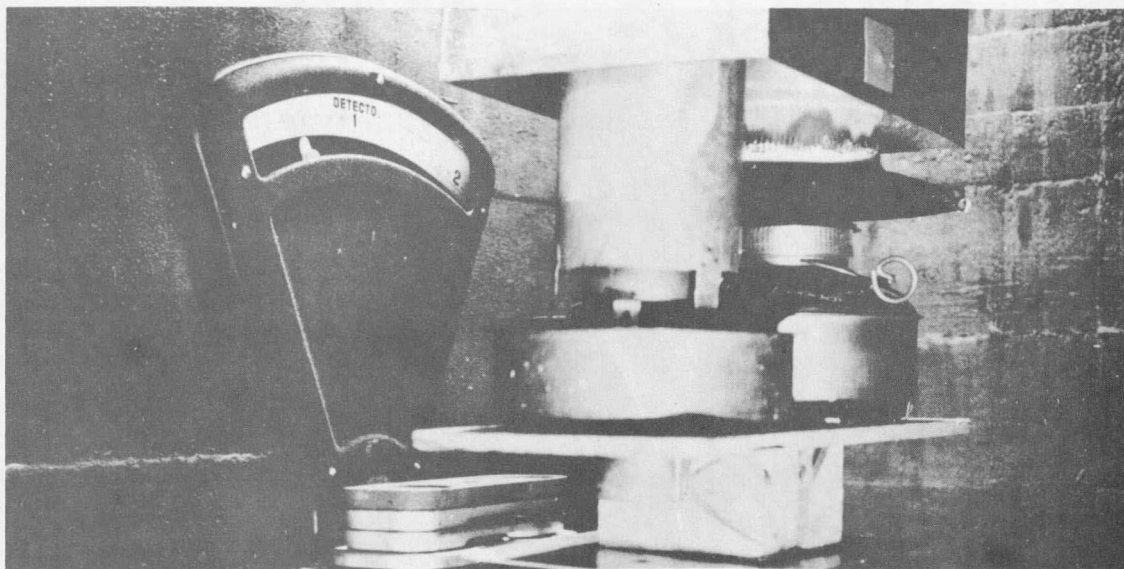


Figure 2 FUEL MEASURING APPARATUS

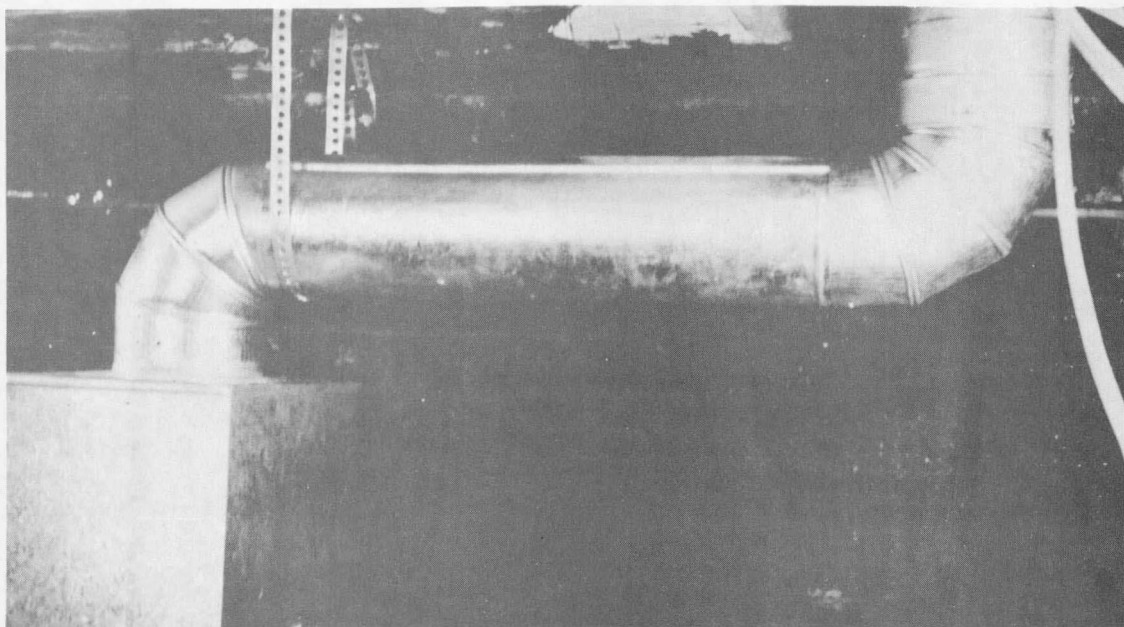


Figure 3 LOWER PART OF STACK

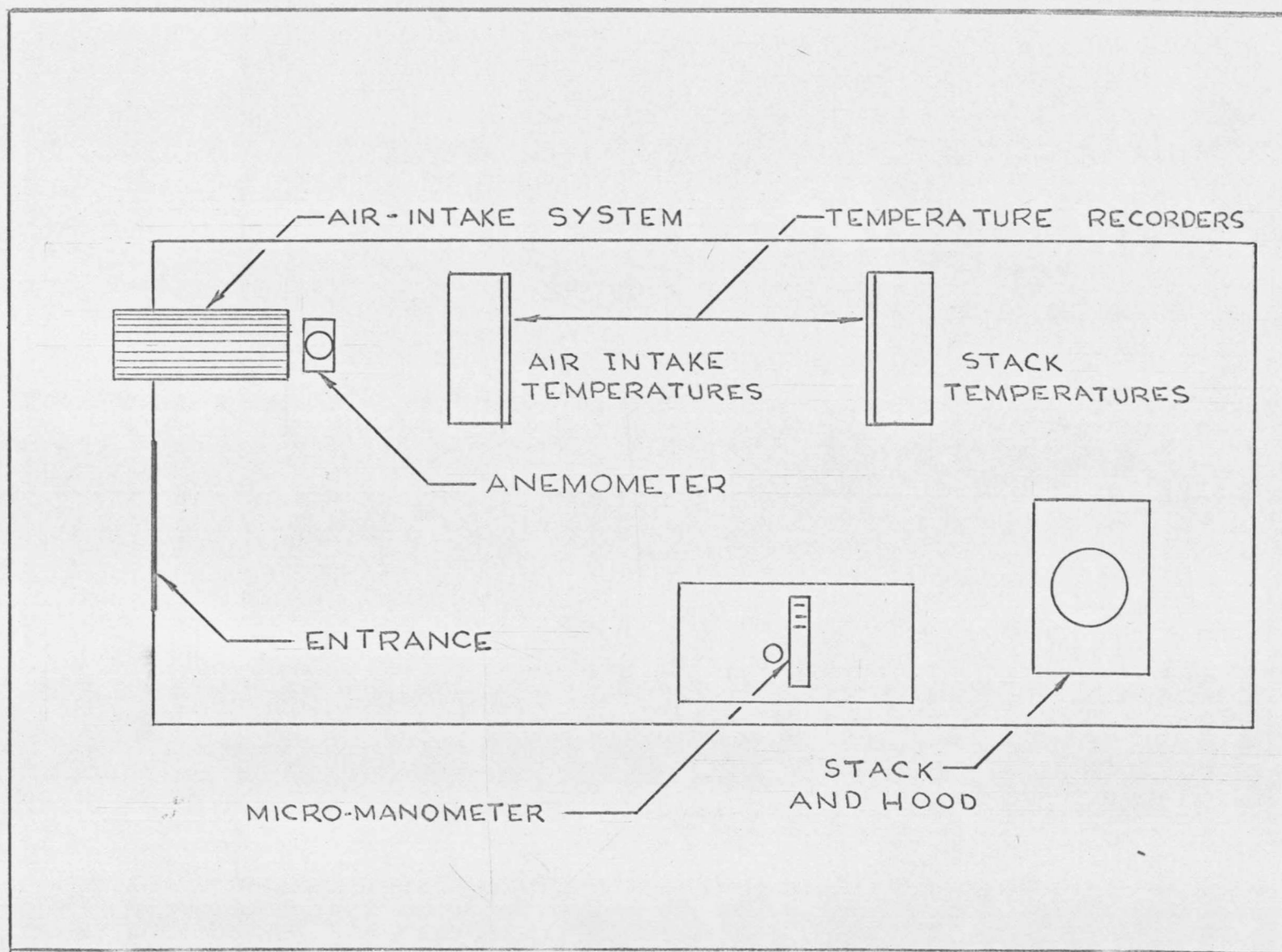


Figure 4 GENERAL LAYOUT OF TESTING ROOM

CHAPTER 4

TEMPERATURE MEASURING APPARATUS

Iron-constantan thermocouples, size 30 gage, were used for all temperature measurements. The temperature of the inlet air and the temperature of the air in the testing room were measured with type J thermocouples. The ends of the thermocouple wire were resistance welded to obtain the desired junction. Thermocouple lead wire was used to connect the thermocouple junction with the temperature recording and indicating potentiometers. These potentiometers were initially and periodically calibrated by inserting one thermocouple in 32° F ice water and adjusting the indicators as required. The temperature values were read directly from these meters.

To measure the temperatures in the stack a different thermocouple design was necessary due to the higher temperatures and the velocity of the flowing gases. Two basic designs were used, one consisted of passing the thermocouple wires through a steel tube and having a large brass ball over the thermocouple end. The large mass of the ball, in comparison with the mass of the normal thermocouple junction, provided more stable readings. This type was used to measure temperatures of points, within the stack, at distances of five feet or more from the chimney entrance. See Figure 5a. This design did not provide accurate temperature readings at points less than 5 feet from the chimney entrance. The inaccurate readings obtained with the above thermocouple design in this region were caused by the hot gases not having been completely mixed in this region. Also the ball would not only receive heat from the hot gases but also

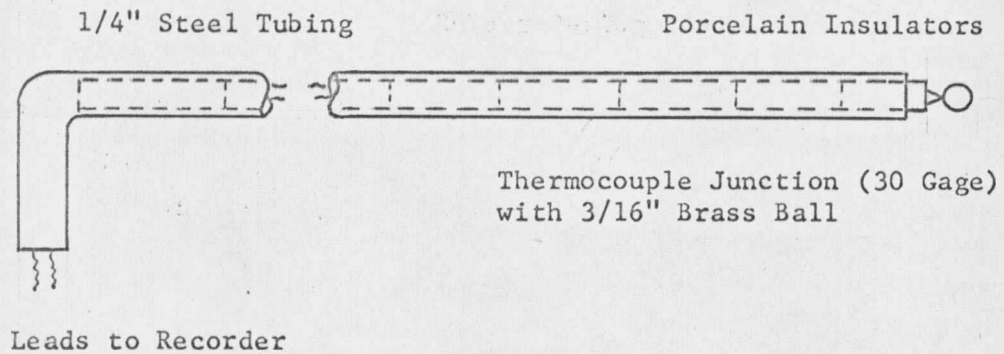


Figure 5a THERMOCOUPLE PROBE

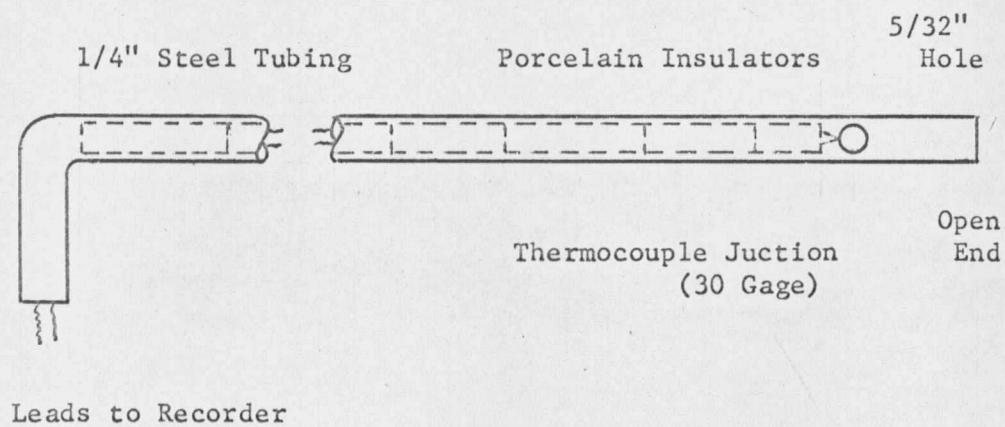


Figure 5b THERMOCOUPLE PROBE

Figure 5 THERMOCOUPLE PROBE DESIGN

radiant heat directly from the flame. To avoid this effect a new design was necessary. This design consisted of a steel tube, 1/4" diameter and 10" long with a 5/32" diameter hole drilled through one side. The thermocouple junction was covered with a brass ball, as before, and the brass ball was placed within the steel tube; the tube was placed so that the opening on the side was perpendicular to the direction of the flowing gases and so that there was no direct line of sight between the junction and the flame. The introduction of this radiation shield considerably reduced the radiant heat transfer to the thermocouple and also increased the thermal inertia of the device so that its sensitivity to transient and/or turbulent variations in the temperature of the fluid was reduced. See Figure 5b for details of this thermocouple probe design.

The thermocouples were placed at distances of 2 1/2, 5, 7 1/2, 10, and 15 feet from the bottom of the stack. Temperature readings were obtained from each thermocouple and from these readings the mean stack temperature was obtained. In order to obtain the average temperature at any stack cross section, plots of temperature variation with stack diameter were made. From these plots it was determined that the average cross-sectional temperature for a 6" diameter stack may be obtained with the probe thermocouples located 2 inches from the near wall.

CHAPTER 5

AIR MEASURING APPARATUS

The air measuring system, see Figure 7 and 9, consists of three 4 ft. long, 8 in. diameter, steel ducts. The middle duct, the only one used during the testing, is equipped with flow straightening vanes at the entrance to insure a straight flow at the duct outlet.

The measuring of the quantity of incoming air was done with a windmill type anemometer. This anemometer consists of a bladed wheel which rotates freely at a velocity proportional to the velocity of the incoming air. The rotating wheel is geared to a needle which rotates over a measuring dial. By dividing the difference in dial readings by the time interval between the readings, a number proportional to the velocity of the incoming air can be determined.

In order to obtain the value of the true air velocity the anemometer had to be calibrated. The calibration apparatus, see Figure 10, consists of a laminar air flow element produced by the Meriam Instrument Company, connected at one end to an air blower and at the other end to an air intake piping system. The amount of air going through the laminar flow element can be regulated by manually controlling the size of an opening situated between the laminar flow element and the blower. The laminar flow element gives a linear relation between weight flow of air and pressure drop in the element. The pressure drop in the element was read with an inclined water manometer to which the pressure taps of the element were connected.

The calibration test runs were conducted in the following way: The windmill anemometer was placed at the outlet of the incoming air duct, and the air intake piping system of the laminar flow element was connected

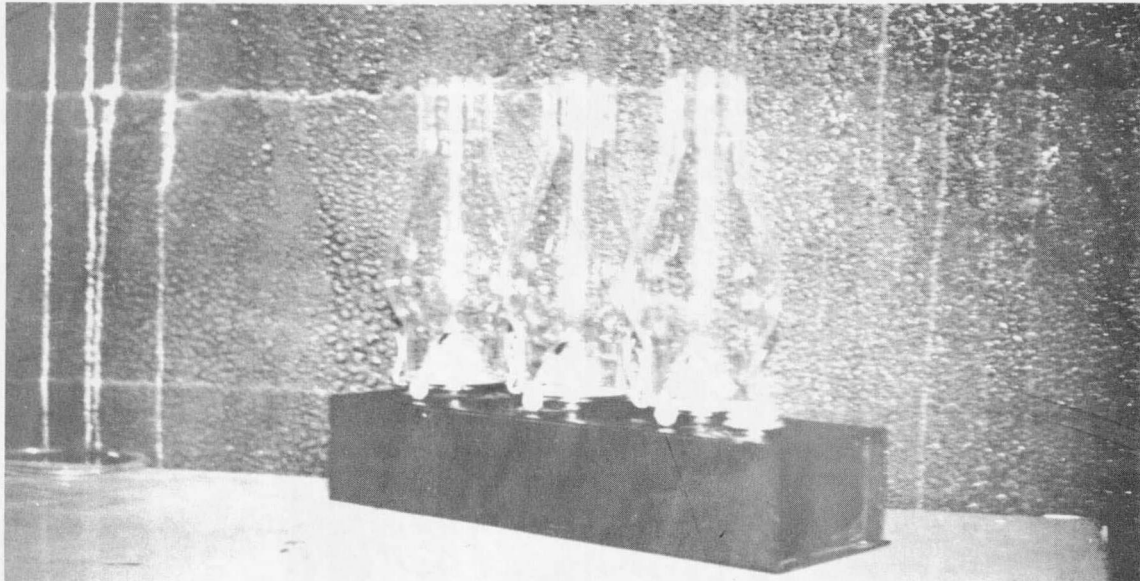


Figure 6 KEROSENE BURNER

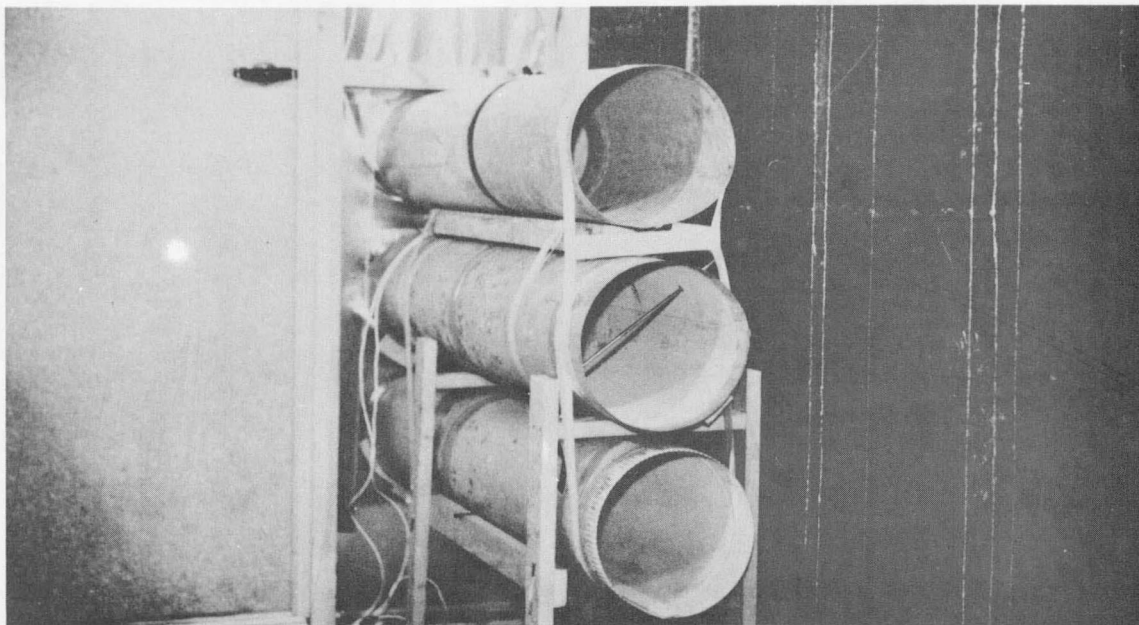


Figure 7 AIR INTAKE SYSTEM

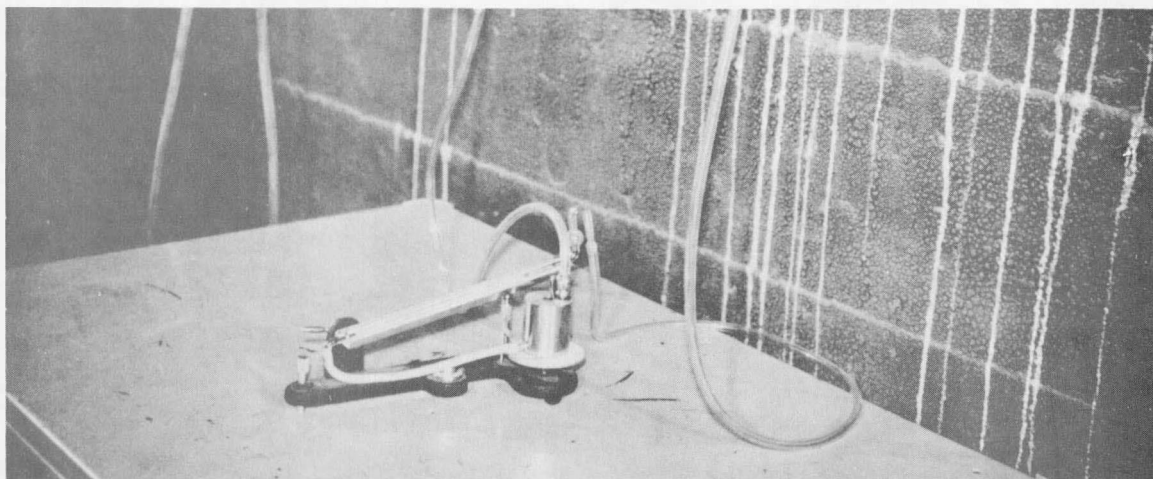


Figure 8 PRESSURE MEASURING APPARATUS

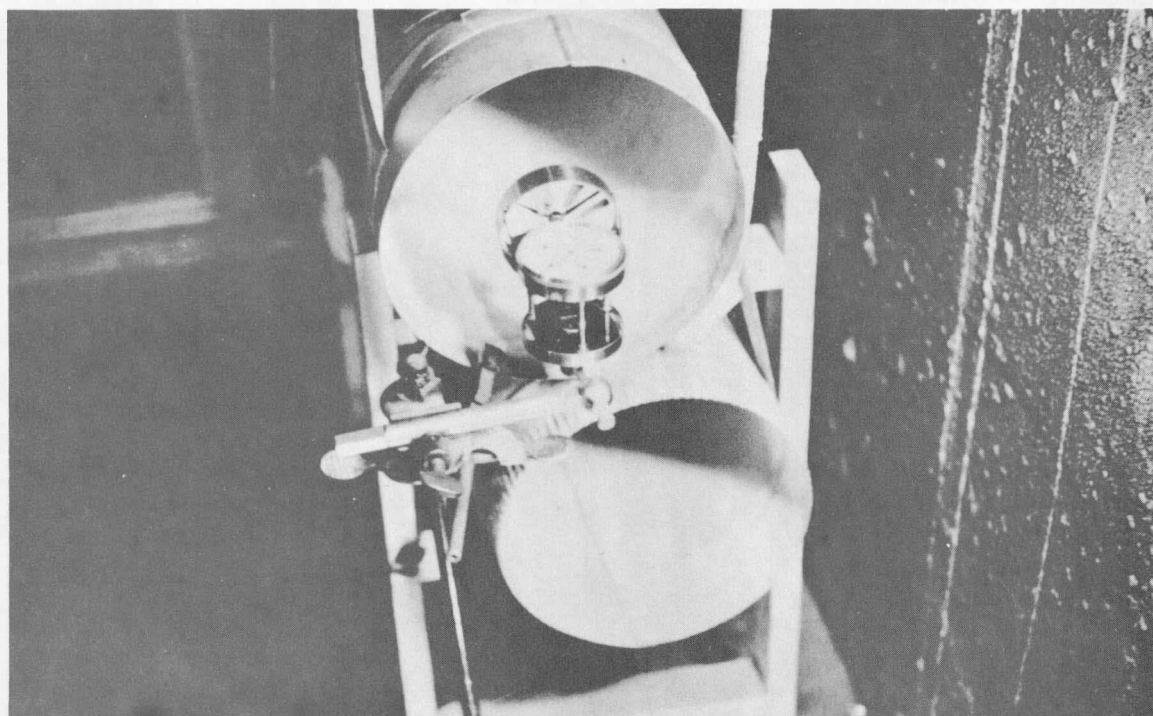
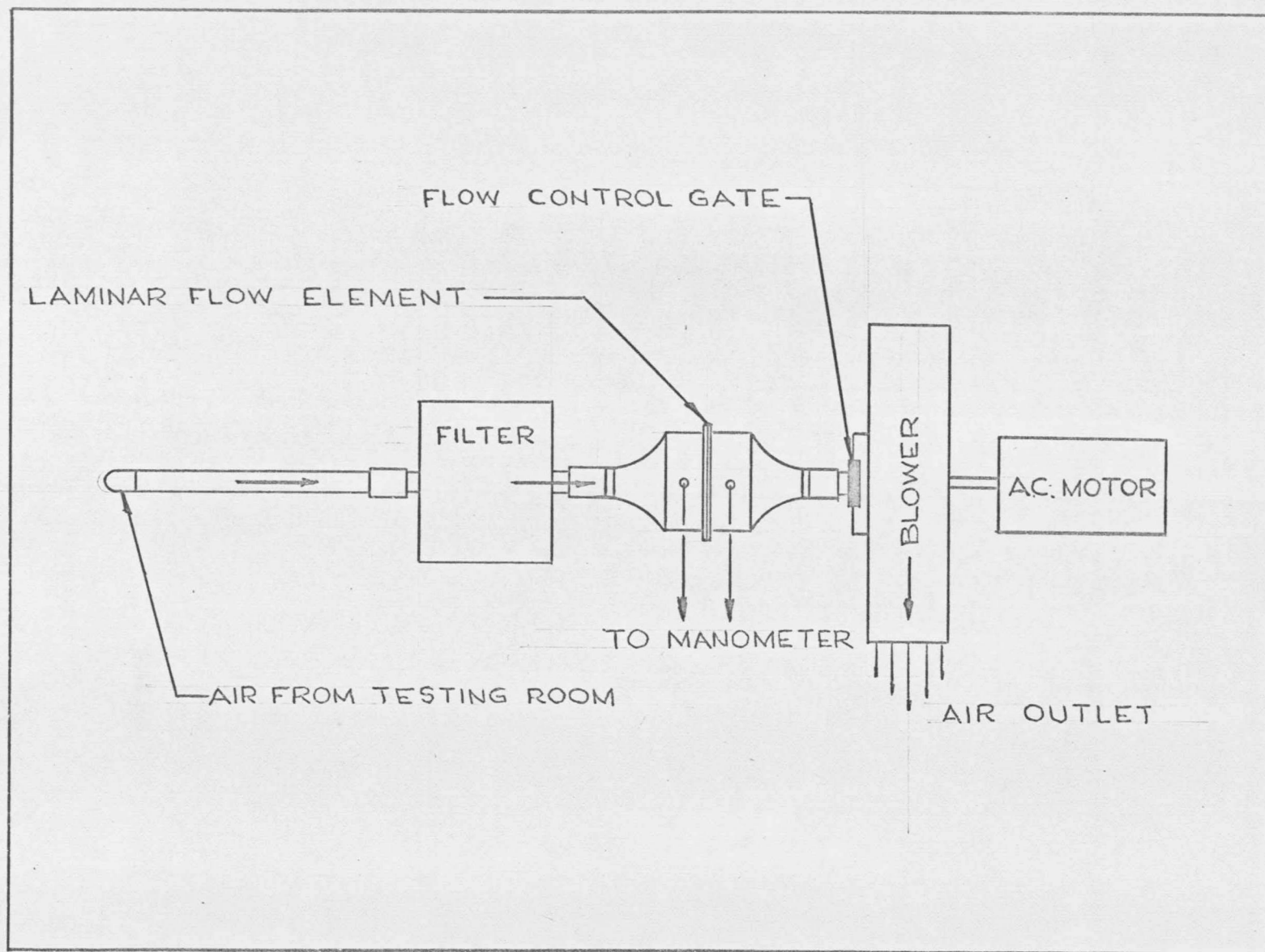


Figure 9 POSITION OF WINDMILL ANEMOMETER



-15-

Figure 10 LAMINAR FLOW ELEMENT - EQUIPMENT LAYOUT

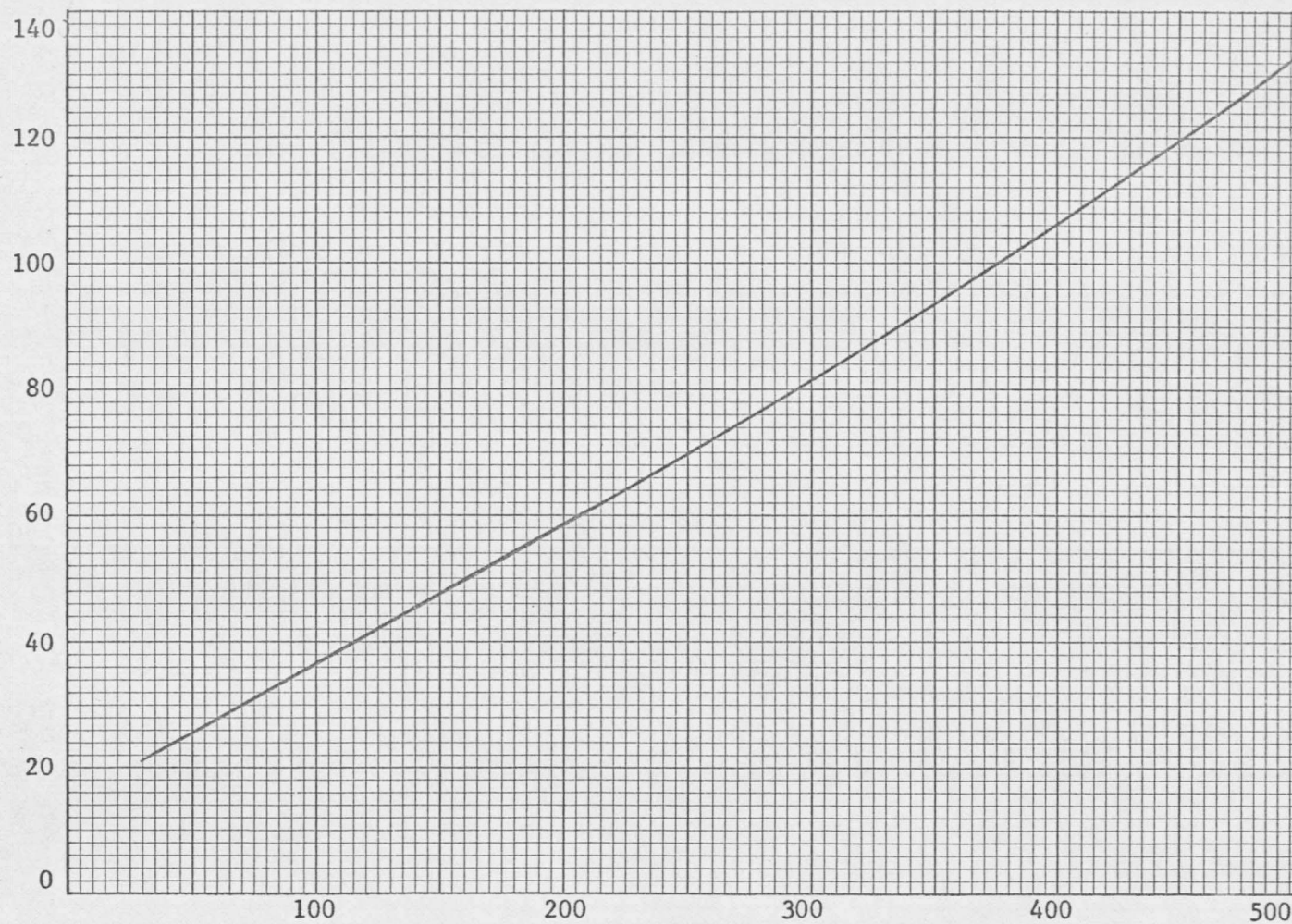
to an opening in the roof of the testing room, besides these two openings the testing room was completely air tight. The blower was then started and the amount of air flowing through the blower, which must be exactly equal to the amount of air coming into the room, was calculated by reading the pressure drop in the laminar flow element. Actual cubic feet of air flowing per minute, were plotted against the velocity rate (difference in windmill dial readings per minute) and a calibration curve was obtained. For a more complete description of the calibration procedure and related calculations, the reader is referred to the "Meriam Instruction Manual A-28836" published by the Meriam Instrument Company and supplied with their laminar flow instruments.

During the testing of the chimney it would suffice to obtain the velocity rate of the windmill anemometer, and from the calibration curve obtain the actual cubic feet of air flowing into the room.

The above method was used to measure the quantity of air flowing into the room through the 8 inch inlet duct. Because the air was not forced to pass through the 8 inch duct during open door tests, the propeller anemometer could not be used to measure the air flow. Instead, a pitot tube installed 7 1/2 feet from the bottom of the chimney was used to measure the air flow during open door tests.

The pitot tube was connected to a micromanometer. From this micromanometer the difference between the static and total pressures was obtained. This difference in pressure was used in the following equation to calculate the velocity of the hot gases.

AIR FLOW RATE - ACFM AT INLET



ANEMOMETER READING - DIFFERENCE IN DIAL READING PER MINUTE

Figure 11 ANEMOMETER CALIBRATION CURVE

$$V_{HG} = C \sqrt{2(P_s - P_1)/\rho}$$

Where V_{HG} = Velocity of the hot gases in the chimney.

C = Pitot tube coefficient, value is either equal to or less than one.

P_s = Total pressure.

P_1 = Static pressure.

ρ = Density of flowing medium.

The pitot tube coefficient was determined by using both the pitot tube and the windmill type anemometer to measure the air flowing into the room through the inlet duct. The velocity value obtained from the windmill anemometer was divided by the velocity value obtained by the pitot tube, the quotient gave the value of the pitot tube coefficient.

CHAPTER 6

FUEL MEASURING & BURNING APPARATUS

The fuels burned during the testing were natural gas and fuel oil #1. Fuel measurements were taken only of the fuel oil #1.

The burning apparatus consists of three different types of burners, the type used at a given time depending on the desired mean temperature of the stack. For low temperatures, the burner shown in Figure 6 was used. This burner consists of a small fuel container with three kerosene lamps, it burned fuel oil #1. For medium stack temperatures, the fuel oil space heaters shown in Figure 2 were used. One or two of the heaters were used at a given time depending on the desired mean temperature in the stack. Because of the fuel consumption characteristics of the space heaters, most of the fuel measurements were taken when the space heaters were being used. The natural gas burners were used when temperatures higher than the temperatures obtained with the two space heaters were desired. Two natural gas burners were used and no fuel measurements were taken, when using this type of burner.

The fuel measuring apparatus can be seen in Figure 2. It consists of a balance scale, which reads weight to the nearest hundredth of a pound. The time interval necessary to burn a given amount of fuel was measured and from these two values the fuel consumption rate was obtained.

CHAPTER 7

PRESSURE MEASURING APPARATUS

The micromanometer shown in Figure 8 was used to measure the pressure drops occurring in various parts of the system. Taps were placed at the bottom of the chimney, at the top of the chimney, inside the testing room and outside the testing room (tap outside the room was used to represent the atmospheric pressure). Connections were made from these taps to the micromanometer such that by means of valves it was possible to obtain the pressure difference between any given tap point and the atmospheric pressure.

Although data were kept for the pressure difference between the top, bottom, and room versus the atmospheric conditions, the only data that were used were the data giving the pressure drop of the intake system. These data pointed out that for the tested system an intake configuration with a pressure drop of over .05 inches of water would make the system inoperable. Since most commercial filters have a pressure drop higher than .05 inches of water it was deduced that the system will not operate under any restrictive filtering conditions. This led to placing a major emphasis on open door testing.

CHAPTER 8

DEVELOPMENT OF THE EMPIRICAL EQUATION

To develop the empirical equation, Bernoulli's equation was applied between points 1 and 2. Where points 1 and 2 refer to the positions shown in Figure 30. Applying Bernoulli's equation between these two points and assuming no heat loss from the stack, we obtain

$$\frac{P_1}{\rho} + \frac{V_1^2}{2g} + h_1 = \frac{P_2}{\rho} + \frac{V_2^2}{2g} + h_2 + \text{Losses}$$

Where $\frac{P_1}{\rho}$ = Static pressure at point 1

$\frac{V_1^2}{2g}$ = Velocity pressure at point 1. Since at this point the velocity of the gases is small, the value of this term is taken as zero.

h_1 = Height above given datum. Since the given datum is point 1, the value of this term is zero.

$\frac{P_2}{\rho}$ = Static pressure at point 2.

$\frac{V_2^2}{2g}$ = Velocity pressure at point 2.

h_2 = Height of point 2 above given datum.

Losses = Losses due to friction. Given by $K \frac{L}{D} \frac{V_2^2}{2g}$

Rearranging terms, we have

$$P_1 - P_2 = \frac{V_2^2}{2g} \rho + h_2 \rho + K \frac{L}{D} \frac{V_2^2}{2g} \rho$$

The term in the left side of the equation $P_1 - P_2$, is given by $\rho_0 h$. That is to say; the difference in static pressure between points 1 and 2 is equal to the pressure exerted by a column of air of height h and density ρ_0 . Where ρ_0 is the density of the air outside the stack. Substituting $\rho_0 h$ for the term $P_1 - P_2$ in the above equation, we obtain:

$$h (\rho_0 - \rho) = \frac{V_2^2}{2g} \rho + K \frac{L}{D} \frac{V_2^2}{2g} \rho$$

Where h = Vertical height of the chimney and hood.

ρ_o = Density of air in the room.

ρ = Density of hot gases in the stack.

V_2 = Velocity of hot gases in the stack.

g = Gravitational constant.

K = Dimensionless constant associated with frictional losses in the stack and at the stack entrance.

L = Length of the chimney.

D = Diameter of the chimney.

In this equation the coefficient K is dependent on the size and shape of the entrance to the chimney and on the friction coefficient of the surface of the chimney. A variety of empirical equations to determine the value of K can be found in literature concerning stack design. But instead of determining K using one of these equation, in this presentation its value was determined experimentally.

To determine the value of K experimentally, measurements of temperature, velocity, height, and diameter of the stack were taken. From these measurements all the terms in the previous equation were known except K . Solving this equation for K a value of .064 was obtained. Substituting this value for K in the previous equation, we obtain:

$$h (\rho_o - \rho) = \frac{V_2^2}{2g} \rho + .064 \frac{L}{D} \frac{V_2^2}{2g} \rho$$

This equation is thus the developed empirical equation.

CHAPTER 9

PARAMETERS CONTROLLING STACK DESIGN

The best stack design is that which produced the greatest amount of air flow at the smallest cost. The importance of the cost factor should be kept in mind throughout the design of the stack.

As stated in Chapter 8, the draft equation is given by:

$$h(\rho_o - \rho) = \frac{V^2}{2g} \rho + K \frac{L}{D} \frac{V^2}{2g} \rho$$

From this equation the parameters controlling stack design can be obtained. These parameters are: vertical height, temperature of the room and of the chimney, K factor, diameter of the chimney, and velocity of the hot gases. A brief discussion on how these parameters influence stack design follows in the next paragraphs.

The vertical height of the chimney appears on the left side of the equation and its magnitude helps to determine the size of the buoyant force. All other terms being the same the higher the stack the greater the buoyant force and the induced draft created by the chimney.

The mean temperature of the stack and the temperature of the surrounding air are used to calculate the difference in density between the hot gases and the room air. Either temperature by itself does not control the amount of draft induced by the stack, but the difference in these temperatures determines the difference in the densities and thus is directly related to the draft induced by the chimney. From equation 8.1 it can be concluded that to obtain the maximum value of draft the difference in these temperatures should be as large as possible. In actual practice the temperature difference is limited by the temperature of the surrounding air,

and by the maximum obtainable gas temperature. The maximum obtainable gas temperature in the stack is limited by the available heating apparatus and by the thermal losses from the stack to the surrounding medium.

The coefficient K is related to the restrictions offered to the flow by the shape and size of the entrance, and the surface characteristics of the material used in the construction of the chimney. The higher the restrictions to the flow, the higher the value of K and the lower the amount of available draft. Therefore when designing a chimney the restriction to the flow should be kept to a minimum.

For a vertical stack the H and L terms in the equation developed in Chapter 8 are equal. When a stack is not completely vertical (contains horizontal sections, bends, or fittings) the term L is no longer equal to H . In this case L is the equivalent length of the system, where equivalent length is defined as the length of a straight pipe of a given size that would have the same pressure drop as the actual pipe with its bends, fittings, and other devices. Other conditions being equal the higher the value of L , the more resistance offered to the flow and the smaller the flow through the system.

The diameter of the stack must be such as to permit delivering, at a reasonably low velocity, the volume of the flue gases generated. Large ducts will lower the velocity of the gases and reduce the frictional losses, but the space and investment required offset the advantages.

CHAPTER 10

STRAIGHT VERTICAL STACK - EXPERIMENTAL PROCEDURE

The simplest possible configuration, a straight vertical stack with no air restriction at the room entrance, was tested first because of the fewer losses. For these conditions the following equates the positive bouyant force of a stationary column of hot gases with the force of friction required to establish and maintain flow.

$$\frac{H}{5.2} (P_{RM} - P_{CH}) = \left(\frac{V}{1096}\right)^2 P_{CH} + K \frac{L}{D} \left(\frac{V}{1096}\right)^2 P_{CH}$$

Where H = Vertical stack height, feet.

5.2 = Conversion factor, quotient 62.4/12, the density of water and inches per foot, so that both sides of the equation can be expressed in inches of water.

P_{RM} = Density of room air, pounds per cubic foot.

P_{CH} = Density of stack gas, pounds per cubic foot.

V = Stack gas velocity, fpm.

1096 = Conversion factor, feet of air to inches of water.

D = Stack diameter, feet.

K = Dimensionless constant. As defined in Chapter 8.

The main purpose of the tests was to verify the validity of the empirical equation developed in Chapter 8. In order to do so a computer program was prepared to solve the empirical equation. The computer was fed with values of length, diameter, pressure, temperatures, and K factor and solved the empirical draft equation for the values of the velocity of the gases in the stack and the actual cubic feet of air flowing into the room. These computed values were then compared with the measured values of air flowing into the room. The results of these tests are

plotted on Figures 12 and 14. As can be seen in these figures the values obtained by solving the empirical equation and the measured values agree closely and thus the validity of the empirical equation was established for the straight vertical stack with no restriction to the air flowing into the room.

The second step was to add an air restriction, at the room entrance. This air restriction was added in order to study the performance of the stack under situations in which the outside air would require filtering before bringing it into the room. The air restriction consisted of passing the incoming air through the intake duct, shown in Figure 4. The dimensions of the intake duct are: 8" diameter and 3.83 feet long. The entrance factor K for the intake duct was obtained experimentally by applying Bernoulli's equation between a point before the entrance to the duct and a point after the entrance to the duct, and measuring the pressure drop across the entrance with a micromanometer. The value of K for the entrance duct obtained in the laboratory is 3.38.

Thus the empirical equation obtained for the system with an air restriction at the intake is given by:

$$\frac{H}{5.2} (P_{RM} - P_{CH}) = K_1 \left(\frac{V_1}{1096} \right)^2 \rho_1 + \left(\frac{V_2}{1096} \right)^2 \rho_2 + K_2 \frac{L_2}{D_2} \left(\frac{V_2}{1096} \right)^2 \rho_2$$

Where $K_1 = 3.38$ and $K_2 = .064$. And the meaning of the terms is the same as in the previous equation. Subscript 1 refers to conditions at the air inlet and subscript 2 to condition in the chimney.

Solving this empirical equation for different lengths and diameters of stack Figures 13 and 15 were obtained. Where the solid lines indicate

the values obtained by solving this equation and the dashed lines indicate the measured values.

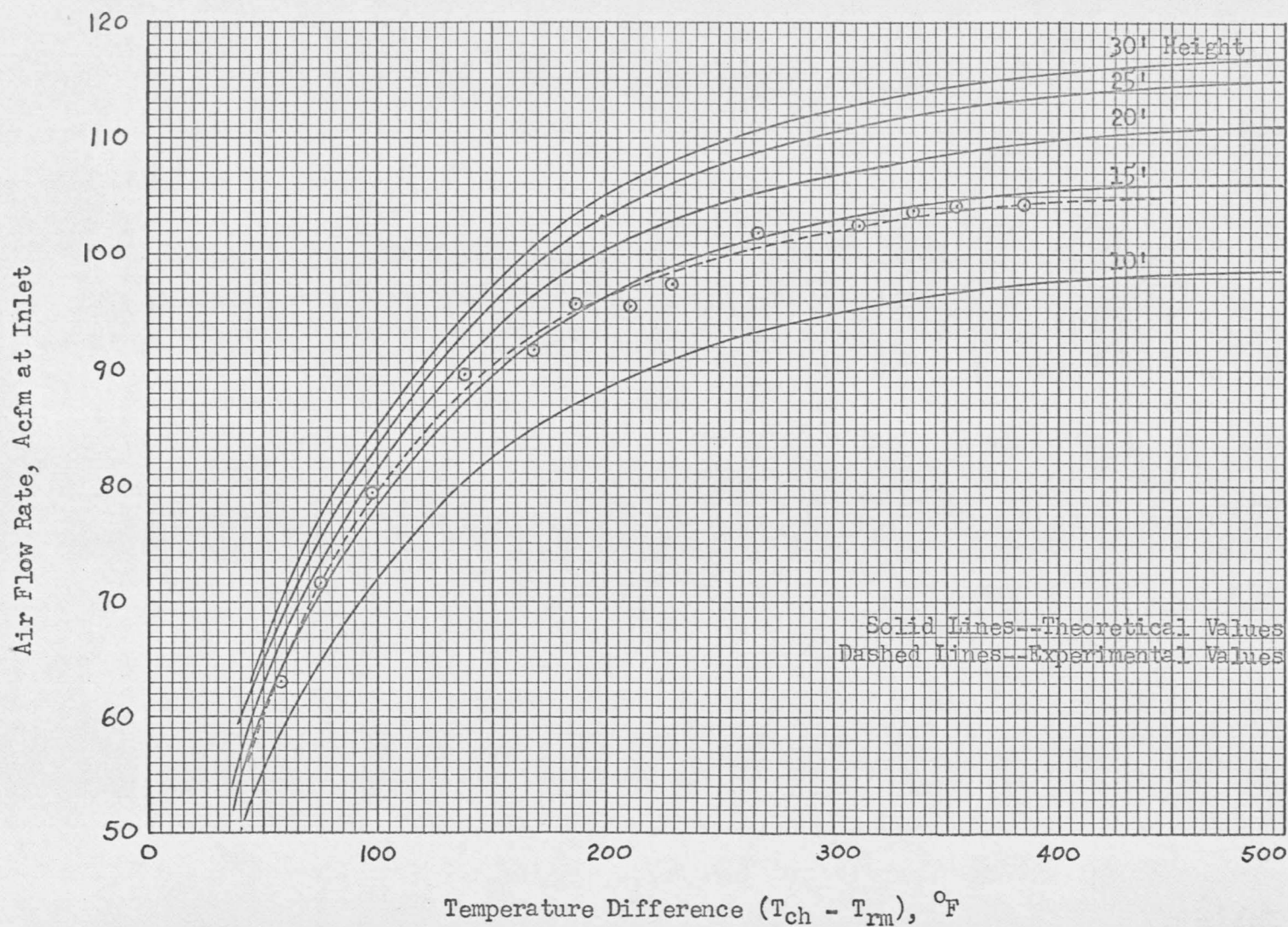


Figure 12. Air Flow vs. Temperature Difference for 6" Diameter Stack
Open Door Tests -- Straight Stack

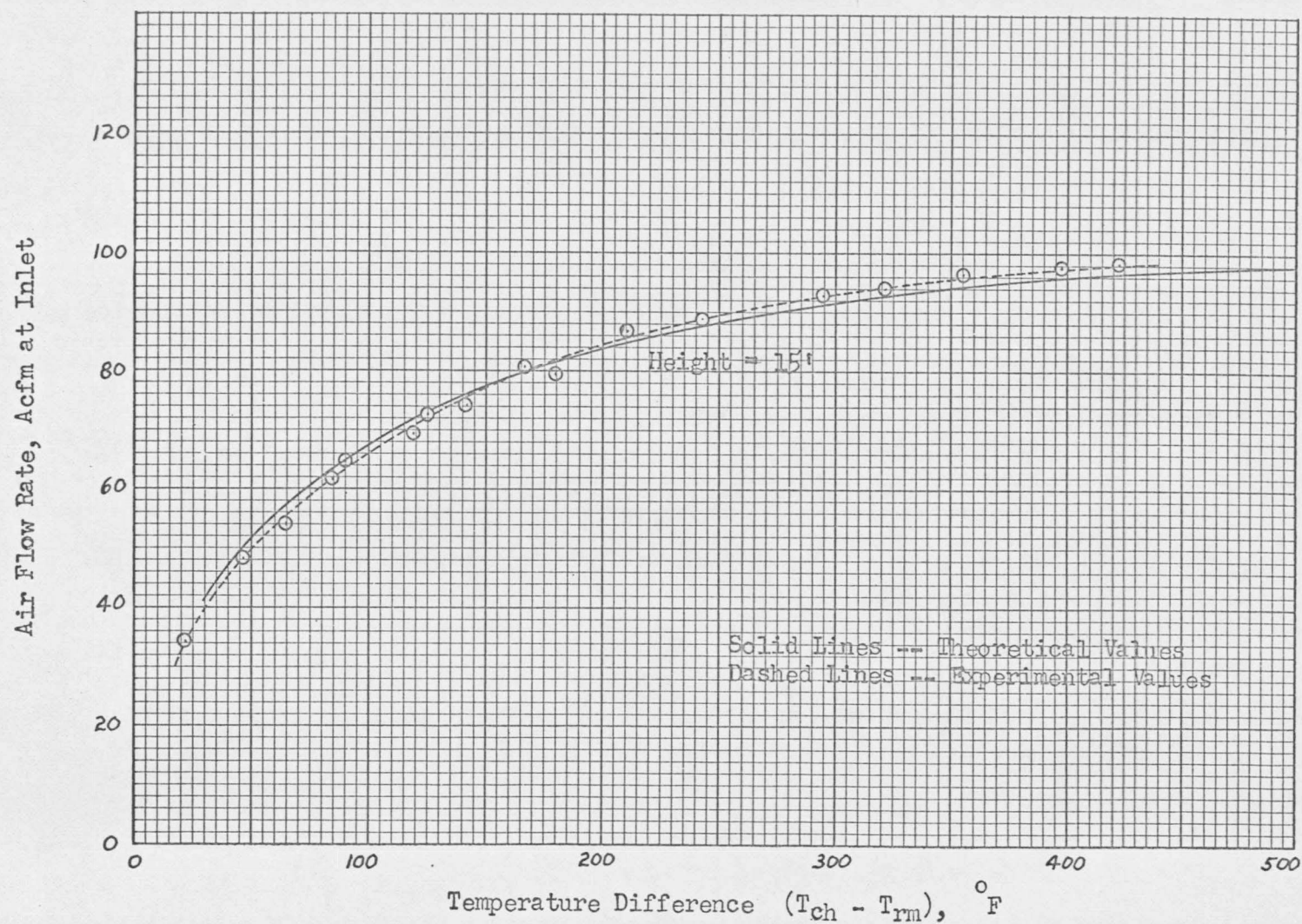


Figure 13. Air Flow vs. Temperature Difference for 6" Diameter Stack.
Closed Door Test -- Straight Stack

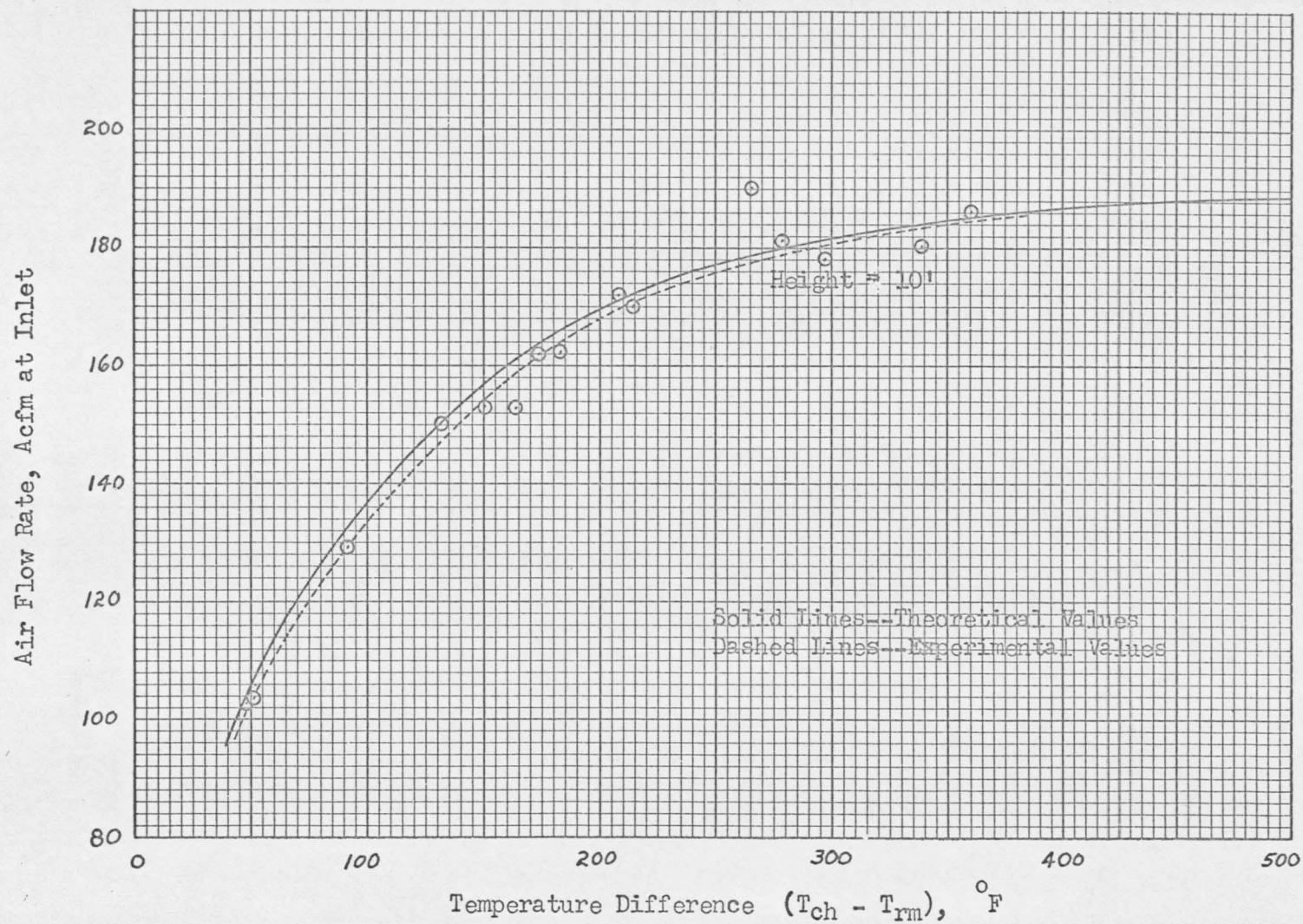


Figure 14. Air Flow vs. Temperature Difference for 8" Diameter Stack.
Open Door Test -- Straight Stack

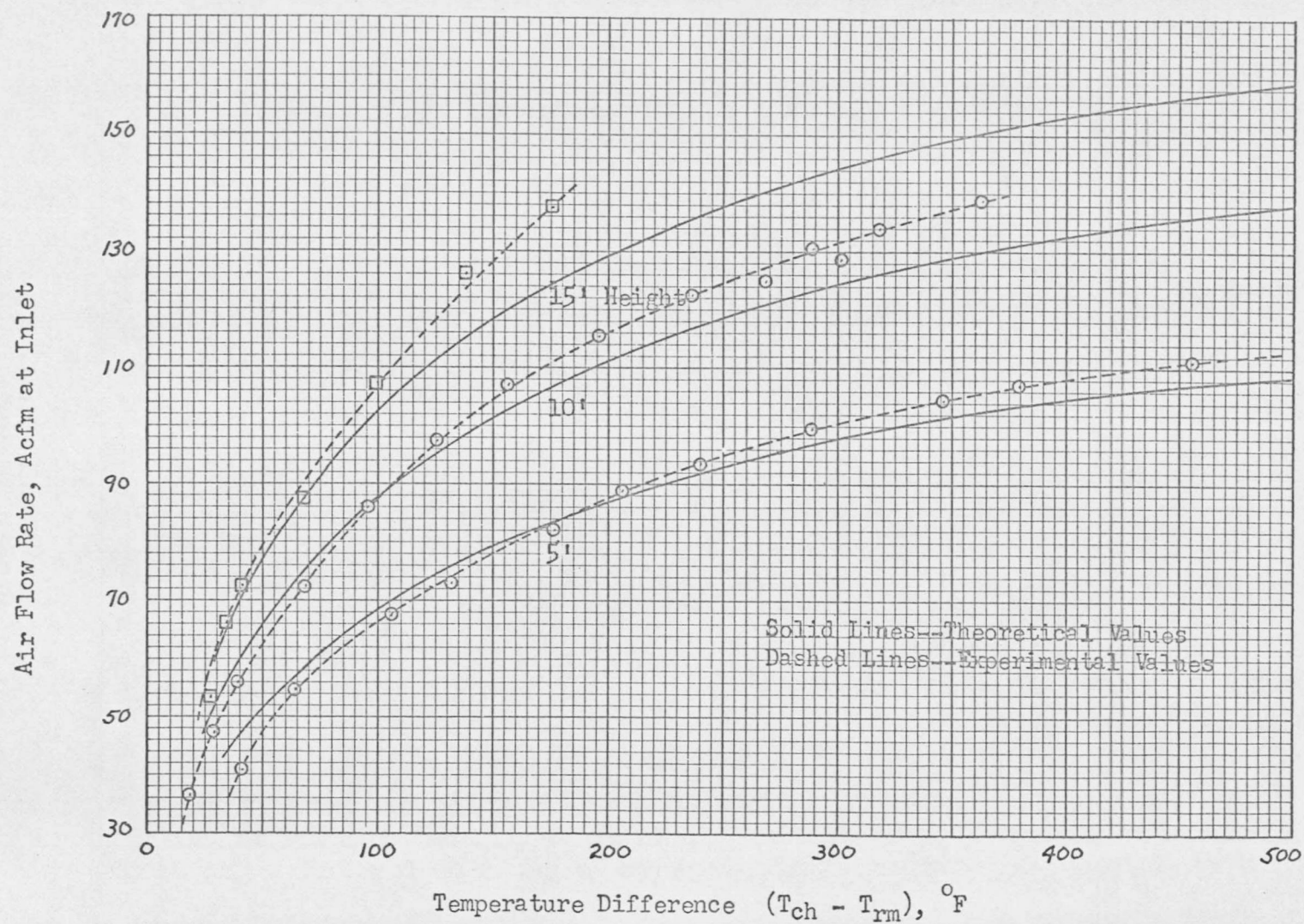


Figure 15. Air Flow vs. Temperature Difference for 8" Diameter Stack
Closed Door Tests -- Straight Stack

CHAPTER 11

VERTICAL STACK PLUS TWO - 90° ELBOWS - EXPERIMENTAL PROCEDURE

Since in many cases it is not feasible, from a construction stand point, to use a straight vertical stack, it was decided to test some stacks containing two - 90° elbows. The addition of the two elbows will permit a more flexible design.

The addition of the two elbows to the straight vertical stack should result, from the theoretical consideration, in an added restriction to the flow. The equations applicable to this design are the same as for the case of the straight vertical stack, the only difference appearing on the L term. The term L, length of stack, has a higher value in this case since it is no longer equal to the vertical height, but to the equivalent length of the system.

To calculate the value of equivalent length added to the system by each of the elbows, testing was done on the stack containing the two - 90° elbows. The testing was done for both cases, with and without inlet restriction. Using the experimental values, the equations in Chapter 10 were solved for the value of L. The value obtained for the equivalent length of each of the elbows was 4.2 feet. This value is approximately one half of the value given for 6 and 8 inch diameter elbows in the Heating, Ventilating, and Air Conditioning Guide. The difference in values can be explained since the values given by the Guide are for forced flow, while in the experiment, the flow was not as high and the elbows did not provide the full value of restriction.

After determining the equivalent length (L) for the stack containing the two 90° elbows the rest of the procedure was exactly the same

as the testing of the straight vertical stack. The results for this set of tests are given on Figures 16, 17, 18, and 19. If we compare the values of the air flowing in, obtained with this stack, to the values obtained with the straight vertical stack it can be seen that for the same value of temperature difference, the air inflow into the testing room is greater for the straight vertical stack than for the stack containing the two elbows. These results are in agreement with the theoretical considerations.

From the above discussion it can be deduced that the addition of the elbows to the original configuration, a straight vertical stack, results in a reduction in the flow obtainable from the stack. In an actual design the feasibility of having a straight vertical stack should be considered first, if the allowable space does not permit the installation of a straight vertical stack then a design having the least possible elbows or fittings is recommended.

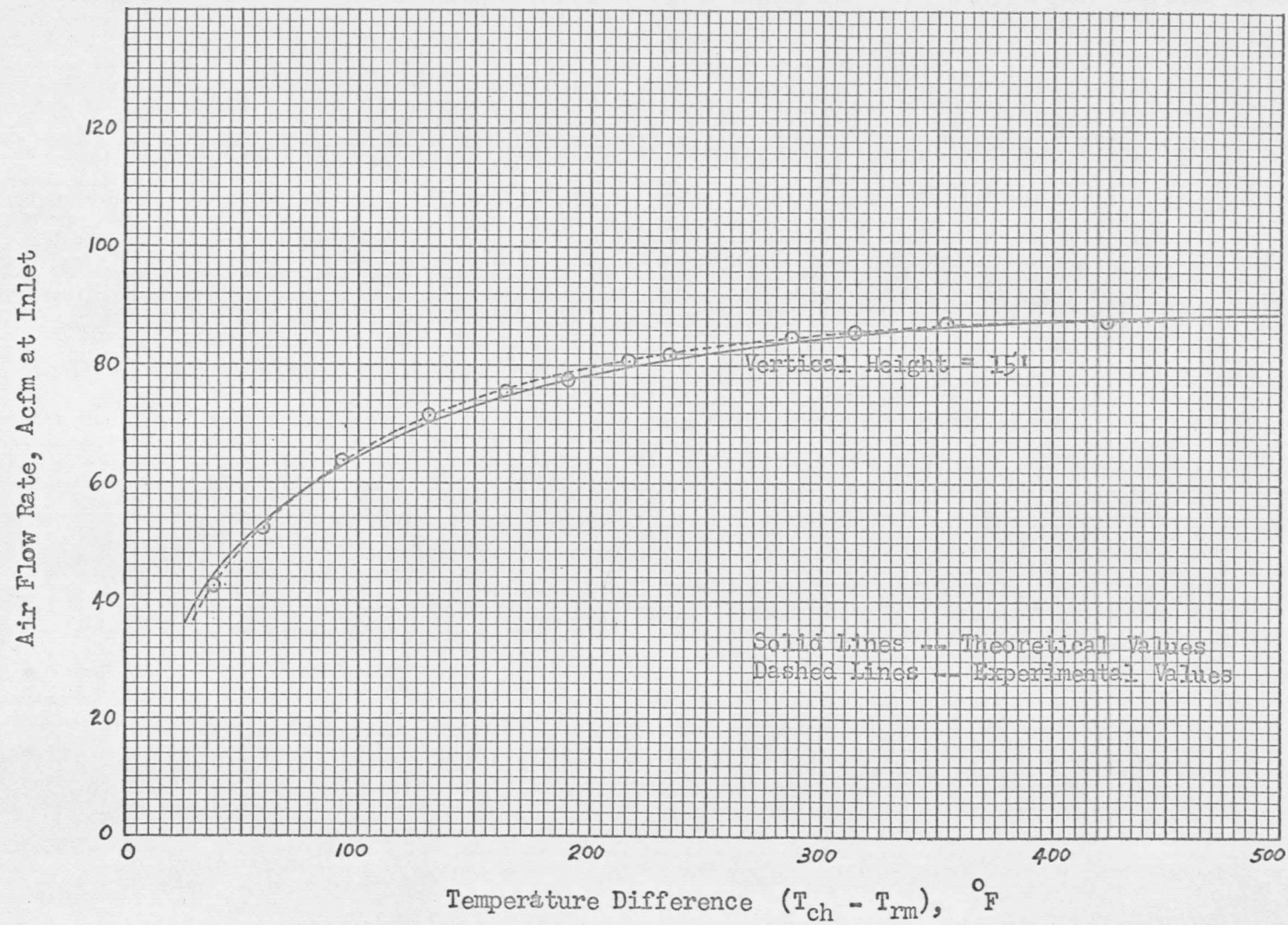


Figure 16. Air Flow vs. Temperature Difference for 6" Diameter Stack.
Open Door Test -- Elbowed Stack

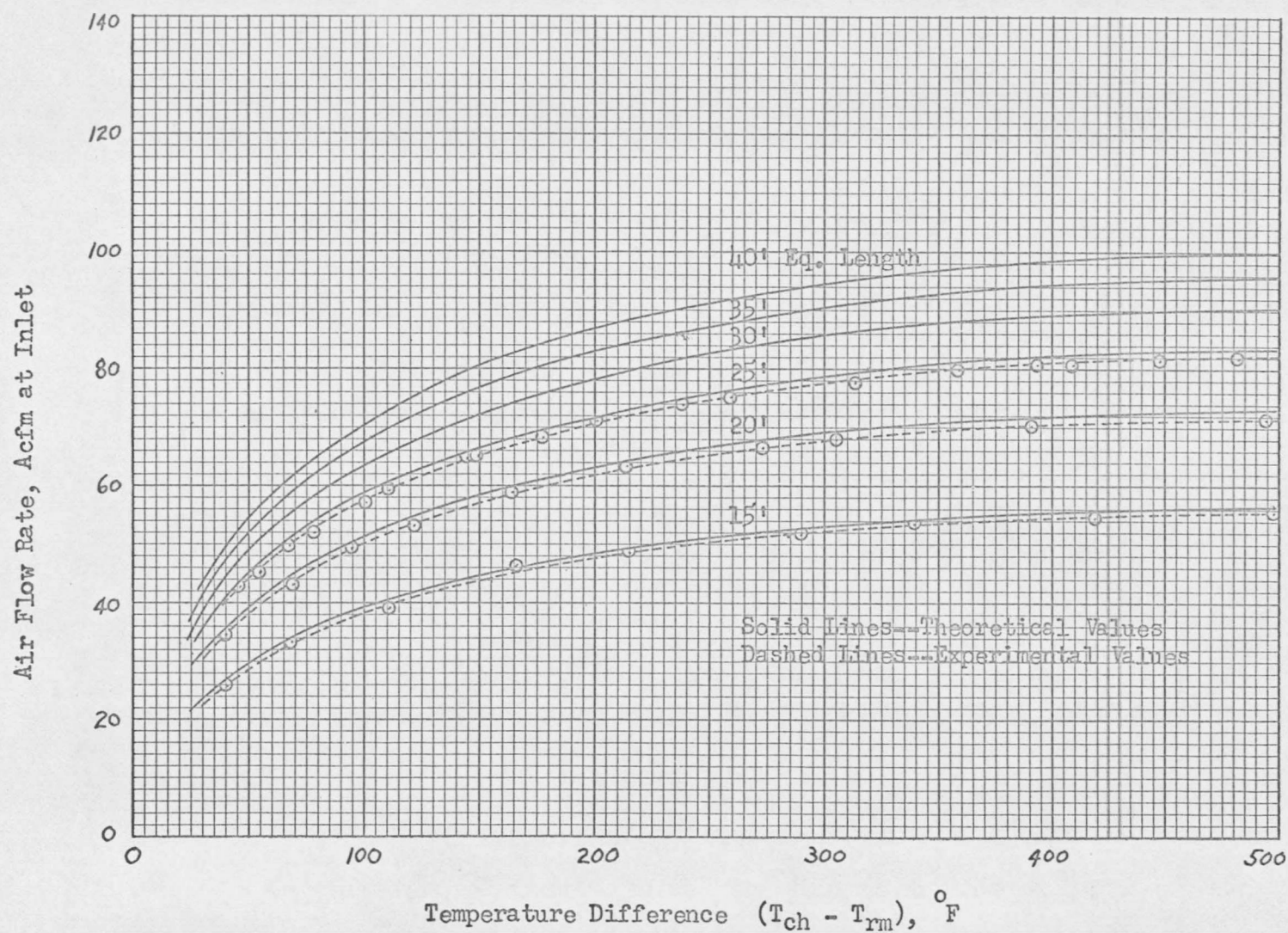


Figure 17 . Air Flow vs. Temperature Difference for 6" Diameter Stack
Closed Door Tests -- Elbowed Stack

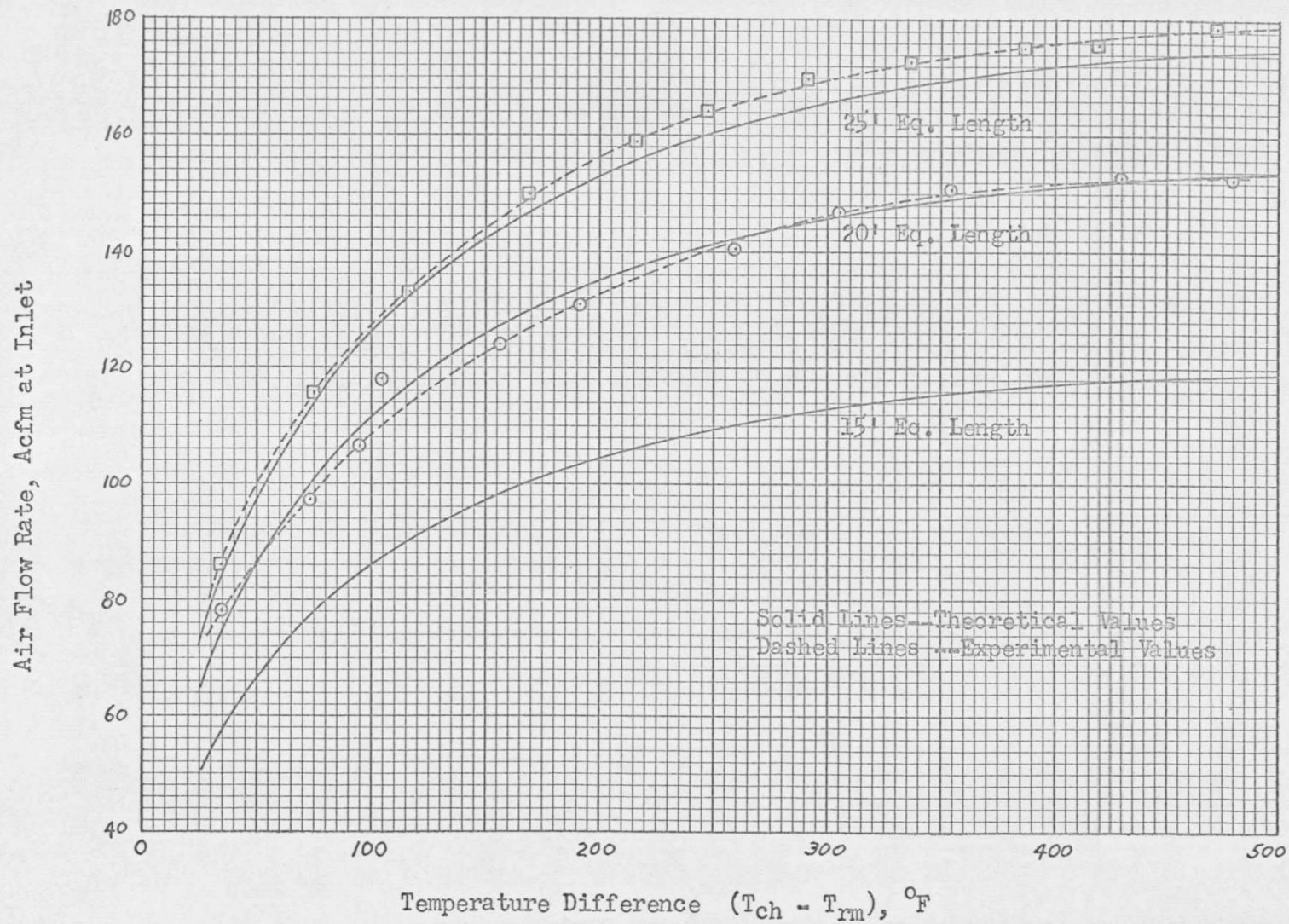


Figure 18. Air Flow vs. Temperature Difference for 8" Diameter Stack
Open Door Tests -- Elbowed Stack

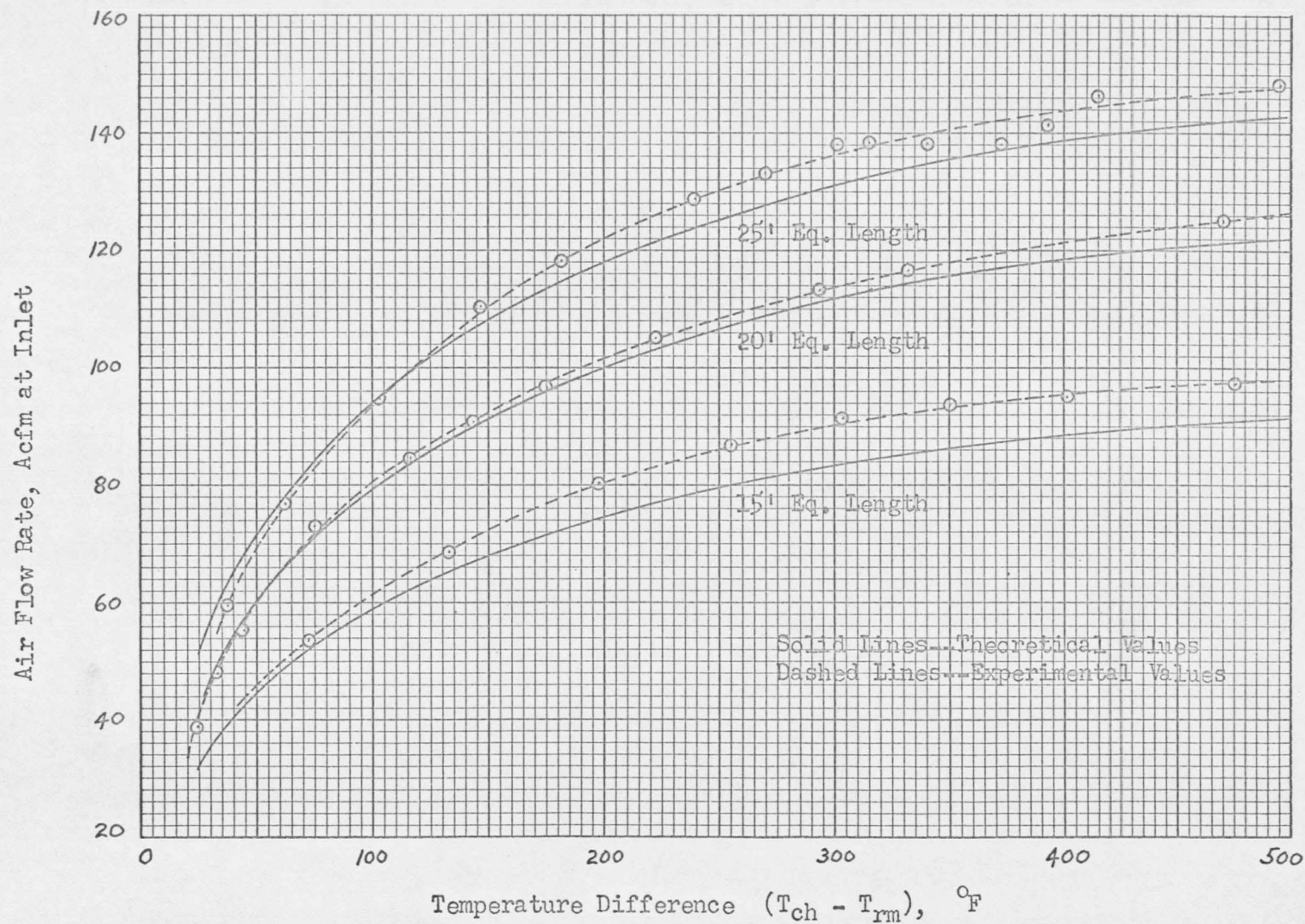


Figure 19 . Air Flow vs. Temperature Difference for 8" Diameter Stack
Closed Door Tests -- Elbowed Stack

CHAPTER 12

TESTING OF STACK CAPS

Because in practice a stack will need some type of cap in order to prevent rain or other undesired substances from falling into it, the influence of stack caps on the value of obtainable air flow from the stack was investigated.

The effect of adding a cap is to increase the equivalent length of the system, this in turn represents an added restriction to the system and should therefore result in lower values of available draft. Testing was conducted on two types of caps, Artis and Belmont. The Artis cap is of such configuration that it permits the air to emerge horizontally at two sides only, with the Belmont cap air emerges from all sides in a downward direction, i.e., air flowing up the stack must reverse directions in order to leave the cap. The experimental procedure followed was to measure the quantity of draft induced by the stack with and without the cap on, see Figure 20. From these values the equivalent length added by the caps was obtained.

The results showed that the Artis cap did not add a noticeable restriction to the system. On the other hand the effect of the Belmont cap was to add 5.5 equivalent feet to the system and thus reduce the amount of available draft. It should be noted that the experiment was conducted inside a building, zero air velocity. Because the caps are designed in such a manner that the effect of wind velocity will aid the air flow, instead of restricting it, it can be assumed that the performance of these caps under actual conditions will be the same or more favorable than in the experimental procedure.

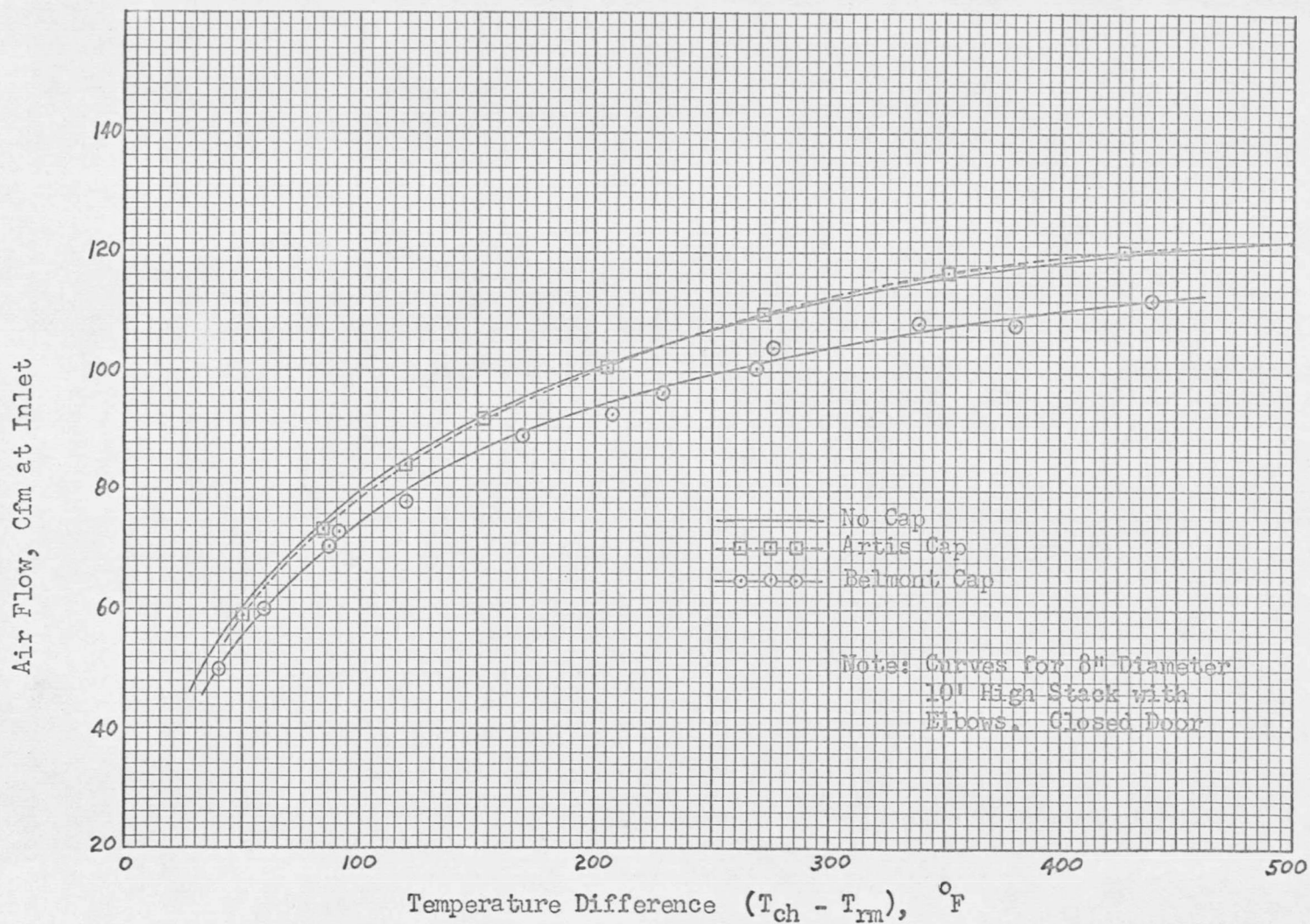


Figure 20. Effect of Ventilator Caps on Air Flow

CHAPTER 13

RESULTS

In the previous chapters the equipment, methods of procedure and the correlation of experimental and theoretical values have been discussed. The purpose of the previous discussion was to provide the reader with the basic theory of stack design and to establish the validity of the empirical equation developed in the laboratory.

In this chapter the end results of the study are presented in the form of graphs. The values used in plotting these graphs were obtained from the computer program of the empirical equation. The graphs are plotted for standard atmospheric conditions (29.9 inch Hg and 70°F) and for straight vertical stacks. The effect of small variations in barometric pressure at inlet conditions is so negligible that correction factors do not need to be applied to these graphs to account for the change in barometric pressure. Figure 21 shows the effect on the air flow of varying the altitude from 0 to 5000 feet.

The effect of adding elbows, horizontal length and vent caps to the vertical stack can be accounted for by the use for the correction factors given in Figures 22, 23, 24, and 25. These factors are to be applied to the required cfm value before entering the graphs for straight stacks. These correction factors were obtained by dividing the air flow rate obtained with a straight stack by the rate obtained with a stack system containing added restrictions of 5, 9, 13, 17, and 21 equivalent feet. The air flow rate for both configurations were acquired from a computer program prepared to solve the empirical equation for the air flow rate at standard conditions.

In this paper air flow is plotted against temperature difference. Because temperature difference is not a practical parameter in actual stack design, it was desired to make a table which relates the temperature difference to the heating system necessary to obtain it. Table 1 gives such a relation.

A more complete explanation of the design curves, tables, and method of design is given in the following example.

Sample Design Problem

It is desired to design a ventilation system which will provide a total of 100 cfm of fresh air. Assume that the available space is such that it will conveniently accommodate an 8 in. diameter, 15 ft high stack with two 90° elbows and 3 ft of horizontal run. A Belmont cap is desired at stack outlet.

Step (1) Calculate equivalent length of added restriction. From previous discussion it is known that:

1 90° elbow = 4.2 eq. ft

1 Belmont cap = 5.5 eq. ft

3 ft horizontal run = 3.0 eq. ft

Therefore, the total equivalent length of the added restrictions

= $2 \times 4.2 + 1 \times 5.5 + 3 = 16.9$ eq. ft added restriction.

Step (2) Figure 24 which gives the correction factor to be applied to an 8 in. diameter straight stack to account for the equivalent length of added restrictions. From this graph at an equivalent length of 16.9 ft and a vertical height of 15 ft, read a correction factor of 1.225.

Step (3) Multiply the desired air flow by this correction factor.

$$100 \times 1.225 = 122.5 \text{ cfm.}$$

Step (4) Enter Figure 28 which gives the desired temperature difference to obtain 122.5 cfm with an 8 in. straight stack of 15 ft vertical height. At 122.5 cfm we read a temperature difference of 57°F. (This means that if ambient air is at 80°F, the mean stack temperature must be $57 + 80$ or 137°F.)

Step (5) Table 1 indicates that a heater with an output of approximately 8,500 Btu/hr will provide the required heat output.

When using this design procedure the designer should be aware that more than one set of conditions will provide the desired ventilation rate. A through investigation of several of the possible sets of conditons should result in the most economical design. In considering the most economical design, the cost of the stack-system, fuel, and type of heater for each of the possible systems should be compared.

Air Flow, Acfm at Inlet

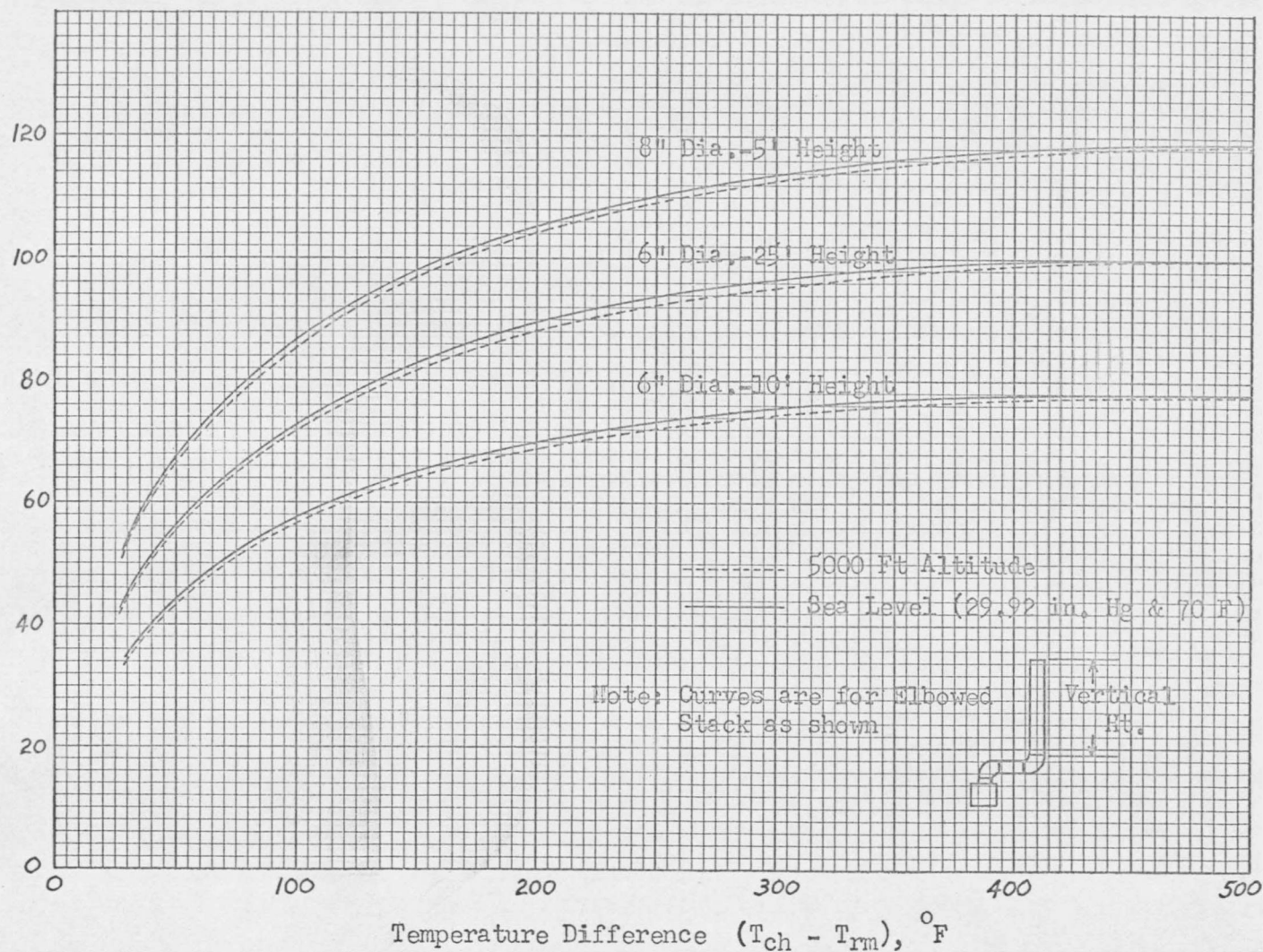


Figure 21. Effect of Altitude on Air Flow

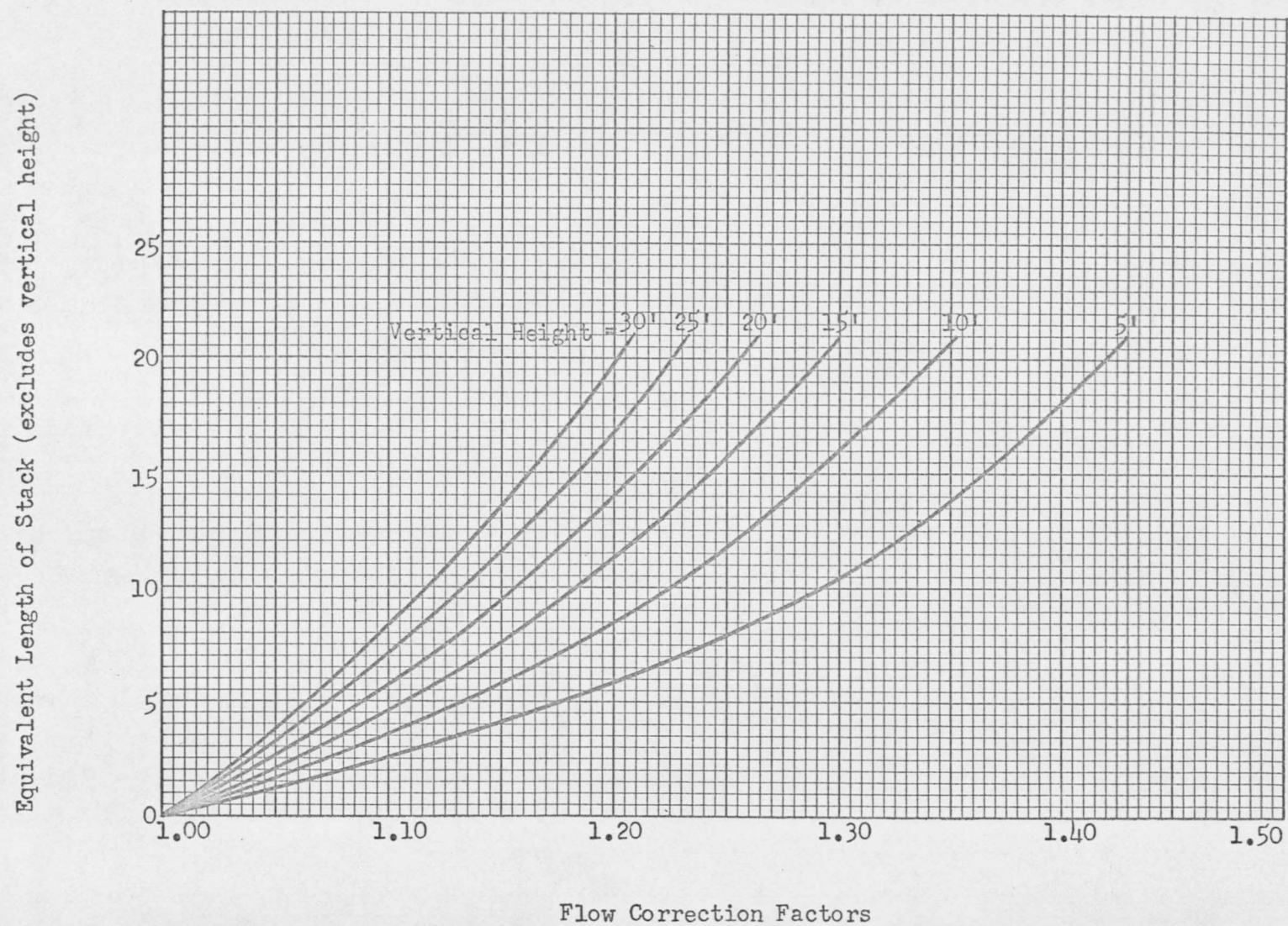


Figure 22. Flow Correction Factors vs Equivalent Length of Stack For 4" Dia. Stack

Equivalent Length of Stack (excludes vertical height)

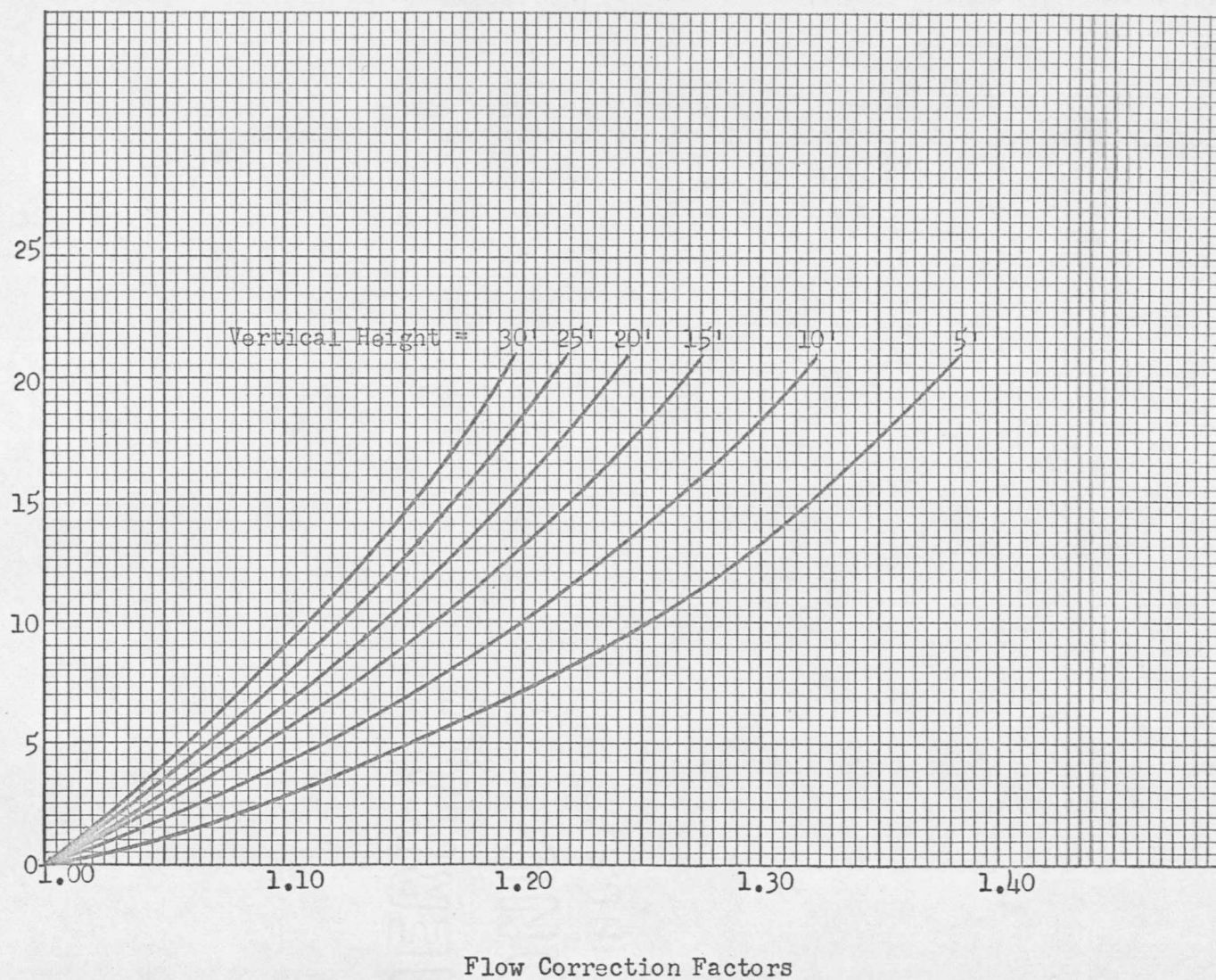


Figure 23. Flow Correction Factors vs Equivalent Length of Stack For 6" Dia. Stack

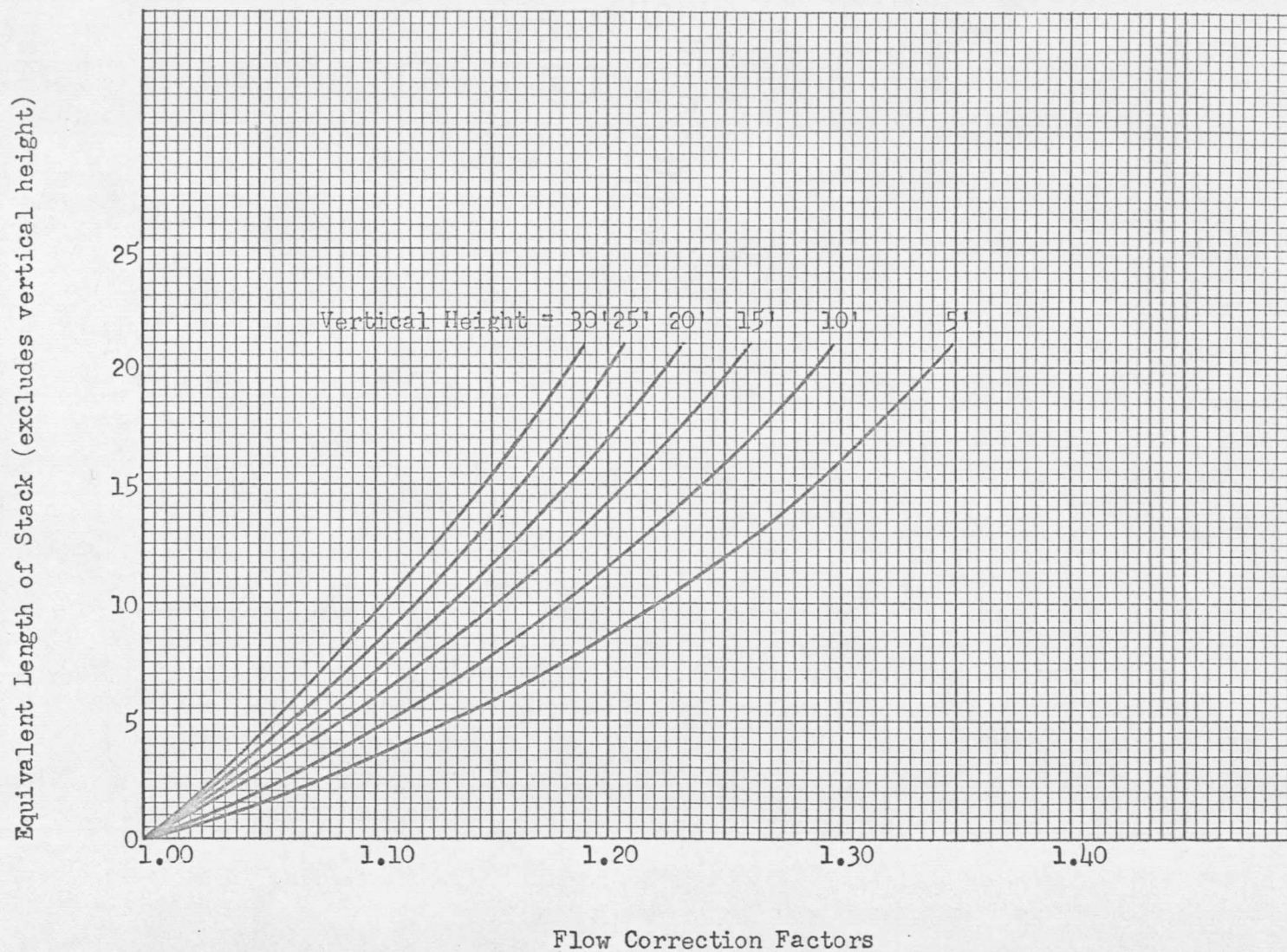


Figure 24 . Flow Correction Factors vs Equivalent Length of Stack For 8" Dia. Stack

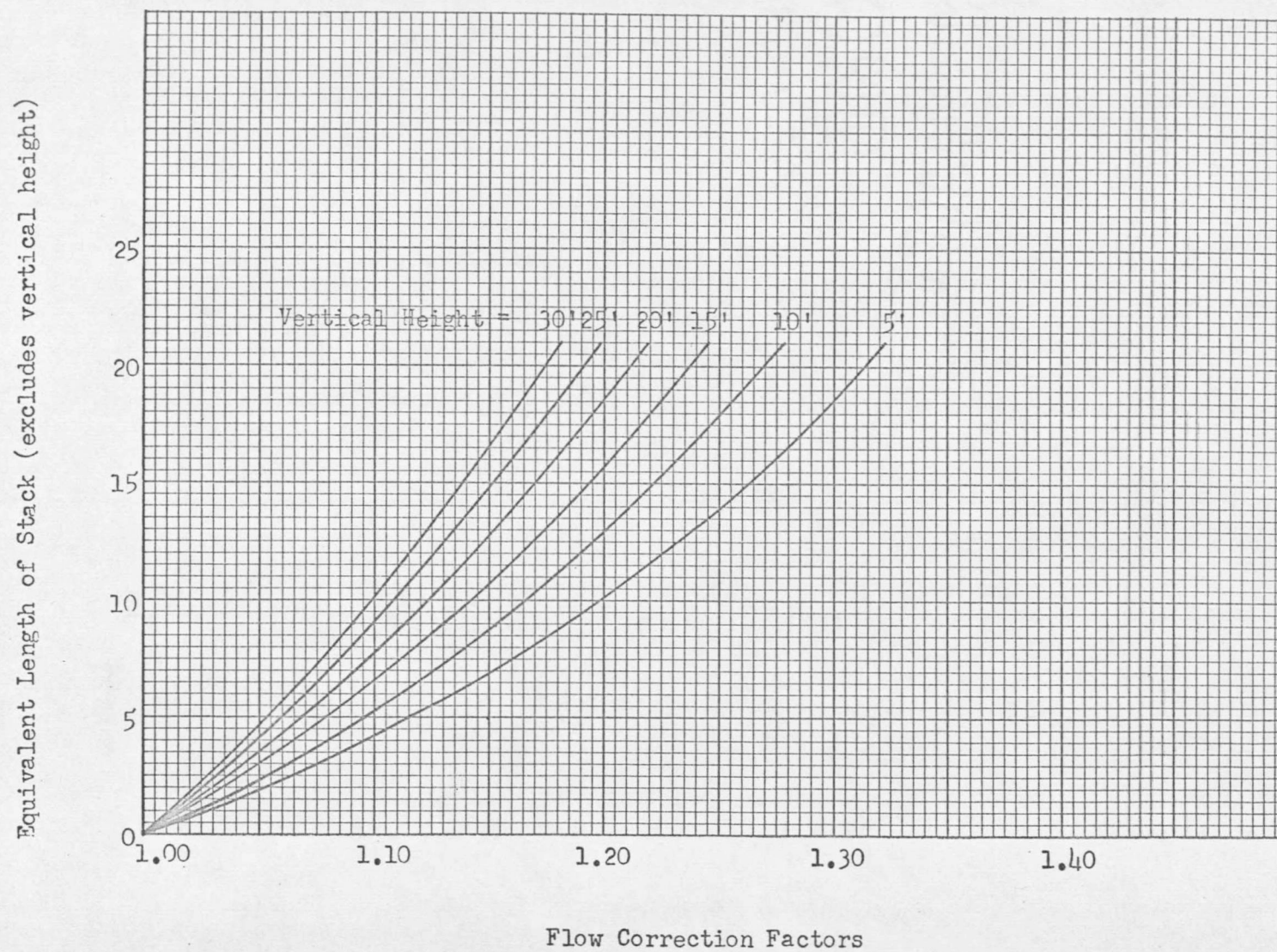


Figure 25. Flow Correction Factors vs Equivalent Length of Stack for 10" Dia. Stack

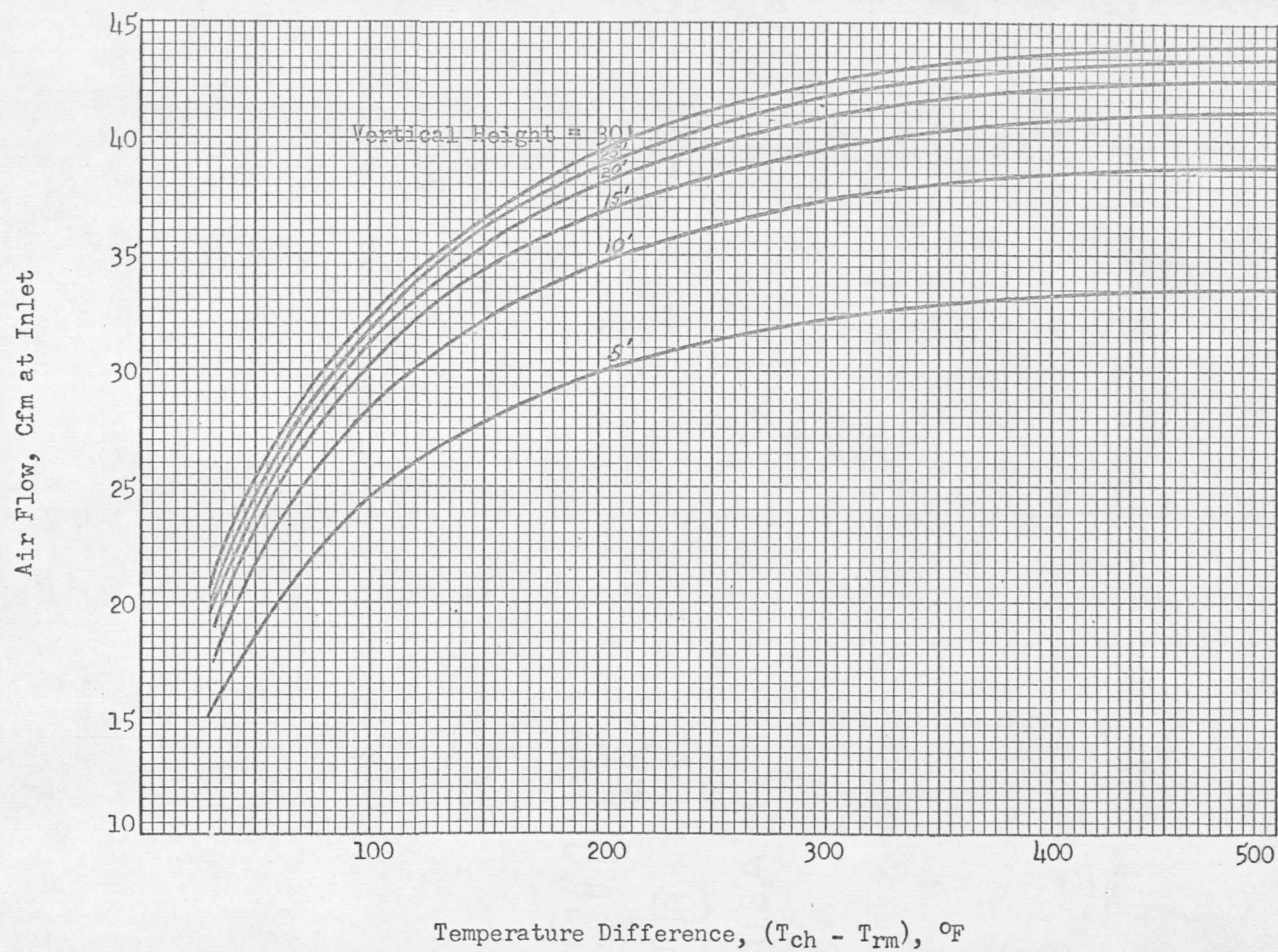


Figure 26 . Air Flow vs Temperature Difference for 4" Dia. Straight Stack

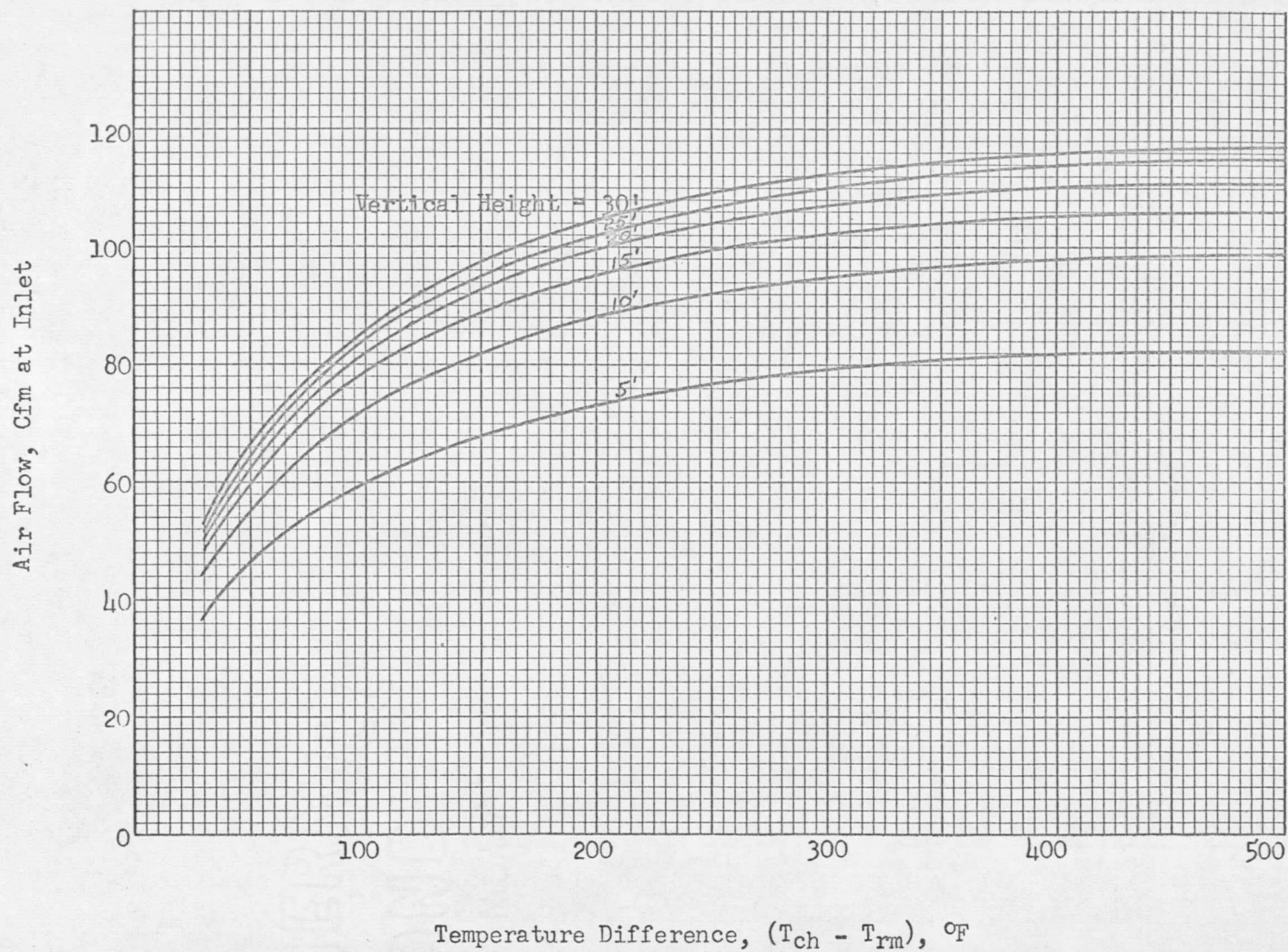


Figure 27 . Air Flow vs Temperature Difference for 6" Dia. Straight Stack

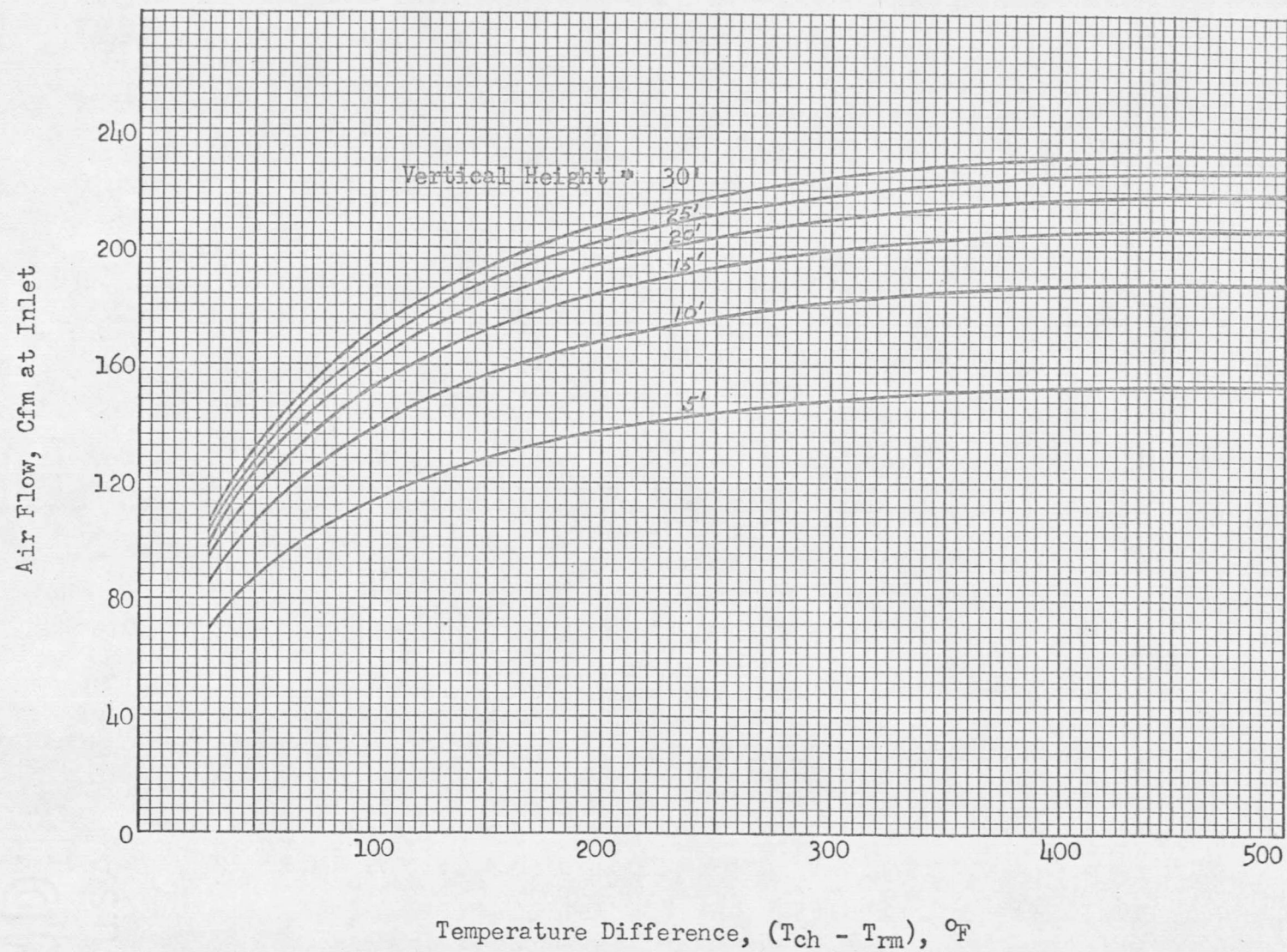


Figure 28. Air Flow vs Temperature Difference for 8" Dia. Straight Stack

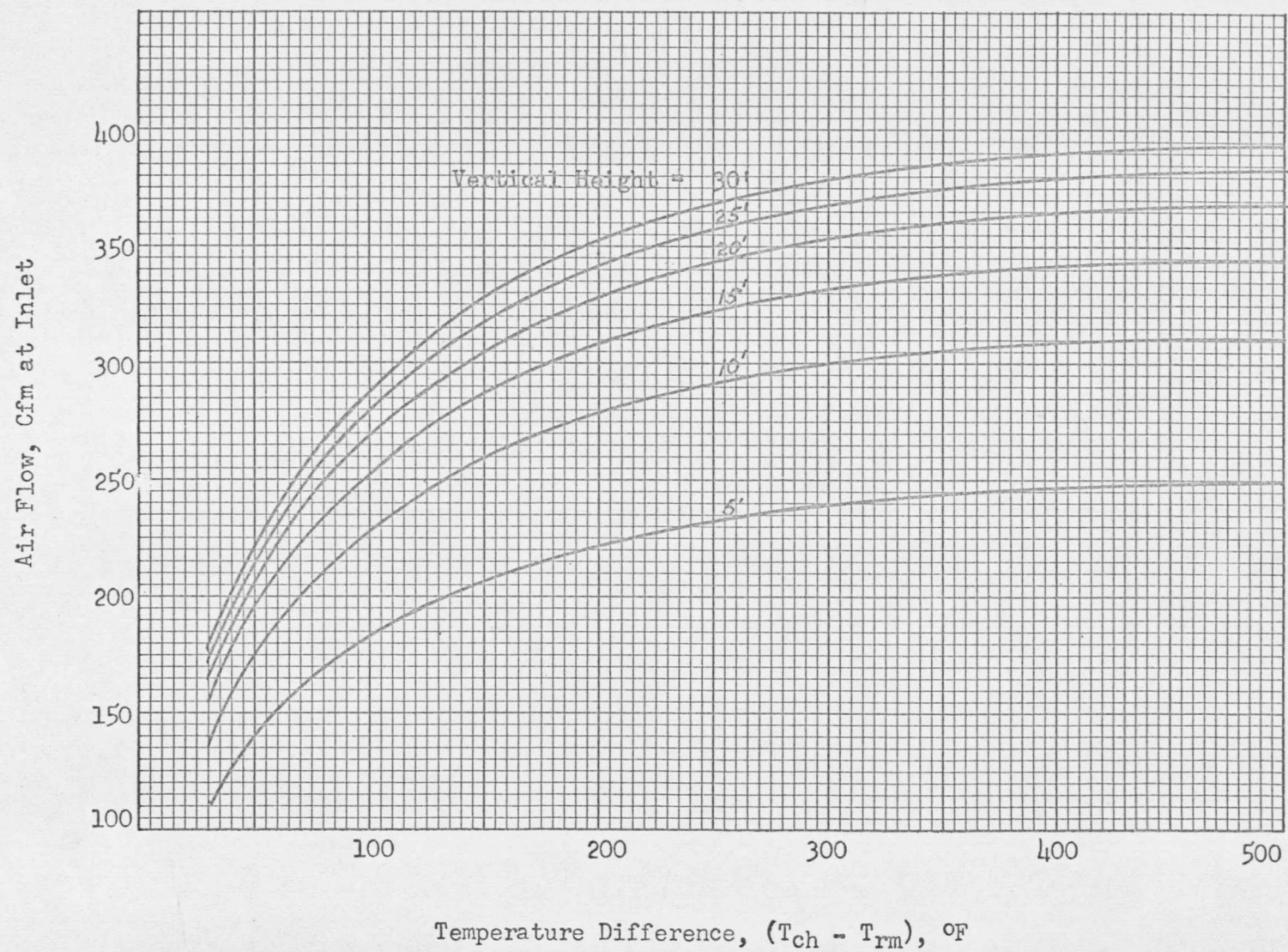


Figure 29. Air Flow vs Temperature Difference for 10" Dia. Straight Stack

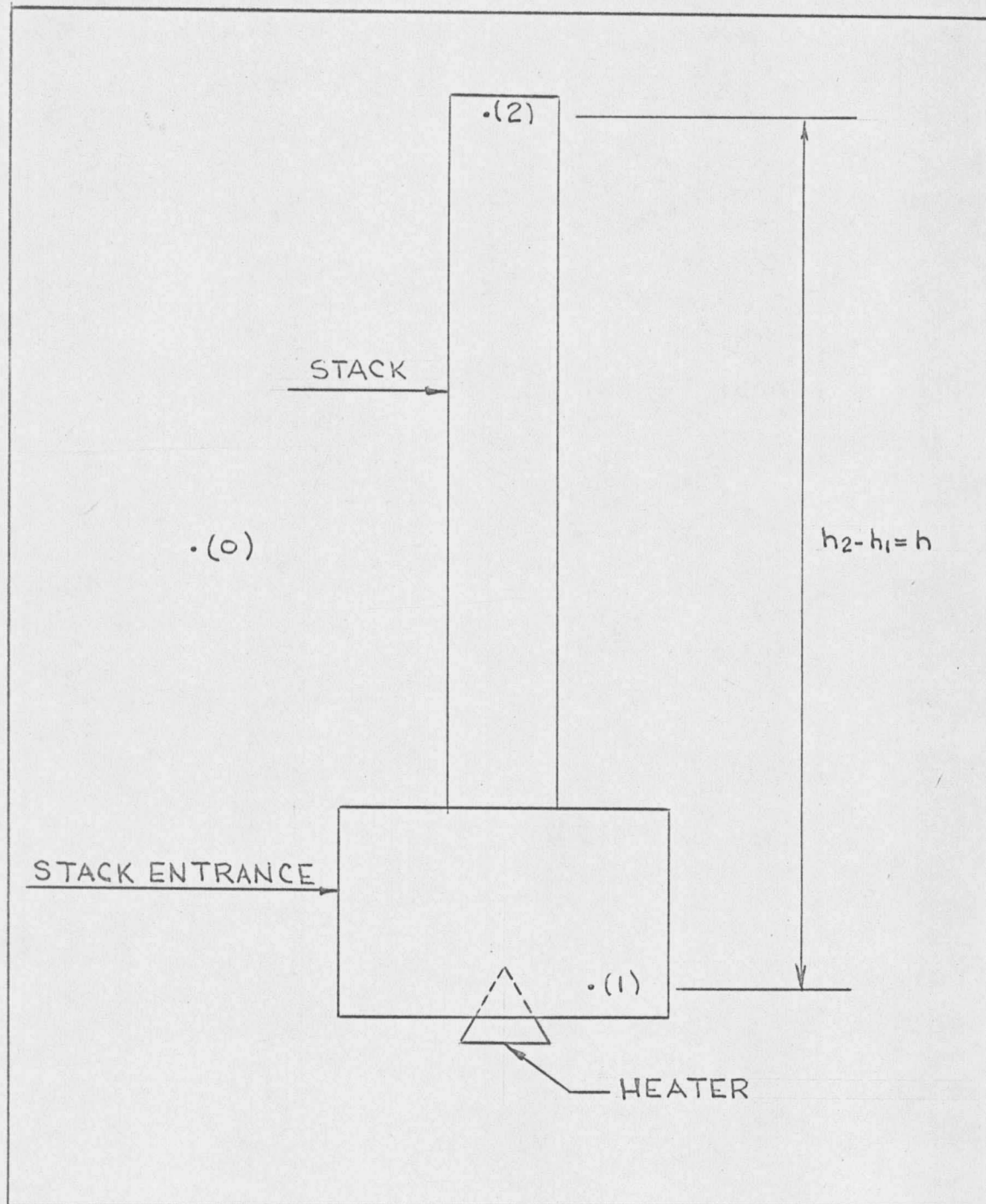


Figure 30 STACK CONFIGURATION

TABLE 1

TEMPERATURE DIFFERENCE VS. Btu/Hr
(KEROSENE FUEL)

Diameter Inches	Vertical Height Feet	Temperature Difference	Btu/Hr
6"	5'	35	2,600
		80	6,800
		120	10,600
		140	15,500
		170	16,600
6"	10'	35	3,200
		80	8,300
		125	16,000
		145	16,500
6"	15'	40	4,600
		80	9,200
		110	16,500
8"	5'	33	3,000
		40	4,000
		55	6,400
		100	12,700
		120	15,200
8"	10'	40	4,400
		70	8,500
		82	12,200
8"	15'	35	4,300
		51	6,600
		63	10,000
		71	11,000

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