



Ventilation by induced draft
by Owen Arthur Kubal

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of
MASTER OF SCIENCE in Mechanical Engineering
Montana State University
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Abstract:

Occupants of family-type fallout shelters require' fresh ventilation air at the minimum survival rate of 3 cfm per person. Because cost limitations exclude the use of auxiliary power plants (diesel or-gasoline engines) to operate ventilating fans or blowers, an inexpensive, simple, and effective method of supplying fresh air to home shelters is needed.

It is demonstrated that minimum air rates can be obtained in home shelters by inducing a draft in the exhaust stack by means, of, a flame from a kerosene burner which can simultaneously provide illumination.

The ventilation test procedure included inducing air to flow through the shelter, determining the actual cubic feet per minute of air flowing, measuring air temperatures at inlet, room, and stack, measuring the pressure drop or restriction to air flow at the shelter inlet, and finding the effects of various stack sizes and configurations upon air flow-rates.

Although not originally intended to be a large part,of the research work, a considerable amount of time and money was spent in finding a sensitive and reliable means of measuring low velocity air flows. This work led to the conclusion (incidental as far as shelter ventilation is concerned) that bead-type thermistors are not reliable air measuring devices when used in a temperature-compensating Wheatstone-bridge circuit. See appendix.

Ventilation of family-type shelters by the induced draft method is effective and reliable if the following conditions are observed: 1. Wind velocities around the stack outlet are kept to a minimum or a good ventilator stack cap is used.

2. Filters are not used at the shelter inlet.

3. The intake area of shelter is much larger than the cross-sectional area of the stack.

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Approved:

Harwood F. Mullikin
Head, Major Department

Cassius F. Whitehall
Chairman, Examining Committee

James D. Smith
Dean, Graduate Division

MONTANA STATE COLLEGE
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Abstract

Occupants of family-type fallout shelters require fresh ventilation air at the minimum survival rate of 3 cfm per person. Because cost limitations exclude the use of auxiliary power plants (diesel or gasoline engines) to operate ventilating fans or blowers, an inexpensive, simple, and effective method of supplying fresh air to home shelters is needed. It is demonstrated that minimum air rates can be obtained in home shelters by inducing a draft in the exhaust stack by means of a flame from a kerosene burner which can simultaneously provide illumination.

The ventilation test procedure included inducing air to flow through the shelter, determining the actual cubic feet per minute of air flowing, measuring air temperatures at inlet, room, and stack, measuring the pressure drop or restriction to air flow at the shelter inlet, and finding the effects of various stack sizes and configurations upon air flow rates.

Although not originally intended to be a large part of the research work, a considerable amount of time and money was spent in finding a sensitive and reliable means of measuring low velocity air flows. This work led to the conclusion (incidental as far as shelter ventilation is concerned) that bead-type thermistors are not reliable air measuring devices when used in a temperature-compensating Wheatstone-bridge circuit. See appendix.

Ventilation of family-type shelters by the induced draft method is effective and reliable if the following conditions are observed:

1. Wind velocities around the stack outlet are kept to a minimum or a good ventilator stack cap is used.
2. Filters are not used at the shelter inlet.
3. The intake area of shelter is much larger than the cross-sectional area of the stack.

INTRODUCTION

As more and more countries attain nuclear capability, we, as a nation must accept the fact that the possibility of the use of nuclear weapons is increasing. Our ability to fight limited wars without having to resort to the use of nuclear weapons has also been increased. Isolation and protection from attack, once provided by oceans and huge land masses, is no longer naturally available to the population. Protection from "modern" attack must evolve from a cooperative, well-planned program. Military strength provides a high level of retaliatory capability, but civilian preparedness is one of the best means of providing defense and protection.

Over the past four or five years the civil defense program has been reoriented around fallout shelter protection. In a nuclear attack, fallout shelters offer the best single non-military defense measure for the protection of the greatest number of people.

Attacks on the United States with weapons totaling from 2,000 to 15,000 megatons would require shelter stays of from two days to two weeks, depending upon the attack pattern and location. A nationwide system of fallout shelters would save tens of millions of lives in any of these attacks. At the lower levels, 25 to 40 million people would be saved by fallout shelters alone. With heavier attacks the lifesaving potential of fallout shelters increases and the proportion of survivors who would be alive because they were sheltered increases even more sharply. If we have an adequate civilian preparedness program and a war does occur, we could conceivably save well over 100 million people by the use of shelters.

A fact proven physiologically is that the average healthy person can

survive two weeks without food, clothing, berthing, light, and with only a minimum of potable water and sanitary facilities. All of these items constitute some of the desirable, though not necessarily mandatory, items for shelter habitability. It also is a physiologically proven fact that the average healthy person cannot survive for more than a few hours without fresh air. With perhaps the exception of adequate shielding, fresh air provided by some means of ventilation is the primary mandatory requirement for shelter survival.

Normal atmospheric air contains, by volume, 20.6 percent O_2 and 0.03 percent CO_2 . Air becomes objectionable to people when it contains greater than 3 percent CO_2 and less than 18 percent O_2 . Serious loss in vitality and ability will occur at CO_2 levels above 3 percent. Below 14 percent, lack of O_2 may cause sleepiness, headaches, inability to perform simple tasks, and eventual loss of consciousness. A ventilation rate of 3 cubic feet per minute (cfm) per person meets these requirements and will maintain the CO_2 level at about 0.6 percent and the O_2 level at about 19 percent.

The ventilation rate of 3 cfm of fresh air per person has been established as a minimum requirement by Civil Defense directors to be used in ventilation system design for shelters. It should be understood that in most shelters, this minimum will meet only brute survival conditions, not comfort conditions. A rate as high as 20 cfm per person may be required in some parts of the country for controlling the chemical content of the air plus providing means for heat, moisture, and odor removal which would approach comfort levels. Because our greatest concern during a nuclear attack would be survival and not comfort, the minimum rate of 3 cfm per

person is the most significant figure concerning shelter ventilation.

Fallout shelters are of two major types; community and family. The community-type shelter is defined as one which provides shelter space for 50 or more people. Because it cannot be assumed that commercial power will be available during shelter occupancy, community shelters must contain some form of auxiliary power for operating ventilation systems. This might be a small diesel or gasoline engine or a mechanical-type manually-operated fan.

Family-type shelters are those that provide shelter space for less than 50 people--usually between 5 and 15 persons. Auxiliary power in the form of a diesel engine for a shelter of this type would provide adequate ventilation, but it is impractical from the cost viewpoint. The average family cannot assume the cost of a standby powerplant of this nature. A manually-operated ventilator is also impractical because of the limited number of people available to operate it.

The solution to the problem of family shelter ventilation must be a means of supplying fresh air effectively yet inexpensively. It is proposed in this thesis that adequate ventilation air can be supplied to small shelters without the use of either electrical or mechanical power. The proposed method of ventilation requires inducing a draft by means of a flame in a chimney. The technique is simple, relatively inexpensive, safe, and easy to use.

CHAPTER 1

DESCRIPTION OF SHELTER

The shelter used for conducting ventilation tests is located on the Montana State College campus in Bozeman, Montana. The shelter was adapted from an underground tunnel located beneath the floor of the Mechanical Engineering Department's power laboratory. The original tunnel structure was rectangular in shape with internal dimensions of 37 ft by 6 ft. The ceiling is flat and approximately 6 ft high. Adjacent to the east end of the room is a 6 by 7 ft open entryway. Modifications were made to the original tunnel to the extent of adding a partition and door at a distance of 7 ft from the east end. This leaves the space used as a test shelter with 30 ft by 6 ft dimensions. The walls, floor, and ceiling of the shelter are all 8 in. thick concrete.

The partition, with 2 x 4 framing, consists of 6 mil polyethylene sheet fastened to the frame with duct tape. Caulking compound and duct tape were used to close all cracks between the partition framing and concrete. A 2-1/2 ft by 6 ft particle-board door was installed in the center of the partition and sealed with weather stripping.

An 8 in. diameter 3.8 ft long sheet metal duct was installed in the polyethylene partition about 2 ft from the floor and sealed with tape. This duct was used as the air intake to the shelter. To assure streamline flow in the intake, 1 in. dia. aluminum foil tubes, 12 in. long, were nested in the upstream end of the 8 in. duct.

At the west end of the shelter a 1 ft 9 in. square section of concrete was removed from the ceiling through which the outlet stacks were installed. Figure 1 on page 3 is a line sketch of the shelter test facility showing

intake duct and stack locations.

Minimums established by the Civil Defense other than the 3 cfm ventilation rate include 60 to 65 cu ft of space per person and 10 sq ft of floor space per person. The test shelter, with a floor area of 180 sq ft and a 6 ft ceiling, provides 60 cu ft of space for each of 18 people. Based on the minimum air rate, at least 54 cfm of outside air is required.

With a rated capacity of 18 people, the test shelter is in the category of family-type shelters. It also represents an ideal family basement shelter in other ways. The stack is completely contained within the power laboratory which has a ceiling approximately 30 ft high. Intake air is drawn at the floor level of and exhausted near the ceiling of the laboratory. Skylight windows in the laboratory ceiling may be opened to permit the hot stack gases to escape into the atmosphere.

Homes with basement shelters can utilize this same principle. The stack can be partially enclosed in stud spaces or run upward through a hallway and can exhaust directly to the atmosphere or to a ventilated attic. If the stack exhausts directly to the atmosphere, a ventilator stack cap should be used, not only to utilize the "suction" forces of prevailing winds to aid ventilation, but also to keep fallout radiation particles and other debris from entering the shelter. If the stack exhausts to an attic containing louvers or other vent openings, a ventilator cap is not needed at the stack outlet because the roof will keep out fallout particles, rain, etc., and will limit wind effects. Intake air can be drawn from the main part of the house. Outside air will enter the house in the normal infiltration method. B. H. Jennings in Heating and Air Conditioning

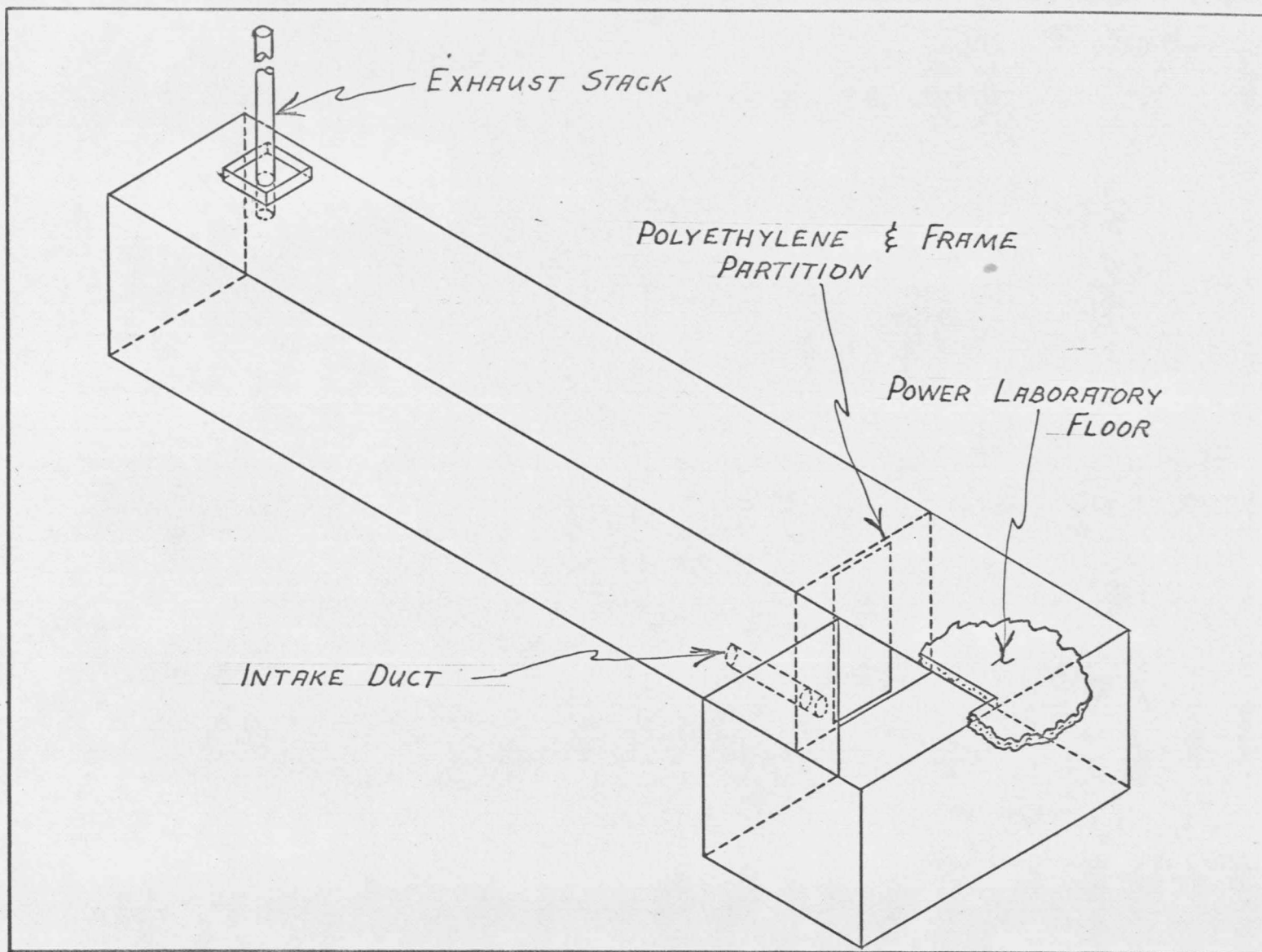


FIGURE 1. FALLOUT SHELTER TEST FACILITY

indicates that for the average residence the probable air change rate is about 1-1/2 air changes per hour by natural infiltration. If the average residence is considered to have a floor area of 1200 sq ft and an 8 ft high ceiling (9600 cu ft of space), this means that normal air infiltration is about 240 cfm. This is more than enough intake air for most basement shelters. Used as the intake air "plenum", the house itself would act as the principal filter for keeping fallout particles from entering the shelter intake. Little additional filtering, if any, would be needed at the intake. Thus a minimum restriction to air flow at the shelter inlet may be obtained.

To obtain the required 54 cfm of ventilation air through the test shelter, a safe, inexpensive, and easily operated combustion unit is needed to generate heat in the stack bottom. Liquefied petroleum gases such as propane and butane were excluded because of their high specific gravities. Being heavier than air, these fuels would collect in the shelter if leaks developed and would be inherently dangerous from an explosion standpoint. They also present a problem of storage because pressurized containers must be used. Again, this would be dangerous if punctures or leaks developed.

Natural gas would be an ideal fuel to use in the shelter, but it, like electricity, cannot be assumed to be available through normal commercial channels during shelter occupancy.

Gasoline would be unsafe in a shelter situation because of its high volatility and flammability. It would require special and perhaps expensive combustion equipment. When held in storage over relatively long periods of time, gasoline sometimes has a tendency to form gums. Their

presence in the fuel could lead to combustion equipment plugging.

At a first consideration it appears that coal, being inexpensive and relatively easy to burn, would be a good fuel. However, coal presents problems from the standpoint of storage space and ash removal. Coal burning devices would be complicated and require more draft than those for oil or gas because of the resistance of the fuel bed. For these reasons coal was deemed undesirable as a test fuel.

It was decided that kerosene (No. 1 diesel oil) would best serve the purpose. This fuel burns with a relatively clean flame, burners need not be complex or expensive, and storage in cans or other ordinary containers is relatively safe. It is also suitable as an illuminant when burned in a wick lamp or wick-type burner.

CHAPTER 2

AIR FLOW MEASURING EQUIPMENT - ANEMOMETRY

For use with ventilation tests a reliable and sensitive method for measuring low velocity air flows was needed. Because the pressure available to cause air flow is very small, the method of measuring the air that flows into the shelter has to be such that there is little or no pressure drop. This eliminates an orifice which, by its nature, requires a pressure drop.

Personnel from Electronics Research, Montana State College, were consulted concerning the possibility of measuring air flow in a duct using a thermistor probe. Since this group had previously conducted apparently successful research on thermistors, they agreed to design and build the necessary circuitry and thermistor probe.

The thermistor circuit used was a temperature-compensating Wheatstone-bridge circuit using transistors to maintain constant current. Several bead-type thermistors were tried in this circuit without success. Much time was spent trying to develop a reliable anemometer from the thermistors, but the problem of temperature compensation was not completely solved.

Details of the thermistor anemometry, including calibration procedures, can be found in the appendix.

Another unsuccessful attempt to measure air flow was made using a hot-wire anemometer. The instrument was neither sensitive enough nor accurate enough at low air velocities.

In search for an instrument or method to measure air flow through the shelter, Professor Drummond suggested that we could design a special thermocouple device consisting of two thermocouples wired in series set

one behind the other in the air flow stream. Between them should be a common resistor giving off a constant quantity of heat. The thermocouples sense the increase in air temperature caused by heating the resistor and produce a voltage output which depends on air velocity. A sketch of the simple anemometer is shown in Figure 2a on page 8. Figure 2b shows in part the thermocouple and resistor wiring diagrams. Assumptions concerning the design parameters of the meter are:

1. Flow rate range expected to be measured in one 8 in. intake duct is 10 to 150 cfm at actual conditions (70°F and 25 in. Hg.)
2. Anemometer will be installed in one of the 1 in. aluminum foil tubes and placed in the center of the intake duct.
3. Accuracy in measurement of air temperature difference at the highest anticipated flow is limited to 1°F using iron-constantan thermocouples.
4. A maximum input of 40 volts d.c. is available as supply voltage.
5. The specific heat of air throughout the expected flow range is constant at 0.24 Btu/lb °F.

Calculations for the required size of resistor are given below. The reader is referred to any standard text on thermodynamics or heat transfer if a more complete explanation of the basic equations is desired.

The weight of air per minute flowing through the 8 in. intake duct (w_8) is given by the perfect gas law as:

$$w_8 = \frac{P_B(\text{in. Hg}) \times V(\text{ft}^3/\text{min}) \times 0.491(\text{psi/in. Hg}) \times 144(\text{in}^2/\text{ft}^2)}{R(\text{ft-lb/lb } ^\circ\text{F abs}) \times T_a(^{\circ}\text{F abs})}$$

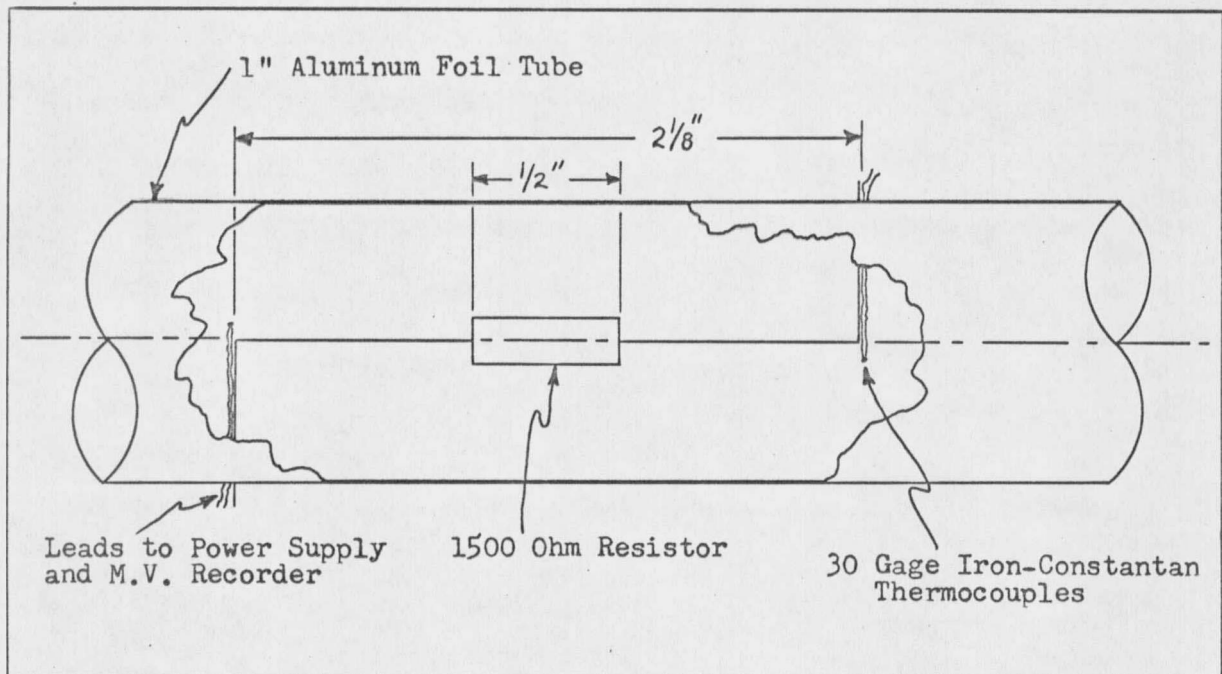


Figure 2a. Special Thermocouple Anemometer

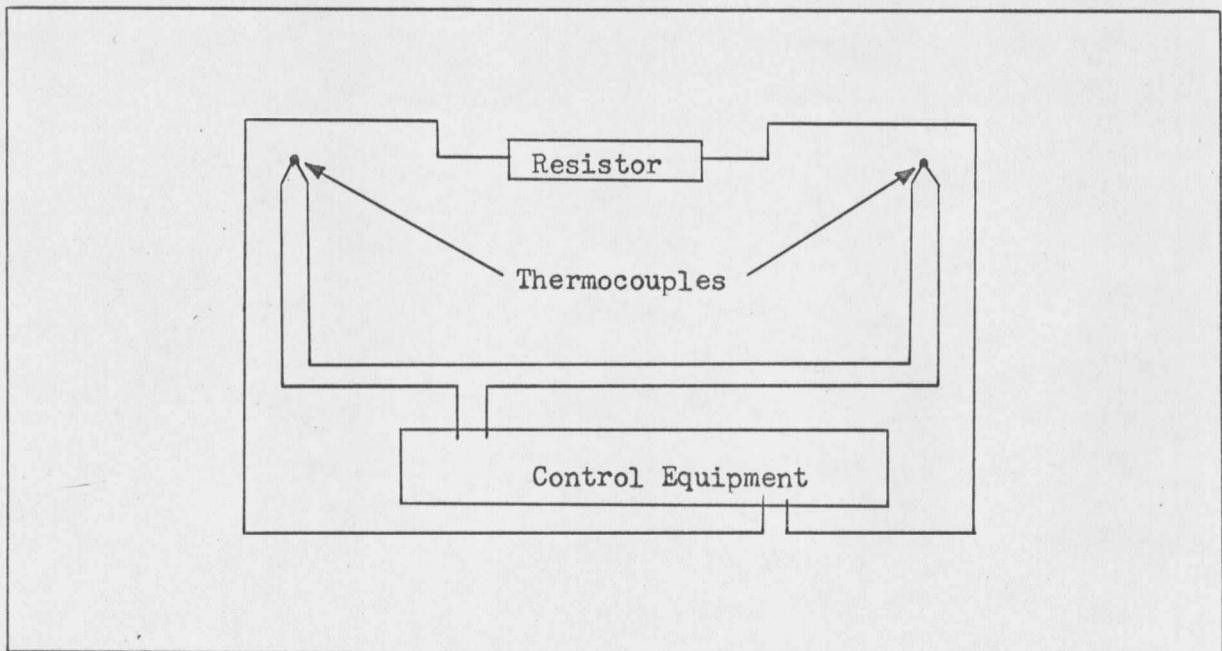


Figure 2b. Resistor & Thermocouple Wiring Diagram

$$= \frac{25(0.491)(144)V}{53.3(530)} = 0.0625 V$$

At the lowest flow (V) of 10 cfm,

$$(w_8)_l = 0.0625 \times 10 = 0.625 \text{ lb/min}$$

At the highest flow of 150 cfm,

$$(w_8)_h = 0.0625 \times 150 = 9.37 \text{ lb/min}$$

Area of the 1 in. foil tube,

$$A_1 = \pi d_1^2/4 = 0.785 \text{ in}^2$$

Area of the 8 in. duct,

$$A_8 = \pi d_8^2/4 = 50.2 \text{ in}^2$$

The weight flow of air in the 1 in. tube is given by multiplying the weight flow in the 8 in. duct by the area ratio. Hence:

$$(w_1)_l = (w_8)_l \times A_1/A_8 = 0.625 \times 0.785/50.2 = 0.00977 \text{ lb/min}$$

$$\text{and } (w_1)_h = (w_8)_h \times A_1/A_8 = 9.37 \times 0.785/50.2 = 0.1465 \text{ lb/min}$$

Heat from the resistor (Q, Btu/min) supplied to the air is given by the specific heat equation for constant pressure, $Q = w c_p \Delta T$. At the highest flow with a temperature difference (ΔT) of 1° F,

$$\begin{aligned} Q_h &= (w_1)_h c_p \Delta T = 0.1465 \times 0.24 \times 1 \\ &= 0.0351 \text{ Btu/min} \end{aligned}$$

If heat from the resistor is held constant at Q_h , then ΔT for any flow is given by

$$\Delta T = Q_h/w c_p = 0.0351/0.24 w = 0.1452/w.$$

Thus the maximum ΔT , at the lowest weight flow $(w_1)_l$, should be:

$$\Delta T_{\max} = 0.1452/0.00977 = 14.87^\circ \text{ F}$$

In order to hold constant the heat from the resistor at Q_h , the power (P) supplied to the resistor should be:

$$\begin{aligned} P &= 0.0351(\text{Btu/min}) \times 778(\text{ft-lb/Btu}) \times 1/33,000(\text{ft-lb/min hp}) \\ &\quad \times 746(\text{watts/hp}) \\ &= 0.617 \text{ watts.} \end{aligned}$$

Because $P = I^2R = vI$, the current (I) required to maintain the above power (assuming v is constant at 40 volts) is:

$$I = P/v = 0.617/40 = 0.0154 \text{ amps.}$$

The maximum resistance for the anemometer is:

$$R_{\max} = v/I = 40/0.0154 = 2595 \text{ ohms.}$$

The thermocouple anemometer was constructed according to the original ideas and assumptions except that the resistor used is one rated at 1 watt and 1500 ohms. This resistor, rather than one of 2595 ohms, was decided upon because of availability and its small physical size.

The one-inch foil tube with the meter installed in its center was placed in the center of the tubular grid at the end of the 8 inch intake duct. Appropriate leads were connected to the resistor and thermocouples. A vacuum tube voltmeter was used to read the generated voltage from the thermocouples. A preliminary check run was conducted with about 38 volts input to resistor. This voltage is used instead of 40 volts to keep the power near but less than one watt. The output of the instrument seemed very stable over the entire range of flow from 10 to 150 cfm, but the change in output voltage over this flow range was limited to about 0.25 millivolts. To extend the output range scale at the vacuum tube voltmeter (VTVM), a 180 K ohm resistor and a small adjustable resistor were added to the circuit. To obtain a permanent record of output, a 0-10 millivolt recorder was indirectly connected to the 0-1 volt output scale of the VTVM through a voltage divider (two arbitrarily chosen resistances) whose purpose is to again extend the scale range. Arrangement of the equipment is shown in Figure 3 on page 11. The 20.1 K ohm resistor in series with the

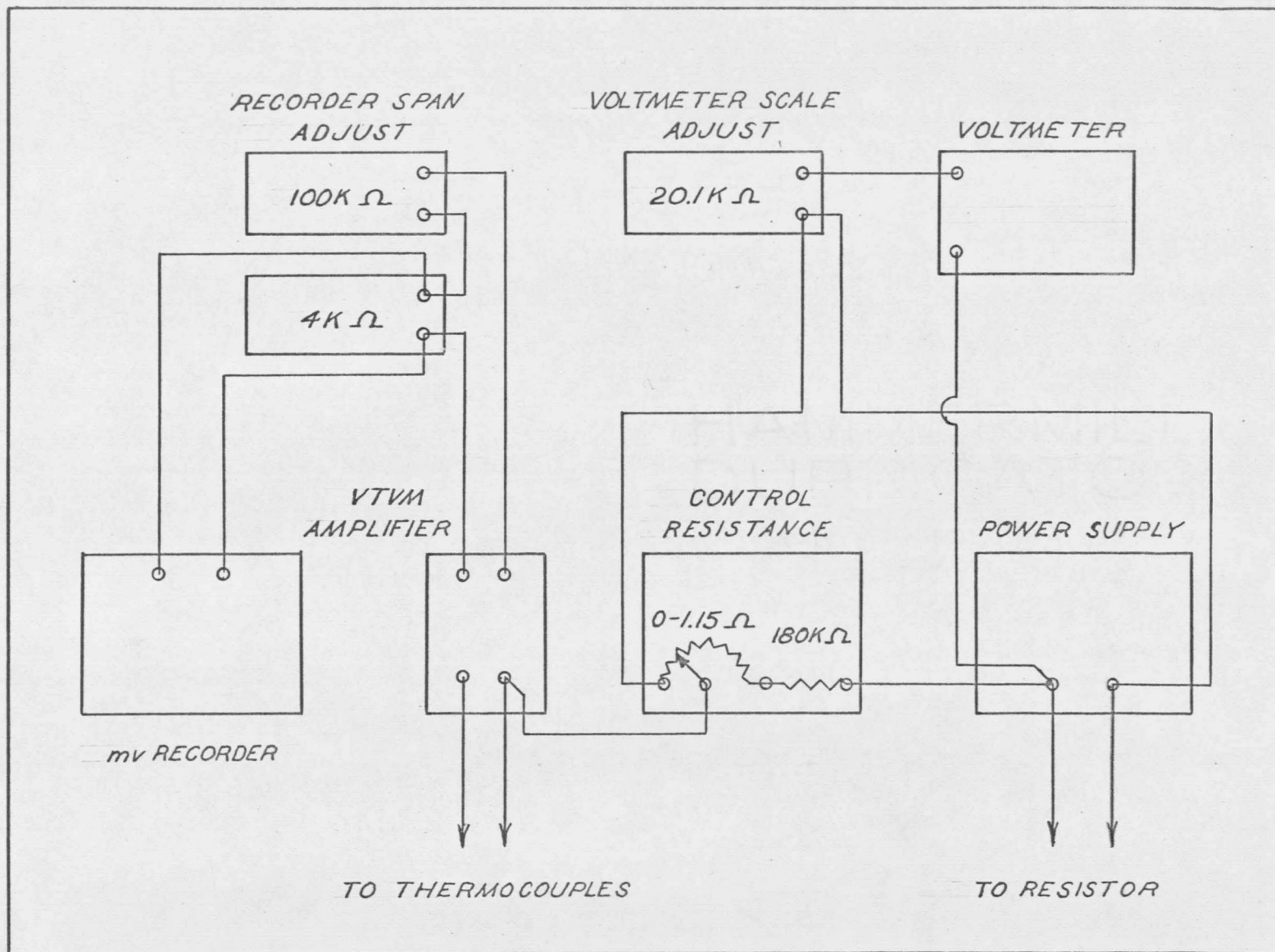


FIGURE 3. CONTROL EQUIPMENT FOR THERMOCOUPLE ANEMOMETER

voltmeter permits close control of input voltage to the anemometer resistor. The 0-1.15 ohm adjustable resistor is used to adjust the recorder scale to zero when the anemometer is in still air. This circuit, with divider resistances chosen by trial and error, allows nearly a .9 millivolt output range at the recorder over the 150 cfm flow range.

Calibration of the thermocouple anemometer indicates that the instrument provides a fairly reliable method of measuring air flows above about 40 acfm. See Figure 4 for the calibration curve. Between flows of 40 and 20 cfm, the millivolt output does not change enough to sufficiently describe a particular point on the circular portion of the curve.

Although the use of the thermocouple anemometer is somewhat limited at low air flows, it is one of the two methods finally used in the shelter ventilation tests. The author believes that flow measurements taken with this device above 40 acfm can be trusted.

The second method of air flow measurement finally adopted employs an all-mechanical propeller anemometer, sometimes referred to in this paper as the "windmill". The instrument consists of an eight-bladed fan, each blade being about 1-3/16 inches long. Motion of the fan shaft is transmitted to a set of indicating dials through a network of tiny gears. Figure 30 on page 70 shows the windmill, its position at the outlet end of the intake duct, and the clamping arrangement for its support.

A stop watch is used with the windmill when a reading is taken because the dials indicate lineal feet of air rather than air velocity. Windmill operation is somewhat maladroit because the dials must be correctly read and the readings recorded each time the watch is started and stopped.

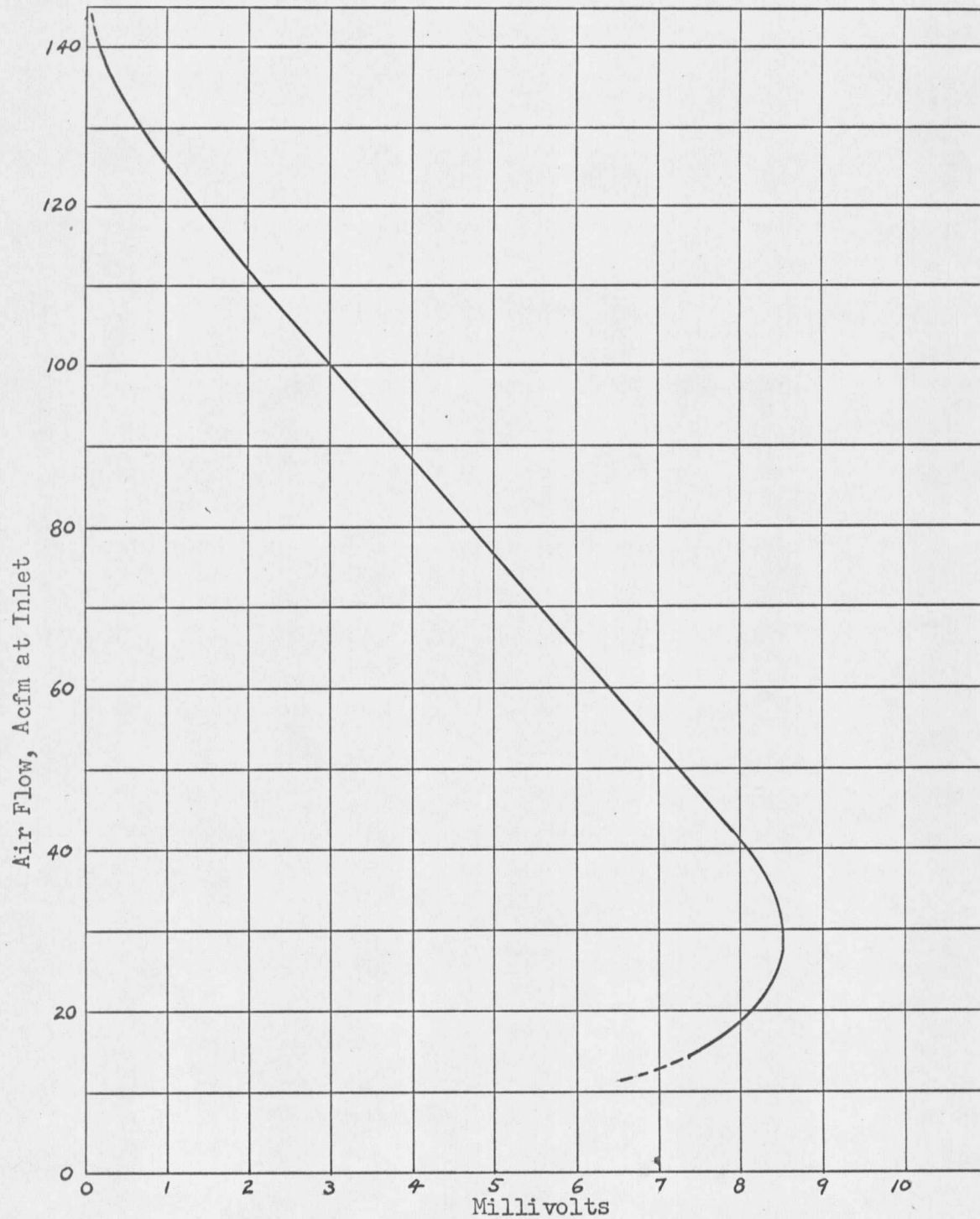


Figure 4. Calibration Curve for Thermocouple Anemometer

Also there is no permanent record from which to later recheck a velocity reading. Despite these drawbacks, the propeller anemometer provides a reasonably accurate and reliable means for measuring air flow.

During calibration of the anemometers, a laminar flow element, produced by the Meriam Instrument Company, was used with an accompanying centrifugal blower as the air flow standard. This laminar flow device, applicable to air flows from 0 to 200 standard cfm (at 29.92 in. Hg and 70 F), consists of parallel, capillary-size tubes through which the air flows. The pressure drop across the tubes has a straight-line relationship with the mass air flow. A calibration curve of this relationship, showing pressure drop across the tubes (in. of water) versus standard air flow (lb per min) is supplied with the element.

With the laminar flow element connected at the shelter exhaust, calibration of both the windmill and the thermocouple anemometer was undertaken. Tests covering the entire expected flow range were conducted at air intake temperature intervals of two degrees from 62 F to 90 F. Humidity of the intake air was varied during several of the runs by spraying steam into the stairwell entrance of the shelter. Calibration of both instruments is not affected by changes in intake air temperature or humidity.

The calibration curve for the propeller anemometer, showing apparent or instrument air velocity (calculated from instrument and stop watch readings) versus actual cfm, is nearly a linear relationship. A representation of this curve is shown in Figure 5. (Below an instrument air velocity of 18 ft per min the anemometer vanes do not rotate.) All calibration points, a total of 87, for all temperatures and humidities lie

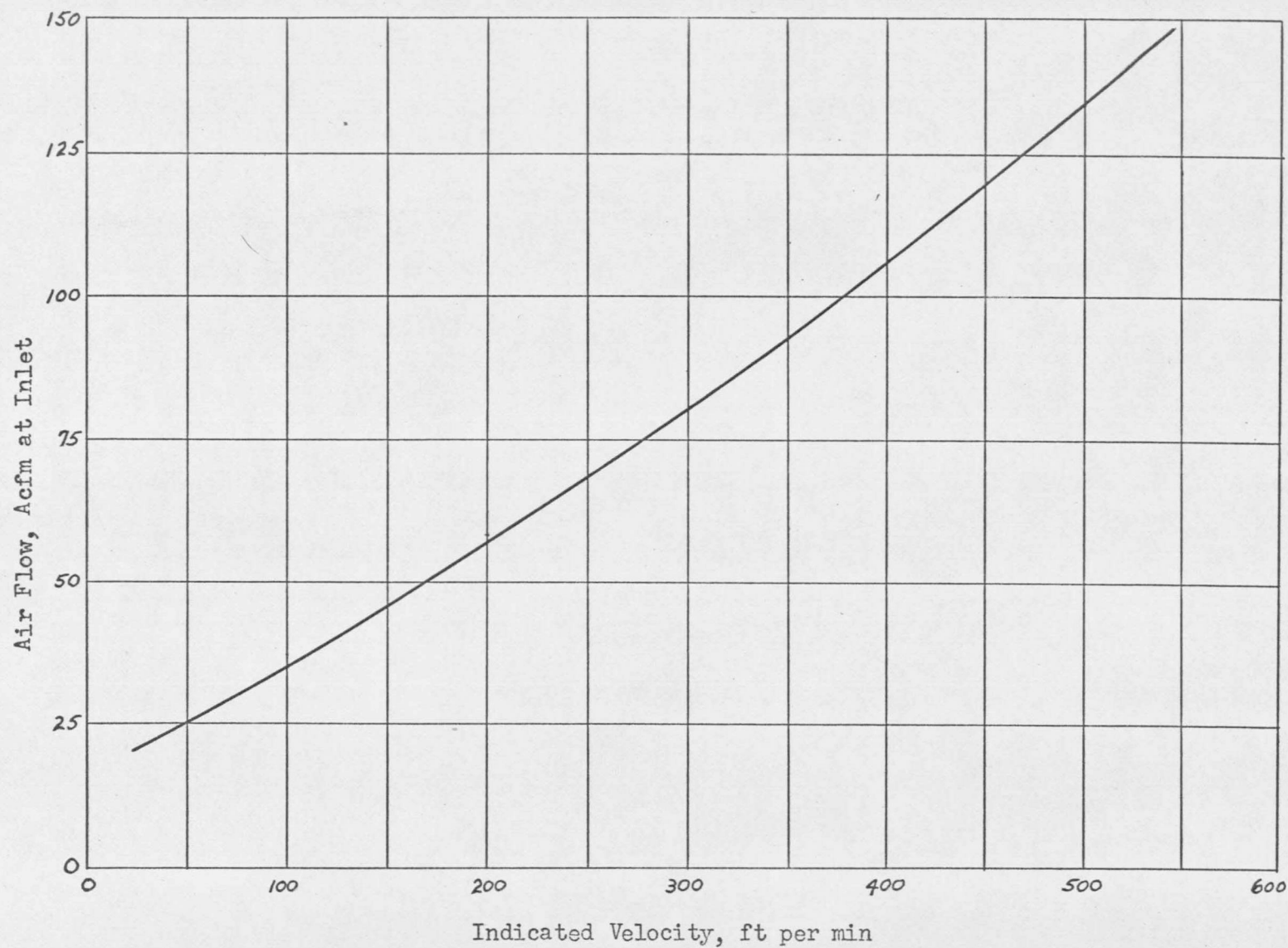


Figure 5. Calibration Curve for Propeller Anemometer

within +3.5 and -3.3 percent of the points dictated by the calibration curve drawn.

Test points for the calibration curve described earlier for the thermocouple anemometer are all within +4.9 and -7.6 percent (these being the two most erratic points) of the values read from the curve drawn.

Although the windmill and the thermocouple anemometer do not extend to extremely low flows, they do permit reliable air flow measurement from 19 to 143 acfm. Both methods are independent of ambient air temperature and humidity changes, and either method provides results within good engineering accuracy. Because the thermocouple anemometer is the least accurate and the least comprehensive of the two instruments, it is used only to confirm the readings obtained by the propeller anemometer. The two instruments are used simultaneously when testing.

CHAPTER 3

TEMPERATURE MEASUREMENTS

To properly evaluate the effect of the flame in the chimney, stack as well as room and ambient temperatures must be determined. The first attempt to measure temperature in a straight 6 in. diameter, 15 ft long stack involved placing copper-constantan thermocouples every few feet. The thermocouple wires, with original insulation only, were inserted through small holes in the stack wall. The thermocouple junctions were located at about the center of the gas stream. This arrangement proved to be inadequate because representative temperatures along the stack could not be obtained. Wide variations in temperatures at different locations along the stack were caused by thermocouples "seeing" the flame at the stack bottom and because of temperature and gas velocity profiles across the stack cross section.

To improve the accuracy of temperature readings and to provide a means of taking a traverse across the stack, movable thermocouple probes were built. The first probes built, shown in Figure 6a on page 19, consist of 20 gage iron-constantan thermocouple wires inside steel tubes about 11 in. long. The tubes pass through brass cinch adapter tubing fittings which are soldered into the stack wall. The angled end of the probes permits the thermocouple to be exposed to the gases, yet provides shielding against direct radiation from flame.

A temperature traverse was taken with these probes at an average stack temperature of about 500° F. Some very unusual results were obtained, especially with the probe nearest the combustion chamber. A temperature difference as high as 175° F occurred between two points, each point being

about 1/2 inch from the stack wall but on opposite sides. The higher of the two temperatures was at the "far wall" point, i.e., with the probe extending all the way across the stack. Similar temperature differences were found at all points along the stack length.

It appeared that heat was being rapidly conducted away from the couple junction by the steel tubing and the large 20 gage wires. The highest conduction rate was when the probe was fully extended, i.e., when the thermocouple junction was at the "near" wall point. The higher rate is due to the cool air surrounding the outside of the stack.

To reduce heat conduction to a minimum, the probe design was changed. The tubes were cut off "square" on the end, the 20 gage thermocouple wires replaced by smaller 30 gage wire, and porcelain insulators placed inside the tubes around the wires over nearly the full length of the tube. As a type of heat sink for the thermocouples, brass balls about 3/16 inch in diameter were fused onto the couple junctions. This probe design is shown in Figure 6b.

Temperature traverses with this type probe gave good results at stack heights of 5 ft and above. Temperature readings at the near and far points all agree within eight percent of each other for these heights. This arrangement permits the determination of representative stack temperatures at points 5 ft or more from the flame.

The brass ball temperature probe is not satisfactory for points close to the flame. The probe at the bottom of the stack was eliminated because it received heat directly from flame and thus gave a reading more representative of combustion chamber temperature rather than stack gas temperature.

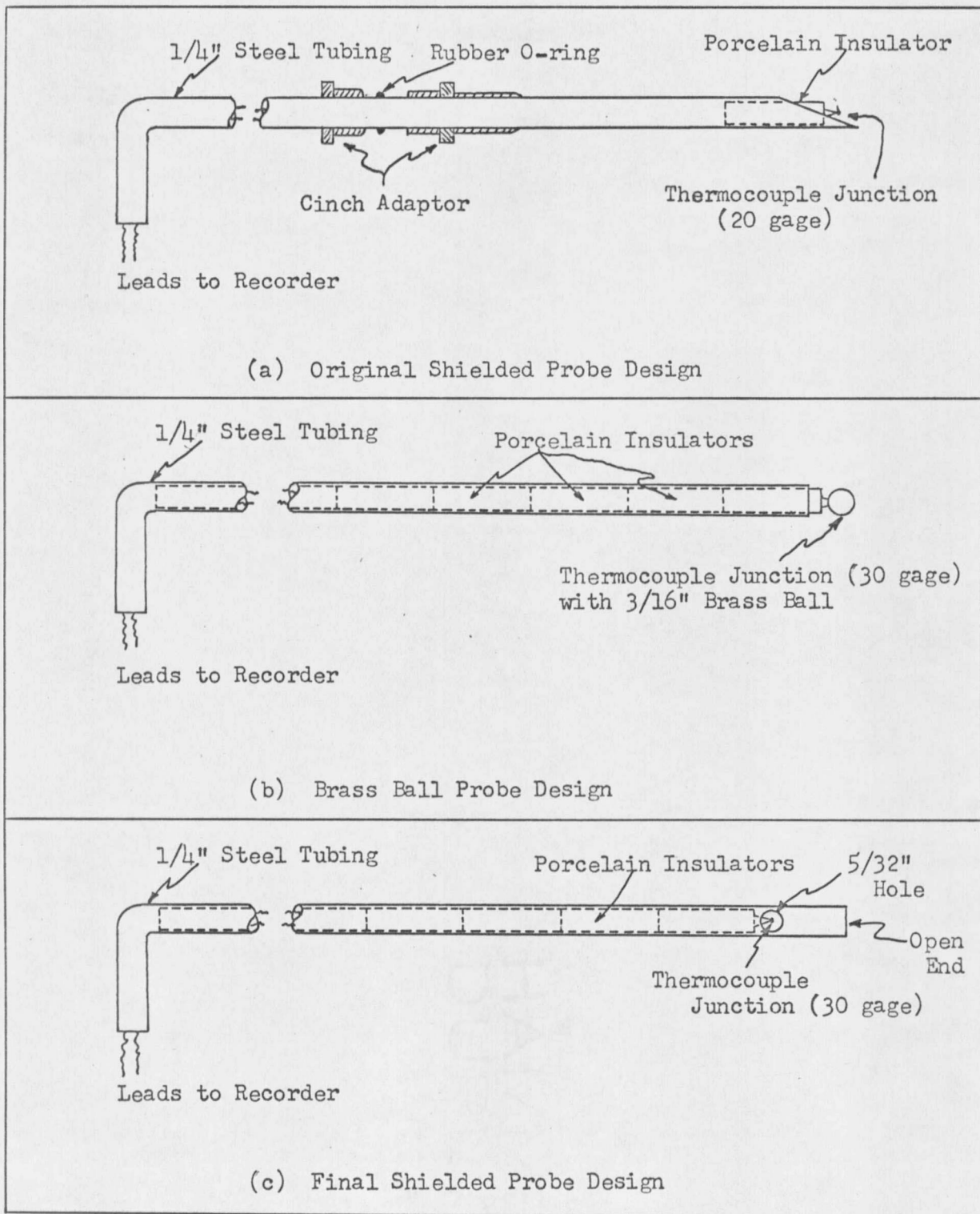


Figure 6. Thermocouple Probe Designs

This reading was of no value in determining the mean or average stack temperature.

It was decided that the first temperature along the stack would be read at a point 2-1/2 ft from the bottom. A third type of probe was used for this point. See Figure 6c. This probe, similar to the ones discussed above uses a 30 gage junction with no additional heat sink. The thermocouple is in the center of a 5/32 inch hole drilled through the tube at about one-half inch from its end. The probe, used with the small hole in the tube perpendicular to gas flow, reduces radiation and turbulence effects and gives fairly reliable and stable temperature readings.

In order to obtain the average temperature at any stack cross section, plots of temperature versus distance across stack were made. From these plots it was determined that the average cross-sectional temperature for all stack heights is given when the probe thermocouples are at 2 in. from the near wall. (This location applies only for a 6 in. diameter stack.) In an article entitled "Measurement of Mean Temperature in a Duct", appearing in the September, 1961 issue of Instruments and Control Systems, it is stated that when only one temperature sensor is used at a particular stack cross section, it should be placed at 0.58 times the radius away from the center. This is in relatively close agreement with the results obtained through temperature traverse readings. This same procedure for locating temperature probes was used each time the stack diameter was changed.

The final locations decided upon for thermocouple probes along the stack length include the shielded probe at 2-1/2 ft and brass ball probes

at 5 ft and every 5 ft interval thereafter. Thus, for a stack 15 ft high, four stack temperatures are read and recorded.

The mean stack temperature for a particular test is determined by plotting a curve of temperature versus stack height (a linear relationship) and then selecting the temperature at the mid-height of the stack. Thus for a 15 ft stack, the mean stack temperature (T_m or T_{ch}) is taken from the curve at the 7-1/2 ft point. It is estimated that this method of determining the mean stack temperature is accurate to within 5 percent. Other temperatures determined and recorded during testing included those of the shelter space, the room above the shelter, and intake air temperature. These three temperatures were taken by using copper-constantan thermocouples. Stack temperatures were read from a 0-1000° F recorder and the rest from a 0-100° F recorder. Both recorders, Honeywell models using type J couples, were periodically calibrated by inserting one thermocouple in 32° F ice water and adjusting the indicators as required.

CHAPTER 4

PRESSURE MEASUREMENTS

At the outset of the project, the need to measure the pressure drop at the inlet to the shelter in relation to air flow was anticipated. This relationship was needed in order to determine the permissible restriction for shelters in terms of filters and still obtain the required ventilation air.

A type "C" micromanometer, made by E. Vernon Hill & Co., Chicago, was cleaned and filled with a special gage fluid having a specific gravity of 0.797. Although the fluid is not water, the manometer reads in inches of water, the smallest dial division being 0.001 in. of water. The micromanometer is shown in Figure 29 on page 70.

The instrument was located inside the shelter near the stack. The main reason for locating it near the stack is because a pitot tube, used to measure exhaust gas velocity, was installed in the stack and it appeared desirable to have its connection hoses as short as possible. Also a pressure probe was installed at the bottom of the stack for measurement of stack entrance pressure drop. To measure the entrance or room pressure drop (ΔP_{rm}), one tube from the manometer was put through the ceiling to measure ambient pressure and the other tube measures the shelter pressure. All pressure tubes from the pitot tube, the room-drop arrangement, and the stack bottom converge at a manifold consisting of glass tees and short pieces of Tygon tubing. Only two hoses run from the manifold to the micromanometer. With the use of screw clamps in the appropriate places, measurement of any desired pressure drop is relatively simple.

An equation for air flow through the shelter is developed in Chapter 7.

Indicated in this equation, written in terms of inches of water, are two terms, the sum of which accounts for the total pressure drop ΔP_{rm} at the shelter entrance. The individual terms account for the friction loss and the velocity head loss in the intake duct. Theoretical values of ΔP_{rm} , taken from a computer program of the flow equation, are plotted in Figure 7 on page 24 along with experimental ΔP_{rm} values as measured with the micromanometer. Pressures are plotted versus temperature difference, i.e., mean stack temperature (T_{ch}) minus ambient temperature around stack (T_{rm}). Although this graph is for only two different stack configurations, it is representative and shows the relative accuracy and reliability of the micromanometer readings. Reference to pressure readings is again made in subsequent chapters in regard to the use of a filter at shelter intake and velocity determination from pitot tube pressure readings.

The pressure probe at the bottom of the stack consists of a straight piece of 3/16 in. steel tubing, closed at one end, with 3/64 in. diameter holes drilled completely through and spaced about every half inch along the tube length. The small holes are perpendicular to the air flow. Used in conjunction with the micromanometer, the pressure probe permits measurement of the difference in pressure between the shelter space and the bottom of the stack, i.e., the stack entrance loss.

At the beginning of ventilation tests, a 6 in. diameter vertical stack was used. The stack entrance loss was measured at a flow of 50 acfm using a wick-type kerosene burner as the heat generator with no entrance hood at the stack. In order to minimize the stack entrance loss, the most economical and practical hood type had to be determined. Tests were

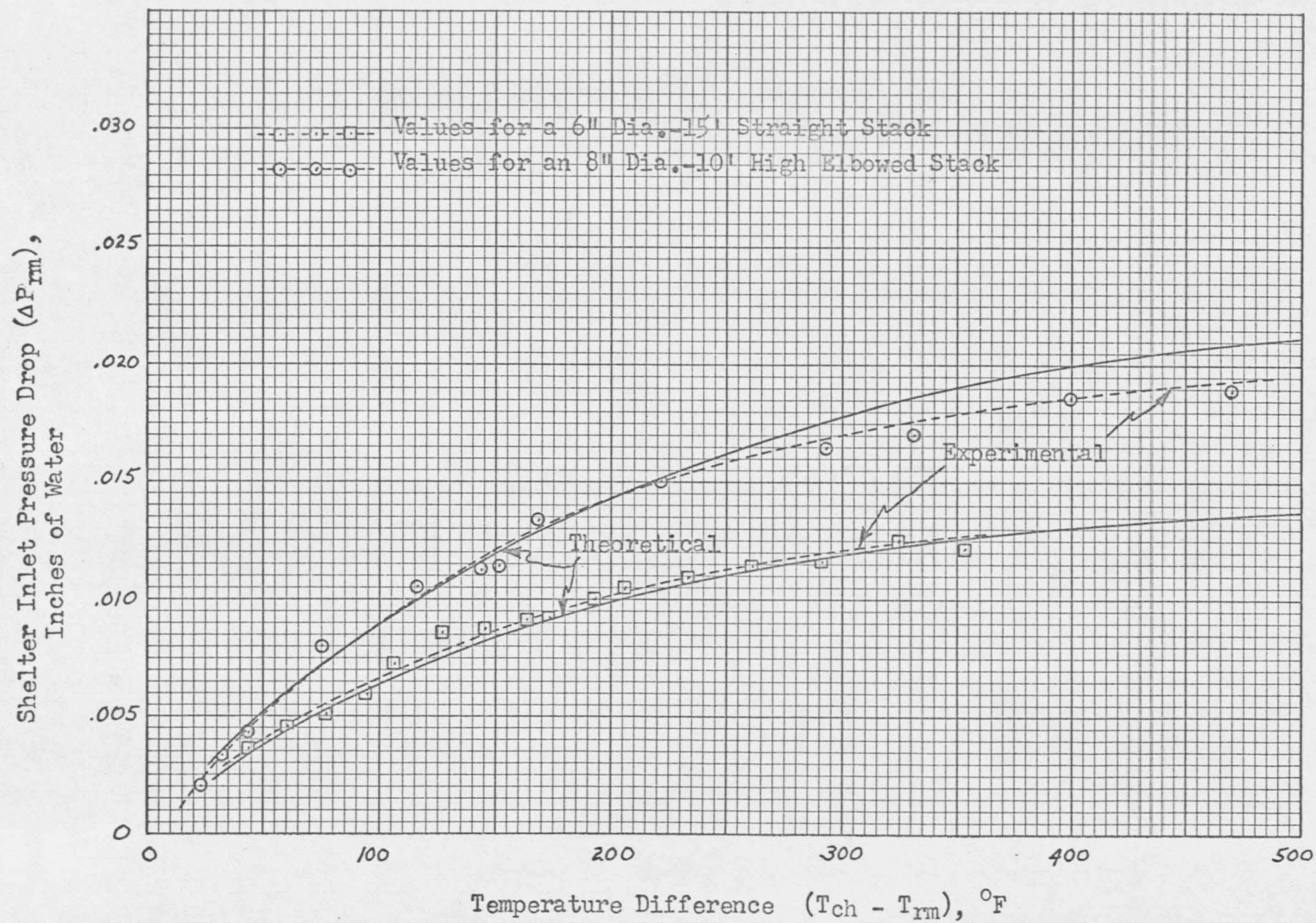


Figure 7. Comparison of Micromanometer Pressure Readings with Theoretical Values

conducted, all at air flows of 50 acfm, with various hood configurations. The first hood, conical in shape, was made from heavy aluminum foil. The top diameter of the hood was 6 in., bottom diameter was 24 in., and height was 16 in. Stack entrance loss was measured and recorded at a flow of 50 acfm.

The second hood, also of a conical shape and made from aluminum foil, was 5.5 in. high with top and bottom diameters of 6 and 12 in., respectively. Again the entrance pressure drop at 50 acfm flow was measured.

The third and final hood tested was made of galvanized sheet metal and had a rectangular shape. Dimensions are: height, 13 in.; width, 10 in.; and length, 17 in. The pressure drop was measured as before. Stack entrance losses for the four conditions described above were compared. Results show that all readings are within 3.5 percent of each other, a negligible difference. From these results it was concluded that air flow by the induced draft method is nearly independent of hood shape and size. Consequently, all further ventilation tests were conducted using the rectangular hood. A rectangular hood is easier to build than a conical one, is cheaper, and better accommodates the various types of combustion equipment discussed in the next chapter. The rectangular hood is shown in Figures 31 and 32 on page 71.

CHAPTER 5

HEAT GENERATING EQUIPMENT

Because it is assumed that no electricity will be available in the shelter, which precludes the use of electrically driven fans for supplying ventilation air, consideration was not given to any combustion equipment requiring electricity for operation or control.

A first thought was that a small pot burner using fuel oil would be ideal, but a test proved differently. Maximum flow obtainable in a 6 in. diameter 15 ft straight stack with the pot burner was about 30 acfm at a stack temperature of nearly 400 F and a heat output of about 36,000 Btuh. The reason for such a low flow is that most of the pressure drop generated was used to induce proper combustion in the pot burner which uses a barometric damper. Little pressure drop was left to induce air to flow through the shelter. The logical conclusion arrived at is that a pot burner which requires a large pressure drop for combustion will not suffice as proper heat generating equipment for shelters.

The next attempt to induce a draft by a flame in the chimney was to utilize a small fuel-oil space heater of the wick-type. With this burner the full benefit of combustion is realized in generating a pressure differential which causes air to flow through the shelter. The range of air flow obtainable with the wick-type space heater in the 6 in. 15 ft high stack is about 35 to 60 acfm. Maximum mean stack temperature is around 160 F.

As another heat source, a single kerosene lamp of the type common a few years back was tried. About 25 acfm of air can be drawn through the shelter when the lamp is operated at "full" flame. A "three-holer" lamp

burner was developed whereby the wicks and glass chimneys of three different kerosene lamps were put on one fuel tank. The rectangular fuel tank holds nearly one gallon of kerosene. The "three-holer" burner is shown in Figure 27 on page 69. With this arrangement it is possible to burn one, two, or three lamps at a time to obtain increased heat output and air flow. The draft generated by burning the three lamps simultaneously is nearly equivalent to that produced by the small wick-type space heater.

In an endeavor to produce air flows greater than 60 acfm, a second wick-type space heater was obtained. A portable stand, consisting of a piece of 1/2 in. plywood with two 2x4 legs, supports the two burners. Two 2 in. holes were cut in the plywood to supply primary combustion air to the burners which set flat on the plywood. With this "dual-burner" arrangement at the bottom of a 6 in. 15 ft high straight stack, a maximum flow of 80 acfm is obtained at a mean stack temperature of about 270 F. The dual-space-heater unit is shown in Figure 31 on page 71 as it appeared during operation. In the same figure are shown the bottom of the intake hood and a scale balance which is used to determine the weight of fuel burned.

As explained earlier, it cannot be assumed that natural gas will be available during shelter occupancy. However, for the purpose of testing, two natural gas burners were utilized in order to reach temperatures and flows considerably above those obtainable by the other methods. Mean stack temperatures of nearly 650 F can be obtained by using the natural gas burners. Details of ventilation results from pertinent tests conducted with each of the heat-generating methods discussed here appear in

Chapter 8.

Besides providing heat to cause air to flow, wick-type burners also furnish illumination to the shelter. Kerosene lamps, because of the transparent glass chimneys, provide more light than the space heaters. The degree of illumination depends upon the distance the burner extends into the hood and upon the wick setting.

CHAPTER 6

EXPERIMENTAL VENTILATION TEST PROCEDURES

The original model for conducting tests of ventilation air flow caused by a flame in a chimney was set up with the intent of making it simple in order to reduce the variables to a minimum. To accomplish this, the stack was installed vertically out of the shelter--no bends or elbows--to minimize friction and velocity loss effects. Intake air was brought into the shelter through a straight length of 8 in. galvanized sheet metal duct--no bends, no elbows, or filters. Because the stack was completely within the surrounding building, the effects of wind, for testing purposes, were eliminated. This is the simplest setup, though perhaps not the most desirable from the standpoint of fallout radiation elimination, that can be achieved in any home shelter. The stack is vertical, there are no variables induced by wind, and inlet air is not restricted by conditions other than the intake duct itself.

It should be mentioned that all tests conducted for purposes of this investigation utilized circular cross-sectioned "Metal-bestos" stack. "Metal-bestos" stack has about a quarter inch air space between an aluminum inside liner and a galvanized sheet metal outside. Thus, it is a type of self-insulated stack. When a stack diameter is mentioned, it always refers to the inside diameter of the aluminum liner.

A group of tests were conducted using varying lengths of 6 in. diameter straight stack. "Straight stack" means one that is completely vertical and contains no elbows, bends, or horizontal lengths. Pressure, temperature, fuel rate, and air flow-rate data were recorded for the heating methods of kerosene lamps and wick-type space heaters. Similar data,

excluding the fuel rate, were taken for the natural gas heating method.

The above data were collected in what the author terms a "closed door" test. A closed door test is one in which the door to the shelter is closed and sealed. All air coming into the shelter must pass through the 8 in. intake duct. In this type of test, the flow rate may or may not be greatly subdued by the restriction offered by the intake duct, depending upon the stack length, diameter, and temperature (or the amount of air trying to flow). This type of test is also designated as closed door in order to distinguish it from an "open door" test which, as the term implies, is one conducted with the shelter door open. In this case there is no measurable pressure drop at the intake and thus flow rate is not limited by any intake restriction. The limiting factor for flow is the stack itself. Thus air flow for an open door test is sometimes called "unrestricted flow" and that for a closed door test is called "restricted flow". In both cases the restriction or absence of restriction refers to intake conditions only.

Open door tests for 6 in. diameter stacks followed the closed door tests. Because the air was not forced to pass through the intake duct, neither the propeller anemometer nor the thermocouple anemometer could be used to measure the air flow. Instead, a pitot tube, installed in the stack at mid-height, was used to measure the intake air flow. To do this the tube was calibrated during the previous closed door test. Further explanation of calibration follows.

The difference between the static and the stagnation pressures ($p_s - p_o$) in the stack is measured using the micromanometer. The apparent stack air velocity is then calculated using the basic pitot tube equation.

$$(1) \quad V_o = \sqrt{2g(p_s - p_o)/\gamma} \quad (\text{ft per sec})$$

in which γ is the air density. Equation (1) can be simplified by letting $\Delta p = (p_s - p_o)$; $V_{sp} = V_o$ = stack velocity as determined from the pitot tube readings; and solving for the air density from the perfect gas law.

Clearly,

$$\gamma = \left(\frac{w}{V}\right) = \frac{P_B}{R(T_{ch} + 460)} = \frac{P_B(0.491)(144)}{(53.3)(T_{ch} + 460)} = \frac{1.327 P_B}{(T_{ch} + 460)}$$

and by substituting these and the conversion factor, 1 in. H₂O = 5.203 psf, into equation (1), we obtain

$$V_{sp} = \sqrt{2g \Delta p / \gamma} = \sqrt{2(32.2)(\Delta p)(5.203)(T_{ch} + 460) / 1.327 P_B}$$

$$(2) \text{ or } V_{sp} = \sqrt{252.2(\Delta p)(T_{ch} + 460) / P_B}$$

where V_{sp} = stack gas velocity as determined from the pitot tube readings, ft per sec

Δp = pitot tube pressure reading, in. of H₂O

T_{ch} = mean stack or chimney temperature, °F

P_B = barometric pressure, in. of Hg.

To find the actual cfm of air flowing in the stack, we can write

$$\frac{(P_i)(ACFM)_i}{(P_s)(ACFM)_s} = \frac{w_i R_i (T_i + 460)}{w_s R_s (T_{ch} + 460)}$$

and because the weight of air flowing per minute and the gas constants are the same at inlet and stack conditions, we may write, if we assume that the barometric pressures are the same,

$$(3) \quad (ACFM)_s = (ACFM)_i \frac{(T_{ch} + 460)}{(T_i + 460)}$$

where T_{ch} = mean stack air temperature, °F

T_i = inlet air temperature, °F

$(ACFM)_i$ = actual air flow at inlet, cu ft per min

$(ACFM)_s$ = actual air flow in stack, cu ft per min.

The terms on the right side of equation (3) are determined from the closed door test. The air flow in the stack is also given by multiplying the cross sectional stack area (A_s) by the true stack velocity (V_{st}). Hence,

$$ACFM_s = A_s V_{st}$$

$$(4) \quad \text{or} \quad V_{st} = ACFM_s / A_s$$

where A_s is in sq ft. V_{st} is then compared with V_{sp} and a constant or correction factor (C_1) determined from the relationship,

$$(5) \quad V_{st} = C_1 V_{sp}$$

For an open door test, then, the pitot tube Δp and the mean stack temperature (T_{ch}) are read, the apparent stack velocity is calculated from equation (2), correction factor C_1 is applied, and the actual flow at inlet conditions is found from equations (4) and (3).

Equation (1) is normally used only for incompressible fluids. It can be used for air with little error if the pitot tube is calibrated for each change in stack configuration over a wide range in velocities.

After tests had been completed with 6 in. diameter straight stacks, two 90° four-piece elbows and a 2 ft long horizontal length of single-walled duct were added to the stack bottom. See Figure 32 on page 71. These additions permit about 2 ft 9 in. between the centerline of the original vertical stack and the centerline of the second elbow. It is anticipated that about 2-1/2 ft of horizontal length is all that will be required in the majority of shelter situations to provide adequate shielding from fallout radiation.

In a prototype basement shelter, the elbows and horizontal length would permit running the stack out of the shelter near the ceiling, through the concrete wall into surrounding earth or adjacent basement space, and then vertically.

Both open door and closed door tests were run with this "elbowed" stack arrangement. The author often refers to this arrangement as an "equivalent length" stack. The equivalent length applies to the elbows and is discussed in detail in the next chapter.

Several tests were conducted with a one-inch layer of fiber glass insulation on the intake hood, elbows, and horizontal length of stack to try to determine the amount of heat that is added to the shelter in the absence of insulation. A prototype shelter in which the horizontal portion of the stack runs through a concrete wall and adjacent earth more closely represents the insulated condition, rather than the uninsulated one, because the concrete and earth would tend to contain the heat to the stack. Insulation on the hood and on that portion of the stack extending into the shelter is all that would be required to complete the insulated test-facsimile.

To find the shelter heat-addition difference, tests with and without insulation at constant mean stack temperatures were run. Surface temperatures of the stack inside the shelter and of the hood were taken for both conditions. At a mean stack temperature of 250 F, the surface of the top of the hood reached nearly 450 F with no insulation. With one inch of fiber glass insulation and the same mean stack temperature, the surface temperature of the insulation on the hood was reduced to about 145 F.

This surface temperature reduction is important when trying to minimize heat addition by radiation.

Fuel rates for the insulated and the uninsulated stacks were determined by weighing the amount of fuel used over a period of time. From this data the heat saved by using insulation can be calculated. See Chapter 8 for results.

The tests described above, with the exclusion of the insulation tests, were also conducted with 8 in. diameter stacks ranging in vertical heights from 5 ft to 15 ft. It was deemed unnecessary to conduct insulation tests for the 8 in. stacks because a percentage value, determined in the 6 in. tests, can be applied to the other diameters if fuel saving figures are desired.

To determine the effect on air flow from the use of ventilator caps, tests with 8 in. stacks were made using two types of caps at the stack outlet--a two-dimensional Artis cap and an all-directional Belmont cap. (Directional refers to wind conditions.) Results appear in Chapter 8.

Testing was concluded after the 8 in. stack series of tests were finished. It was believed that sufficient data had been collected with which to prove or disprove the original hypothesis.

CHAPTER 7

THEORETICAL ANALYSIS OF SHELTER VENTILATION

To predict air flow through the shelter for stacks and conditions in addition to those tested, an equation which can be applied to various inlet and outlet configurations and conditions is herein developed and discussed.

The following equates the positive buoyant force of a column of hot gases with the force of friction required to establish and maintain flow.

$$(6) \quad \frac{H}{62.4} (\rho_{rm} - \rho_{ch}) = (h_v) + (h_L) = K \frac{v^2}{2g} + f \frac{L}{D} \frac{v^2}{2g}$$

where H is vertical stack height, ft; and 62.4 is the density of water at 70° F in lb/cu ft, included so that both sides of the equation are expressed in the common "feet of fluid flowing" units. In the case of air, it is more convenient to deal with the velocity head (h_v) and the friction loss head (h_L) in "inches of water" units rather than "feet of air" units. The transformation can be made by again using the fundamental buoyancy relation,

$$h_a \rho_a = \frac{h}{12} \rho_w$$

where h_a = feet of air; ρ_a = density of air in lb/cu ft; h = inches of water (gage); and ρ_w = density of water at 70° F in lb/cu ft. Thus, it is seen that

$$(7) \quad h_a = \frac{h}{12} \frac{62.4}{\rho_a} = 5.2 \frac{h}{\rho_a}$$

For the velocity head term, h_v , in inches of water,

$$K \frac{v^2}{2g} = 5.2 \frac{h_v}{\rho_a} \quad \text{or} \quad h_v = K \frac{v^2 \rho_a}{5.2(2g)}$$

With stack air velocity, V , as feet per minute,

$$(8) \quad h_v = K \frac{v^2 \ell_a}{(5.2)(2)(32.2)(3600)} = K \frac{v^2 \ell_a}{1,202,000} = K \left(\frac{V}{1096} \right)^2 \ell_a$$

By substituting equation (8) into equation (6), we obtain the driving force relationship expressed in inches of water. Hence,

$$(9) \quad \frac{H}{62.4}(12)(\ell_{rm} - \ell_{ch}) = \frac{H}{5.2}(\ell_{rm} - \ell_{ch}) = K \left(\frac{V}{1096} \right)^2 \ell_a + f \frac{L}{D} \left(\frac{V}{1096} \right)^2 \ell_a$$

Equation (9) applies to shelter air flow in general if terms with the appropriate subscripts are used. Thus,

$$(10) \quad \frac{H}{5.2}(\ell_{rm} - \ell_{ch}) = K_1 \left(\frac{V_1}{1096} \right)^2 \ell_{in} + f_1 \frac{L_1}{D_1} \left(\frac{V_1}{1096} \right)^2 \ell_{in} + K_2 \left(\frac{V_2}{1096} \right)^2 \ell_{ch} + f_2 \frac{L_2}{D_2} \left(\frac{V_2}{1096} \right)^2 \ell_{ch}$$

where: H = vertical height of stack, ft

ℓ = air density, lb/cu ft

K = duct or stack entrance loss coefficient

V = air velocity, ft/min

L = duct or stack equivalent length, ft

D = duct or stack diameter, ft

f = duct or stack friction factor

rm = conditions of space surrounding the stack, usually ambient air conditions

1, in = conditions at shelter intake

2, ch = conditions at stack or shelter exhaust

For the purpose of using equation (10) to predict shelter ventilation rates that can be compared to test results, the constants and coefficients had to be evaluated for conditions that existed during experimentation.

As stated in a previous chapter, all closed door tests were made with the

air passing into the shelter through an 8 in. diameter duct. This allows the D_1 and L_1 terms of equation (10) to be fixed at 0.667 ft and 3.834 ft, respectively. Values for both of the entrance loss coefficients, K_1 and K_2 , are assumed for the worst possible entrance conditions, i.e., $K_1 = K_2 = 1$. This assumption is based on several facts and conditions. The intake duct is a re-entrant type of pipe entrance for which the nominal value of loss coefficient is 0.8. In the end of the duct, however, is an aluminum-foil tubular grid which offers not only additional entrance loss, but also additional friction loss. Entrance to the stack, for all test conditions, is made through the rectangular hood described earlier. It represents a hood condition similar to that described by W. C. L. Hemeon in Plant & Process Ventilation. He states that for a hood with low face velocity and relatively low stack temperatures, the entry coefficient is 0.9. This condition is aggravated in the test case because the heat generating equipment extends into the hood a short distance and thus disturbs the flow. For these reasons the author feels that the assumption of $K_1 = K_2 = 1$ is very reasonable.

In considering the friction factors for the intake duct and the stack, it is felt that the standard tabulated values of f , given as a function of Reynolds number and pipe relative roughness, are not entirely applicable. Both the intake duct and the stack contain obstructions not normally found in pipes or ducts. The intake duct houses the tubular grid near the entrance, the special thermocouple anemometer, the propeller anemometer, and a thermocouple probe. The stack contains pressure taps, thermocouple probes, and, in the case of the 6 in. diameter stack, a layer of soot on

the wall surfaces. The friction factors were thus obtained experimentally from a test performed with a 6 in. diameter 15 ft long straight stack. A flow was induced in the system and the room pressure drop, the flow at inlet conditions, and the necessary temperatures were measured. The friction factor at inlet, f_1 , was then calculated by equating the measured ΔP_{rm} to the first two terms on the right-hand side of the equation (10) and solving for f_1 . The value arrived at in this manner is $f_1 = 0.415$. When this is compared to tabulated friction values, it appears to be extremely high. But when the resistance offered by all of the equipment in the duct is considered, this value of f_1 seems reasonable. With f_1 thus established, equation (10) was solved for the remaining unknown quantity, f_2 . The value of f_2 determined in this manner is 0.064. Another assumption made at this point was that the friction factors would remain constant for all diameters and lengths of stack considered. It is shown in the next chapter covering results that this assumption is on the conservative side.

The constants and coefficients determined above can be substituted into equation (10) to yield a somewhat refined flow equation. Performing the substitutions we get,

$$\frac{H}{5.2}(\ell_{rm} - \ell_{ch}) = 1\left(\frac{V_1}{1096}\right)^2 \ell_{in} + \frac{(0.415)(3.834)}{0.667}\left(\frac{V_1}{1096}\right)^2 \ell_{in} + 1\left(\frac{V_2}{1096}\right)^2 \ell_{ch} + 0.064 \frac{L_2}{D_2}\left(\frac{V_2}{1096}\right)^2 \ell_{ch}$$

which, when simplified further, yields

$$(11) \quad \frac{H}{5.2}(\ell_{rm} - \ell_{ch}) = 3.385\left(\frac{V_1}{1096}\right)^2 \ell_{in} + \left(\frac{V_2}{1096}\right)^2 \ell_{ch} \left[1 + 0.064 \frac{L_2}{D_2}\right]$$

Upon close inspection of this last equation, the variables are seen to be barometric pressure and ambient, stack, and inlet temperatures (all from density considerations, i.e., $\rho = P_B(0.491)(144)/R(T+460)$), intake and stack air velocities, stack diameter D_2 , stack vertical height H , and stack equivalent length L_2 .

The equivalent stack length (L_2) warrants further discussion. When the stack is one with no elbows or bends, $H = L_2$. The other condition considered is when the stack contains two 90° elbows and a 2 ft section of horizontal pipe. In a manner similar to that for finding the friction factors, an equivalent pipe length for one 6 in. 90° elbow was calculated to be 4.2 ft. The total equivalent length of both elbows and the horizontal run is thus 10.4 ft. Reference to the 1963 ASHRAE Guide & Data Book indicates that the equivalent length of such an elbow is 8 ft. This latter value, however, is based on a relatively long duct system containing "full" flow, i.e., forced circulation. When used in this sense, it is not an extremely critical value. In the test stack the equivalent length has a great influence on flow as can be seen from the equation. The use of the lower value of 4.2 ft per elbow is justified because of the very short duct arrangement and because "full" flow around the elbows is never realized.

The final assumption for use with equation (11) is that the equivalent length for all diameter elbows is constant at 4.2 ft per elbow. This is again a conservative estimate.

A digital computer program of equation (11) was written in order to predict flows for many different conditions of stack diameters, stack

lengths, temperatures, pressures, etc. Values from the computer program are termed "theoretical" values and are presented in the next chapter. It may be well to mention that for open-door test predictions, the first term on the right-hand side of equation (11) vanishes. This means that there is essentially no restriction to flow at the shelter inlet.

All tests were conducted at approximately 5000 ft altitude (Bozeman, Montana) and thus the results are in terms of actual conditions, i.e., barometric pressure ($P_B \approx 25$ in. Hg) and temperatures at 5000 ft. The computer equation was also solved using standard or sea level conditions ($P_B = 29.92$ in. Hg and $T_{\text{ambient}} = 70$ F) to determine the effect of altitude on flow. See Chapter 8 for results.

CHAPTER 8

RESULTS OF PERTINENT TESTS AND THEORY

Much of the data collected from experiments and from the computer programs are presented in this chapter in the form of graphs. The first eight figures of air flow versus temperature difference compare experimental results with theoretical results in order to give the reader an idea of the relative accuracy of the flow equation with its associated assumptions. The remaining figures supply information on inlet pressure drop, fuel rates, theoretical air flow for stacks of many diameters and lengths, altitude, and ventilator caps.

Figures 8 and 9 show ΔT , i.e., mean chimney temperature (T_{ch}) minus ambient temperature around stack (T_{rm}), versus rate of air flow for 6 in. diameter stacks. The friction factors, f_1 and f_2 in equation (10), were determined from the experimental curve for the straight stack closed door test, Figure 8. Theoretical values are within 2-1/2 percent or less of the experimental values at all points.

The 25 ft equivalent length experimental curve in Figure 9 was used to determine the equivalent length of the two 90° elbows. Experimental values are again about 2-1/2 percent from theoretical ones. The theoretical values for the other curves in this figure, as well as for all other curves of all diameters, were determined with fixed values of friction factors and elbow equivalent length. These fixed values are $f_1 = 0.415$, $f_2 = 0.064$, and eq. length = 4.2 ft/elbow. It should be mentioned that an equivalent length stack of, say 40 ft, means one with a vertical height of 30 ft. That is, (30 ft vertical) + (2 ells) $\left(4.2 \frac{\text{eq. ft}}{\text{ell}}\right)$ + (2 ft horizontal) = 40.4 total eq. stack length. The total eq. length is rounded off to the

nearest whole number when presented in the graphs.

The open door comparisons for the 6 in. stacks are shown in Figures 10 and 11. Experimental values are again in close agreement with theoretical values. This indicates that our choice of f_2 and of elbow equivalents is appropriate, at least for the 6 in. - 15 ft. stack, since the intake terms in equation (10) go to zero with an open door. The resulting theoretical values are therefore not influenced by the inlet friction factor.

Air flow-temperature relations for 8 in. diameter stacks with closed door are shown in Figures 12 and 13. Because experimental values are somewhat higher than the theoretical ones, it appears that in equation (10) one or more of the factors, perhaps the entrance coefficients, friction factors or elbow equivalents, are not constant as assumed. The author believes that the principal factor contributing to the higher experimental values is the stack friction factor, f_2 . Original determination of f_2 was for a 6 in. stack which had been in place for nearly five months. Over this period of time a heavy layer of soot had collected on the stack walls which inherently offers more restriction to flow than clean walls. The 8 in. stack, however, had no soot on the walls when tested. The aluminum liner was shiny, smooth, and clean and consequently offers less restriction to flow than a sooted one. The assumption of a constant friction factor is thus a conservative one. As a result, theoretical curves include a margin of safety for predicting air flow.

Results of open door tests on 8 in. stacks appear in Figures 14 and 15. Agreement between experimental and theoretical values is again within

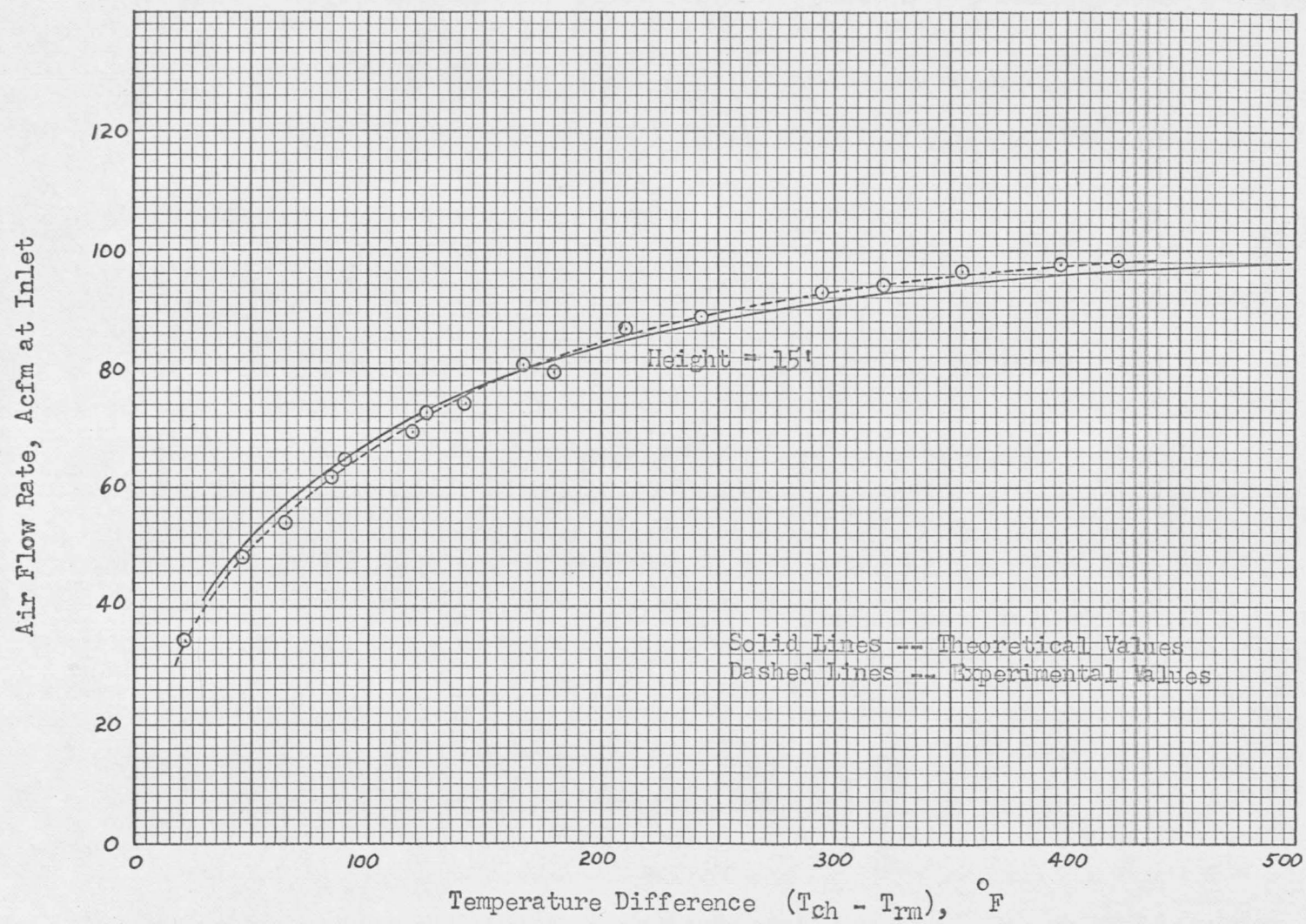


Figure 8. Air Flow vs. Temperature Difference for 6" Diameter Stack.
Closed Door Test --- Straight Stack

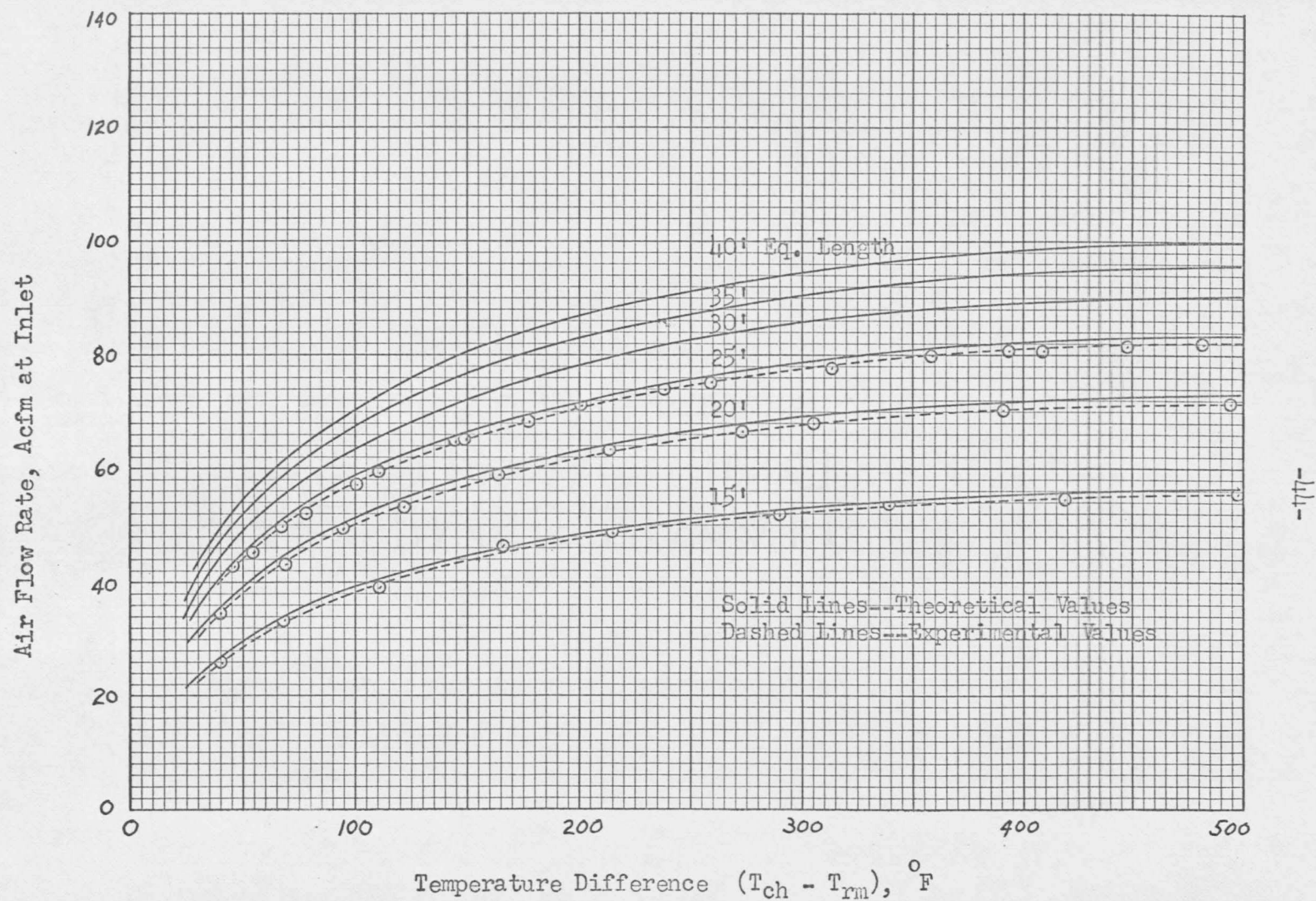


Figure 9. Air Flow vs. Temperature Difference for 6" Diameter Stack
Closed Door Tests -- Elbowed Stack

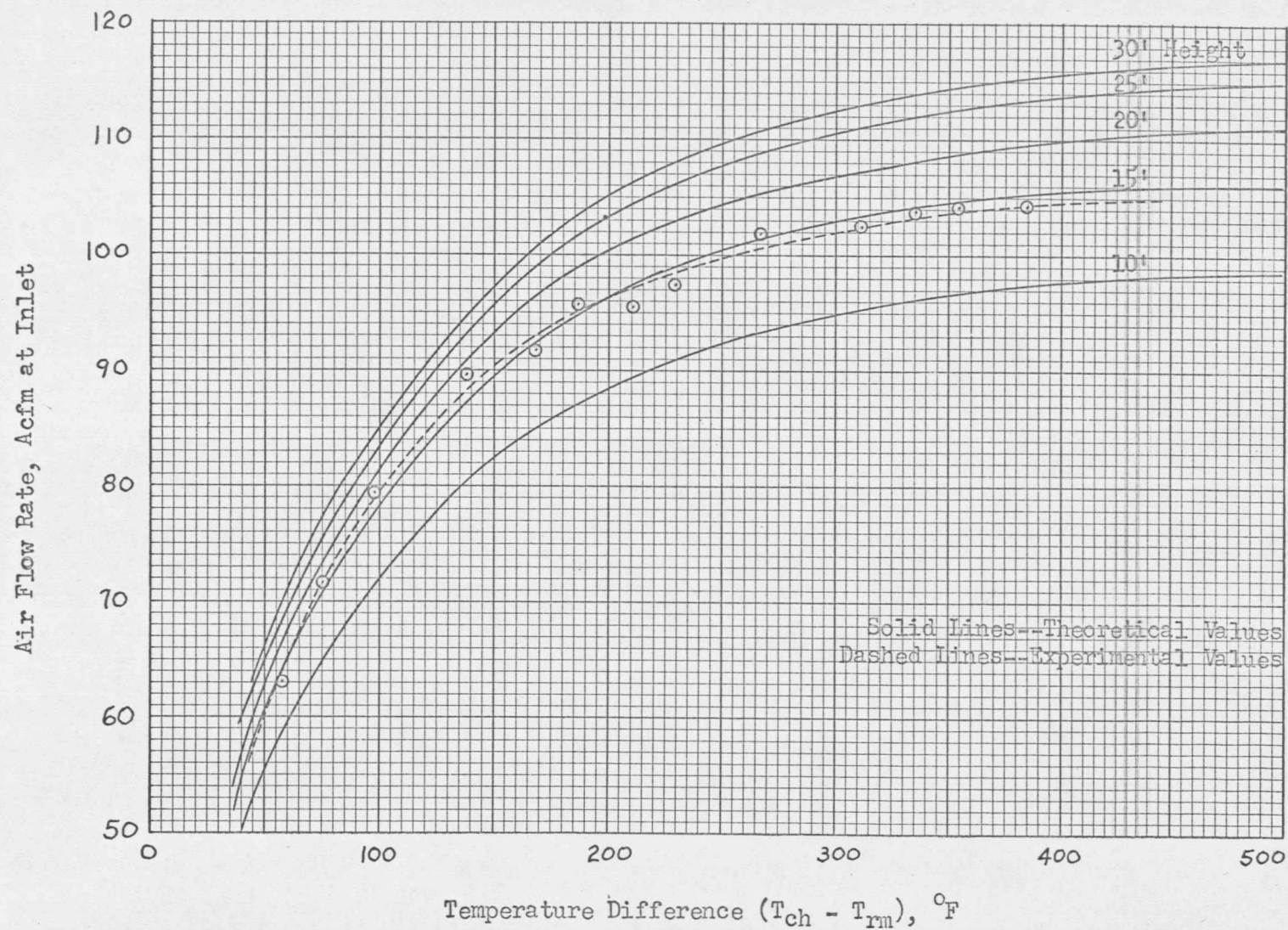


Figure 10. Air Flow vs. Temperature Difference for 6" Diameter Stack
Open Door Tests -- Straight Stack

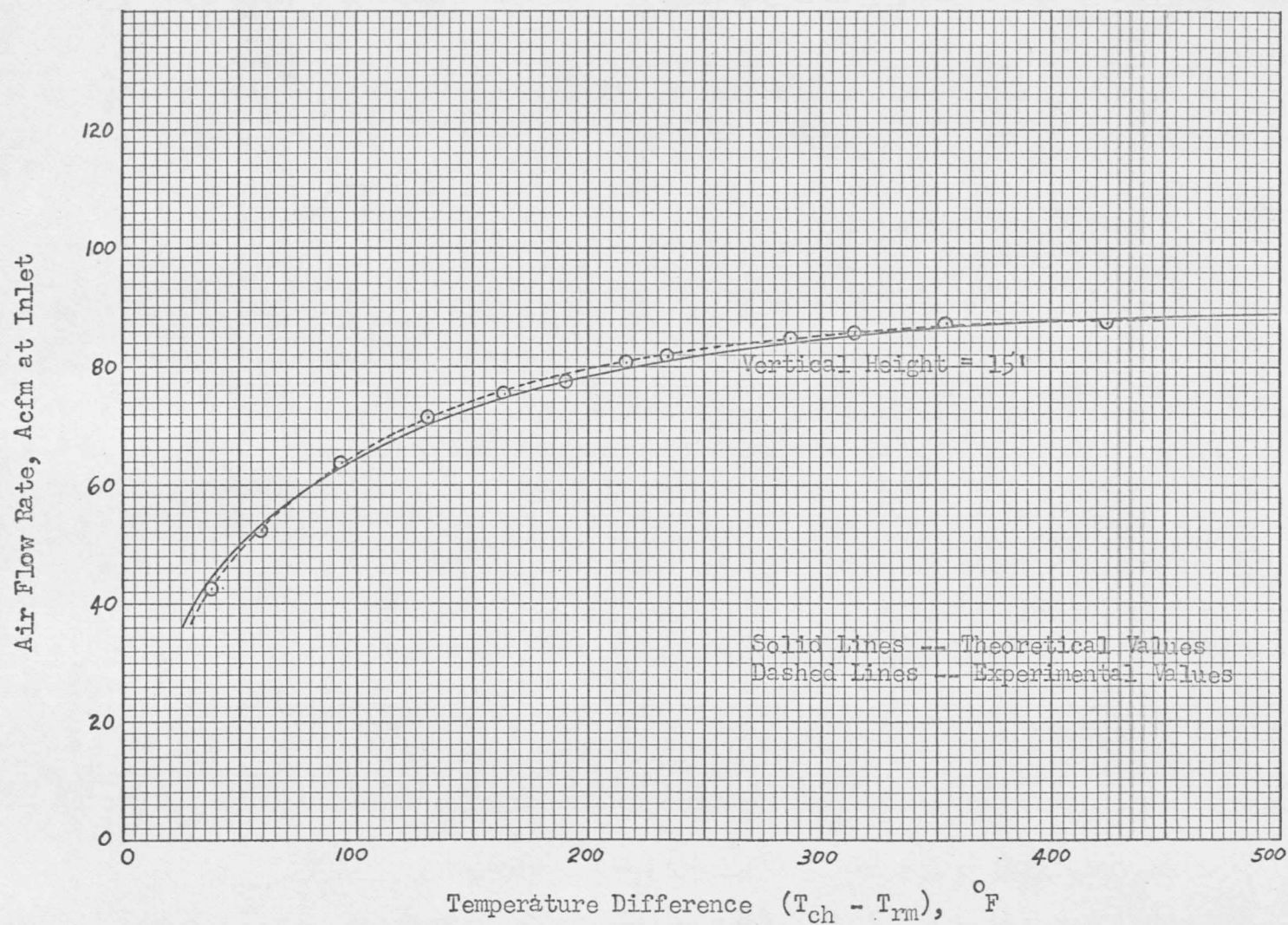


Figure 11. Air Flow vs. Temperature Difference for 6" Diameter Stack.
Open Door Test -- Elbowed Stack

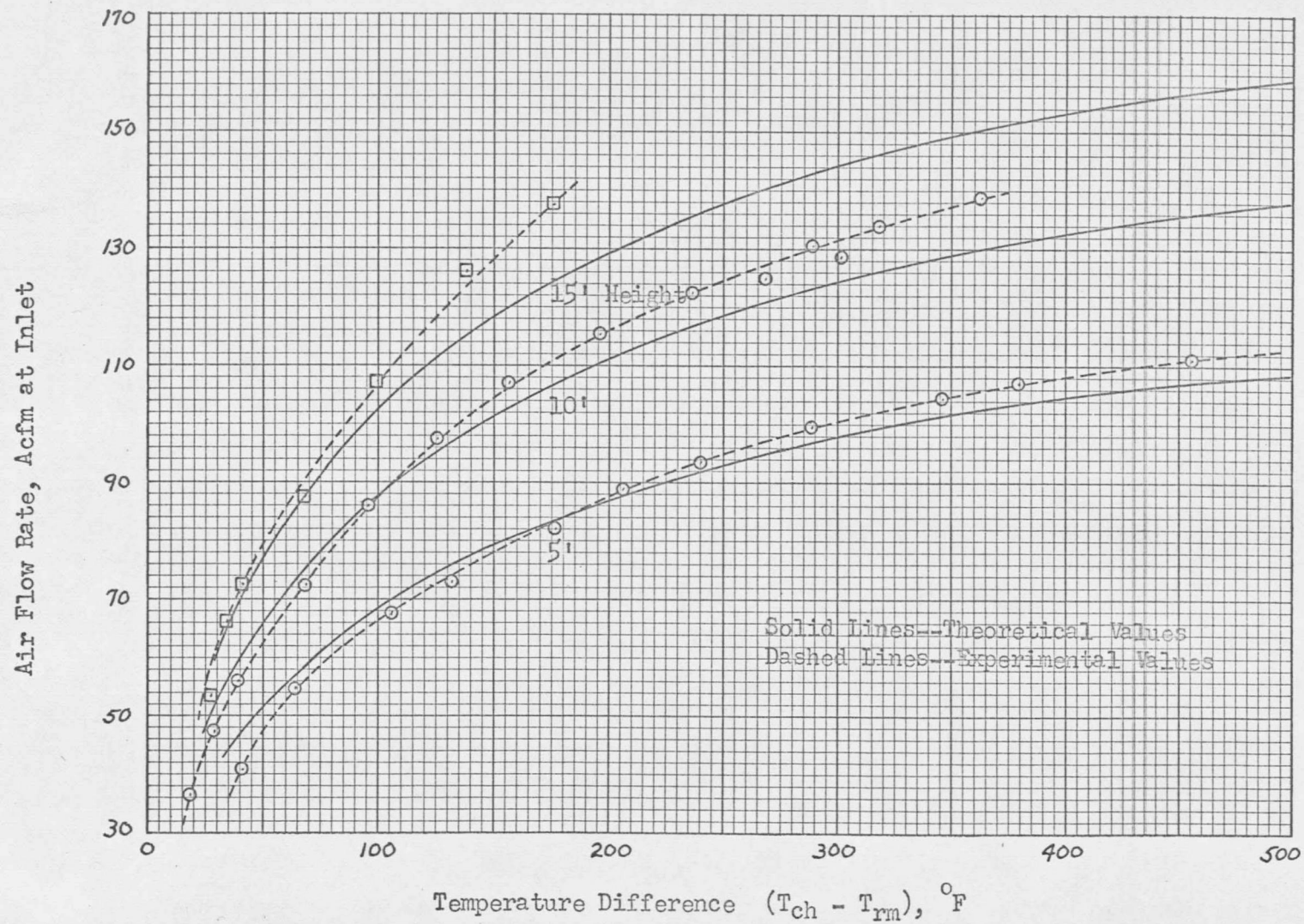


Figure 12. Air Flow vs. Temperature Difference for 8" Diameter Stack
Closed Door Tests -- Straight Stack

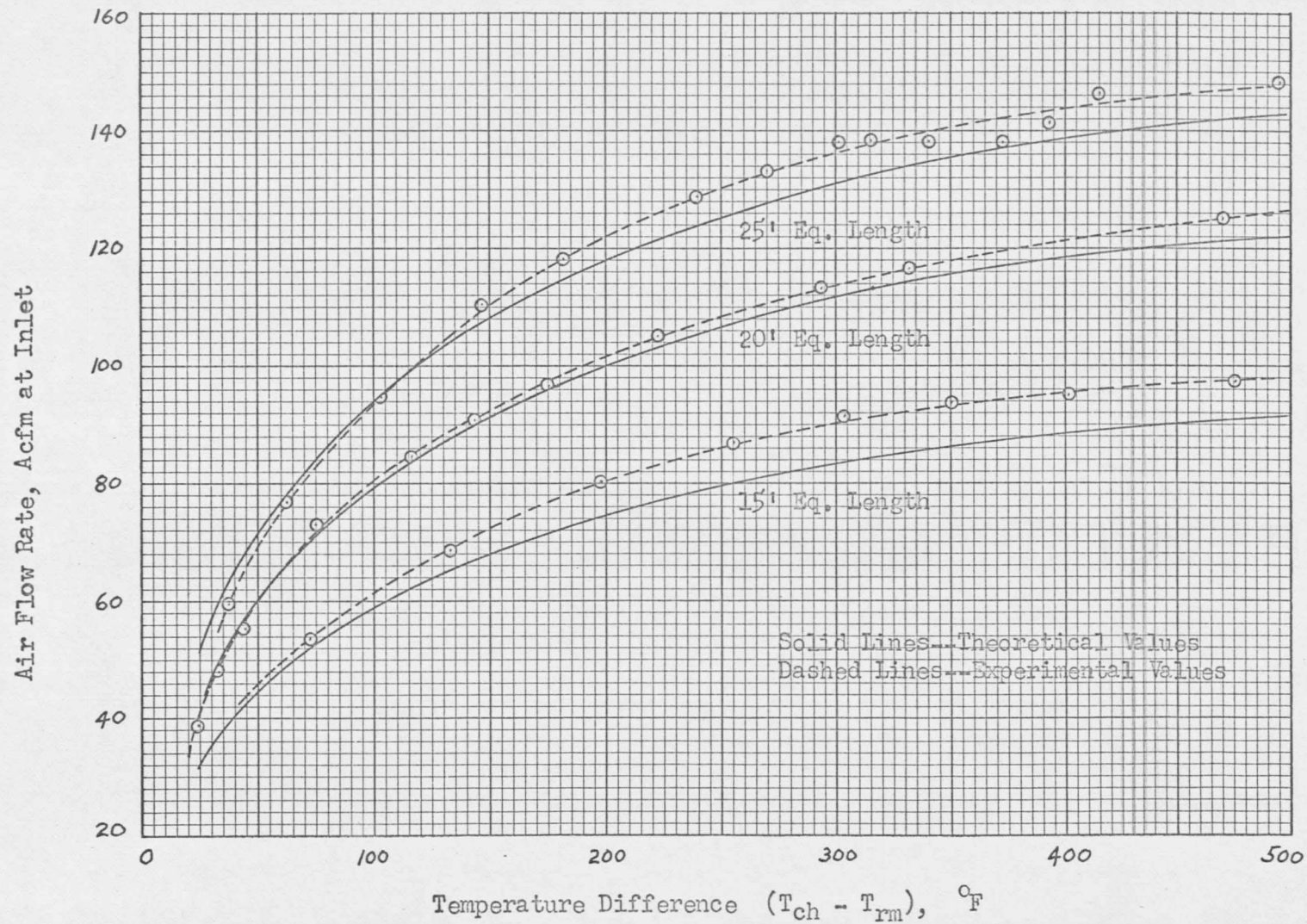


Figure 13. Air Flow vs. Temperature Difference for 8" Diameter Stack
Closed Door Tests -- Elbowed Stack

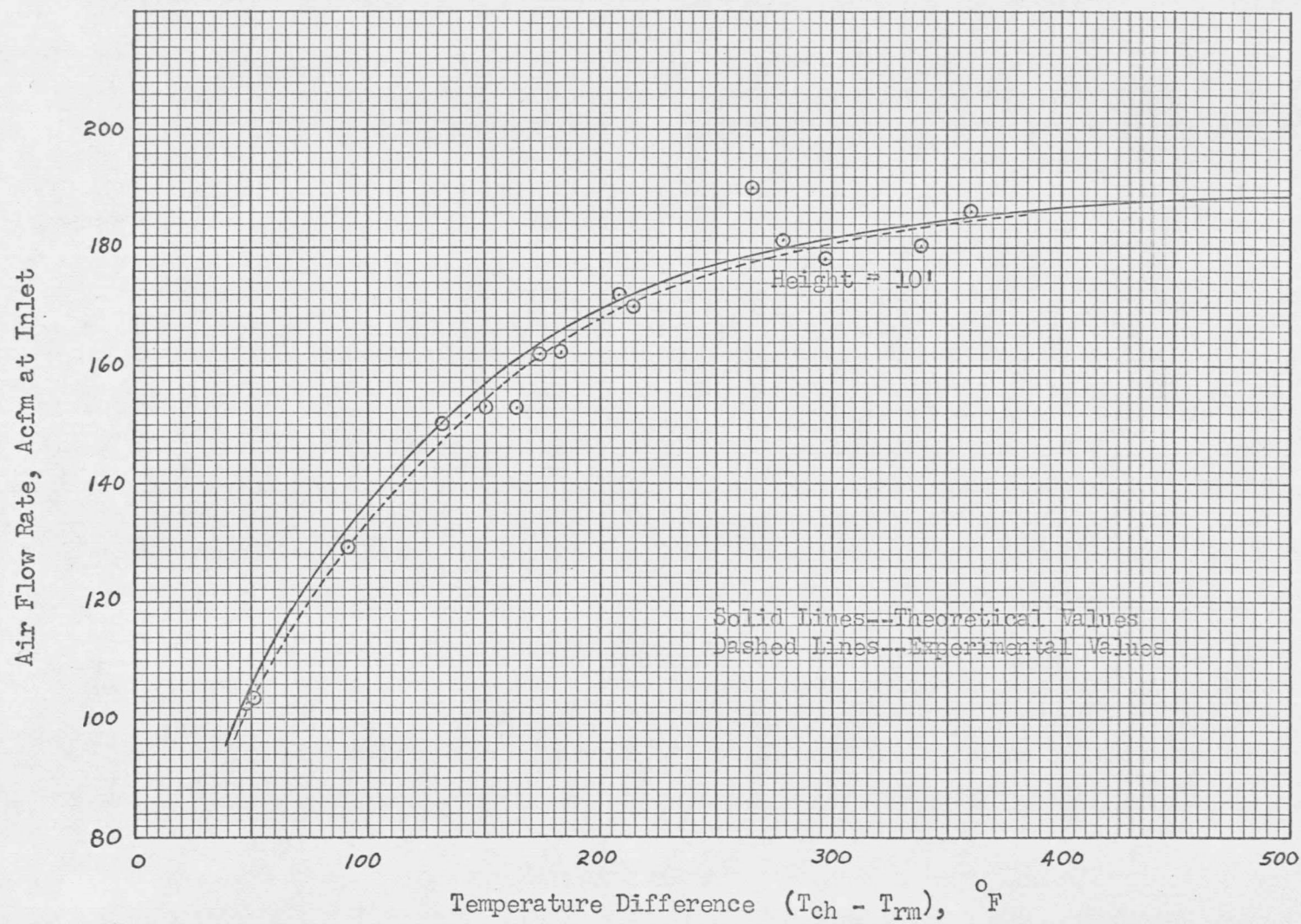


Figure 14. Air Flow vs. Temperature Difference for 8" Diameter Stack.
Open Door Test -- Straight Stack

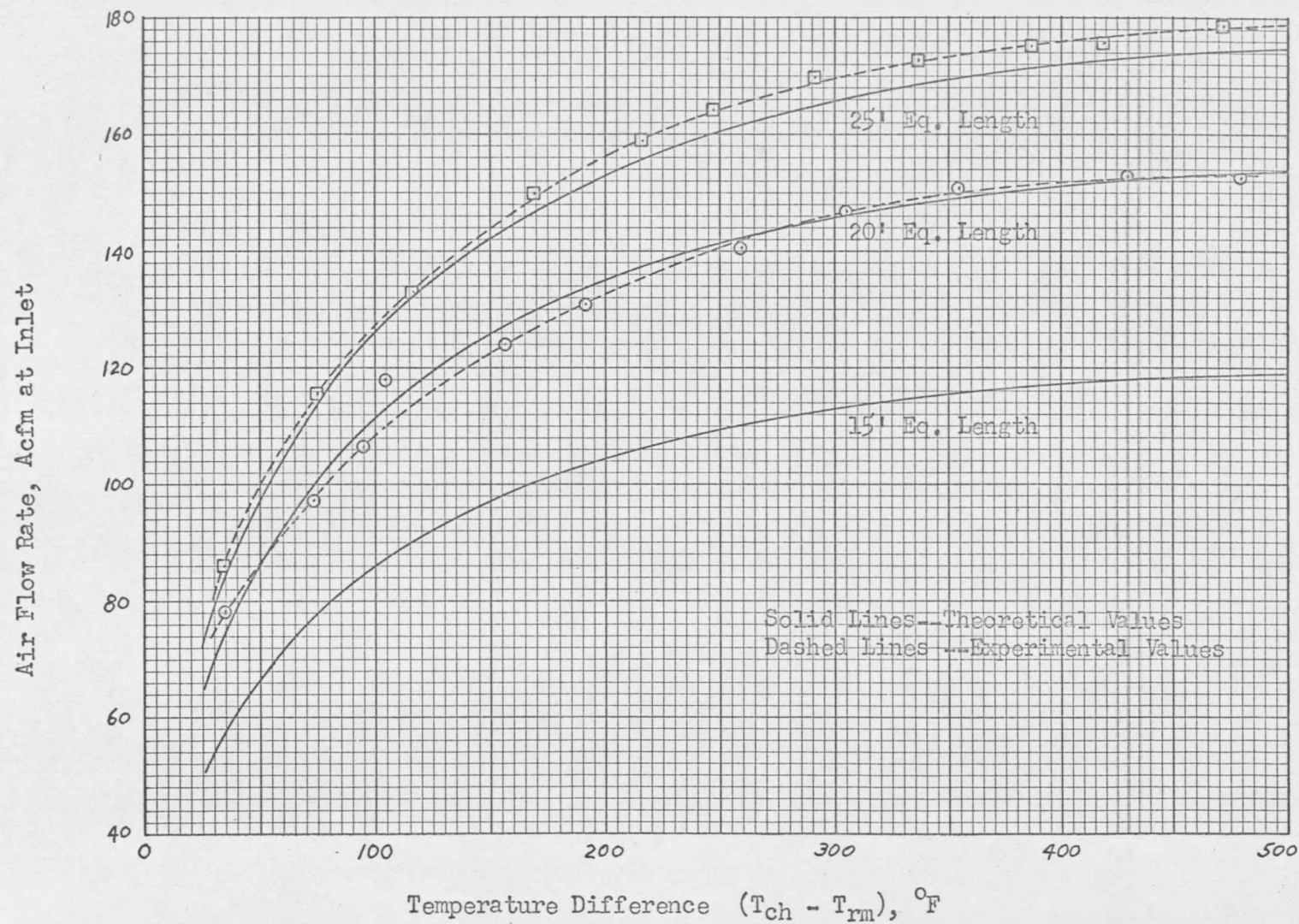


Figure 15. Air Flow vs. Temperature Difference for 8" Diameter Stack
Open Door Tests -- Elbowed Stack

tolerable limits. There are several wild points on the graph in Figure 14, especially at flows above 150 acfm. The erratic behavior of the pitot tube pressure readings is attributed to the fact that the pitot tube in the stack was only seven feet directly above the flame or point where the air velocity was being established and thus, at high flows, was in the path of very turbulent air. It is seen that with the elbowed arrangement, Figure 15, the points again follow a smooth curve above 150 acfm air flow. The elbows and short horizontal length of stack tend to straighten the flow by the time it reaches the pitot tube.

Figure 16 on page 52 illustrates the effect of inlet restriction given by an 8 in. diameter intake duct on air flow in 6 and 8 in. stacks of 15 ft height. At a temperature difference of only 100° F, unrestricted flow for the 6 in. stack is reduced by 18 percent by an inlet restriction of only 0.0064 inches of water. Under the same conditions, unrestricted flow for the 8 in. stack is reduced by 32 percent by an inlet restriction of 0.0145 in. of water. It appears from this comparison that, in an actual shelter situation, if an intake duct is to be used, its cross-sectional area should be at least double that of the stack area in order to nearly eliminate any effect from inlet restriction. This "double the area" rule should be used as a rule-of-thumb for estimating purposes only.

To further investigate the effects of room ΔP in relation to ΔT and air flow, a test was conducted in which a piece of cardboard with several small holes through it was taped over the inlet end of the intake duct. This arrangement simulates an air filter, at least in its effect upon restraining flow. Pressure drop, temperatures, and air flow were

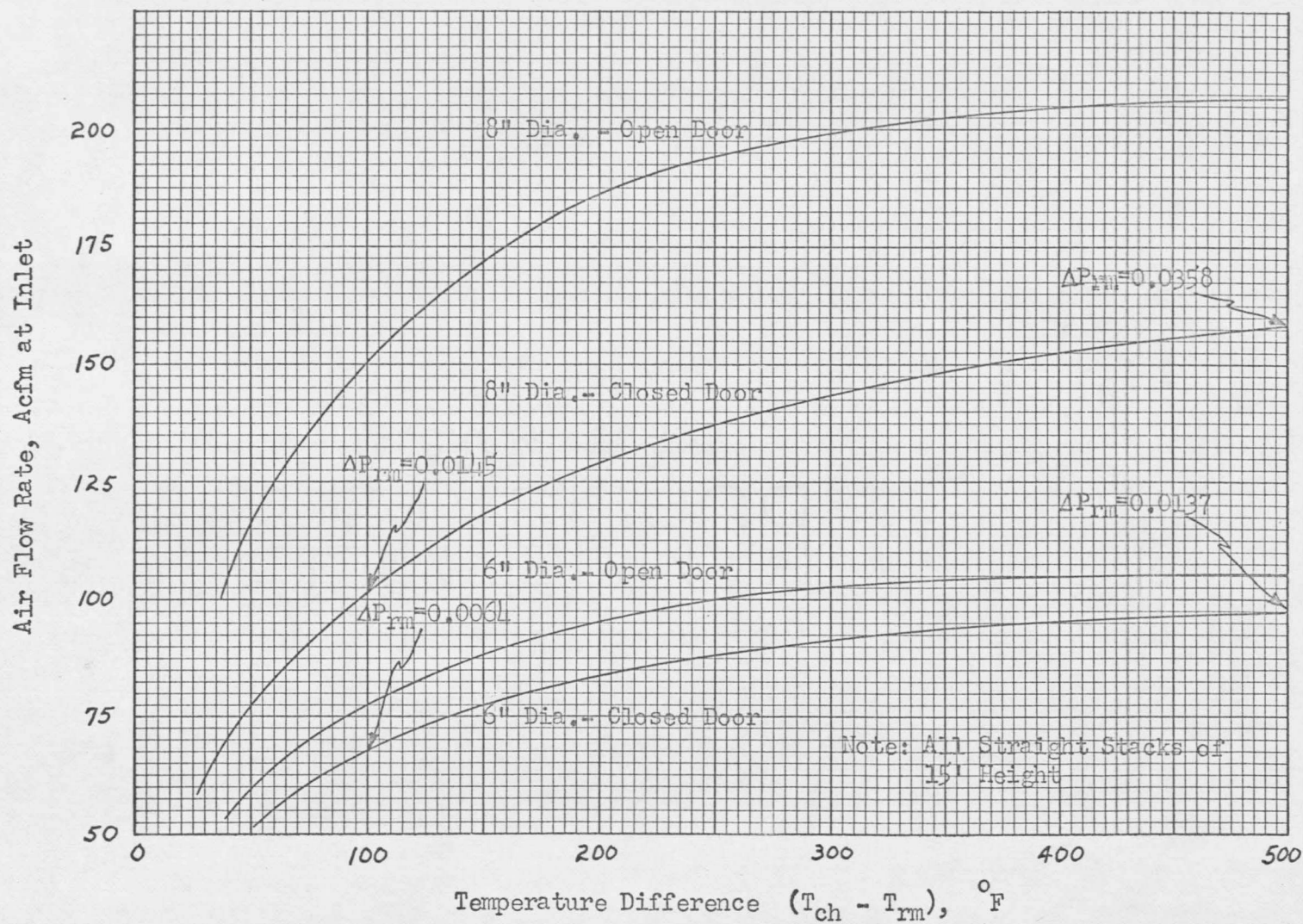


Figure 16. Comparison of Air Flows for Open & Closed Door Conditions to Illustrate Effect of Intake Restriction ΔP_{rm} . Intake Duct -- 8" Diameter

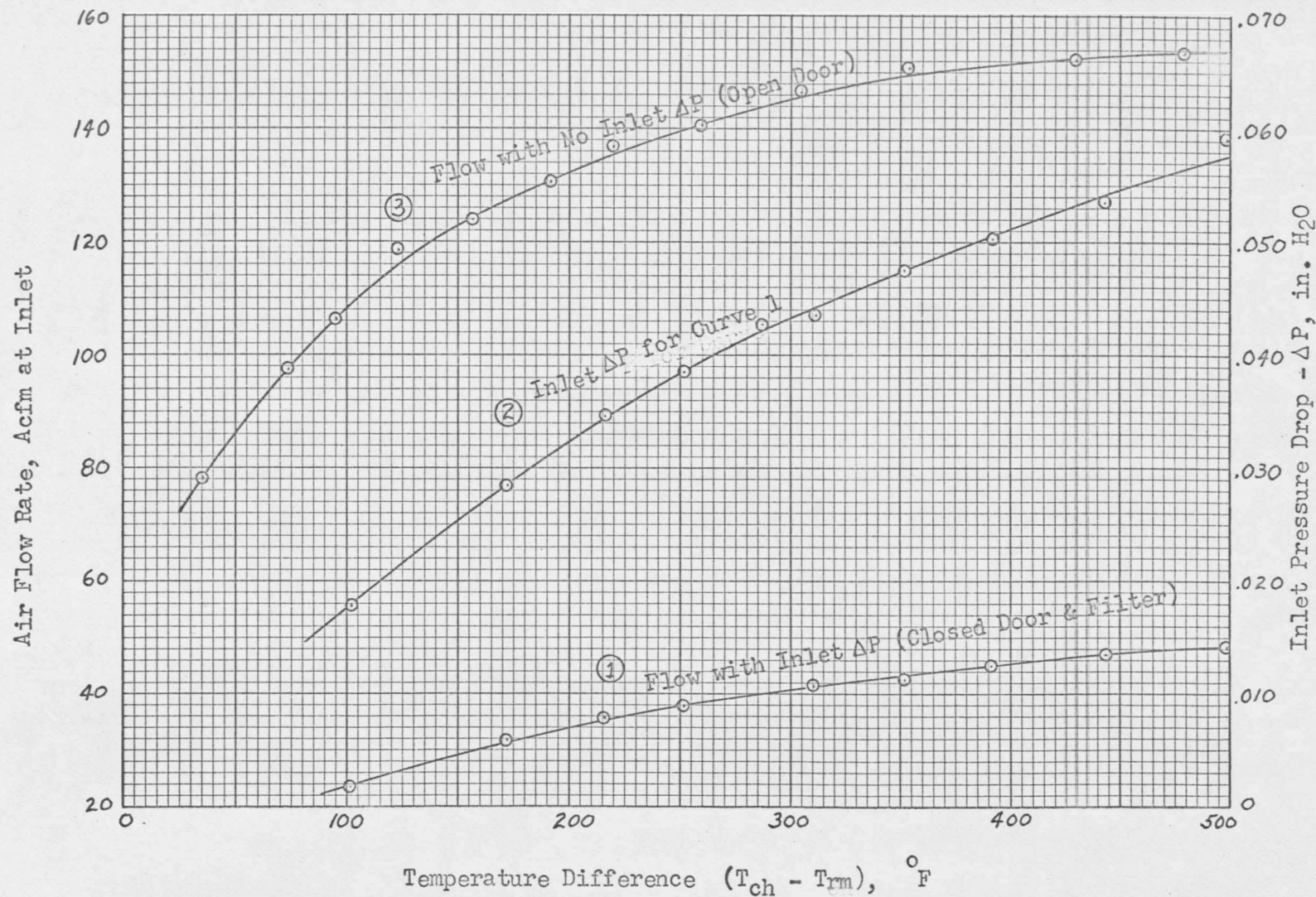


Figure 17. Effect of Inlet Restriction on Rate of Air Flow
8" Diameter - 20' Eq. Length Stack

measured. Results of the test appear in Figure 17. The curves show that a small pressure drop causes a severe reduction in flow from that of the unrestricted condition. At a ΔT of 500° F, for example, air flow is 154 acfm with an open door and 48 acfm with the restricted condition. The diminution in flow, almost 70 percent at a ΔT of 500° F, is caused by a ΔP_m of only 0.0577 inches of water. This indicates that filters cannot be used with the induced draft method of shelter ventilation because there are few, if any, filters available that offer less than 0.1 inch of water restriction, especially at the air flow rates being considered.

Curves on the following four pages give theoretical ventilation rates from computer calculations for 4, 6, 8, and 10 in. diameter stacks for vertical heights of 5, 10, 15, 20, 25, and 30 ft. Because the use of filters is not feasible, because the intake area must be large in relation to stack area for effective ventilation, and because stacks should have elbows and a short horizontal length for fallout radiation elimination purposes, Figures 18 through 21 include results only for open door elbowed stack conditions.

An inspection of the curves reveals that there is little increase in air flow beyond temperature differences of about 300 F. If 300 F is arbitrarily chosen as the maximum practicable temperature difference, shelter ventilation rates ranging from 20 to about 350 cfm or higher can be obtained, depending upon the choice of stack height and diameter.

The effect of altitude on air flow for any given temperature difference is negligible as shown in Figure 22. Values for these three sets of curves were taken from the computer results. Actual conditions for the

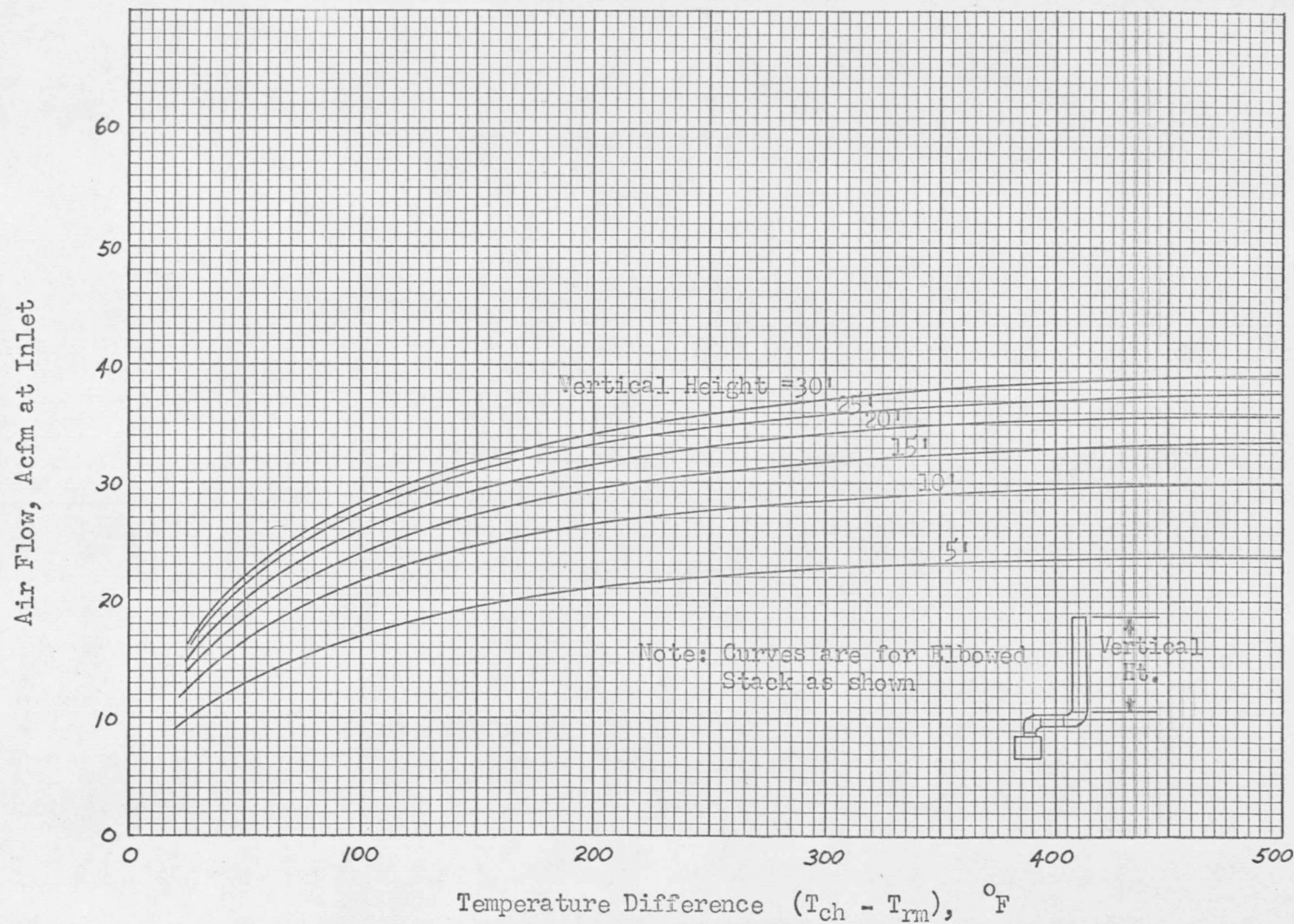


Figure 18. Air Flow vs. Temperature Difference for 4" Diameter Stack--Open Door

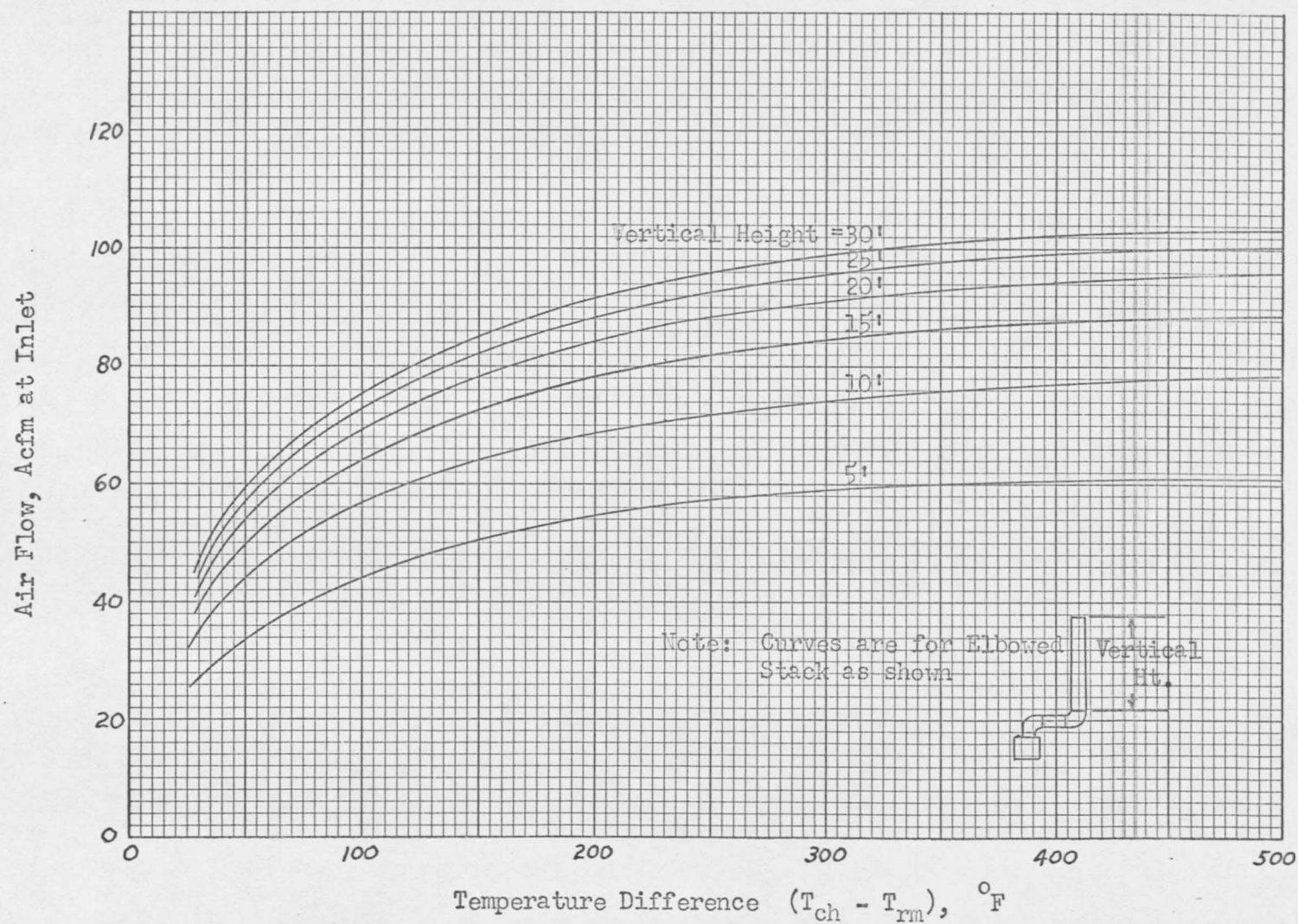


Figure 19. Air Flow vs. Temperature Difference for 6" Diameter Stack--Open Door

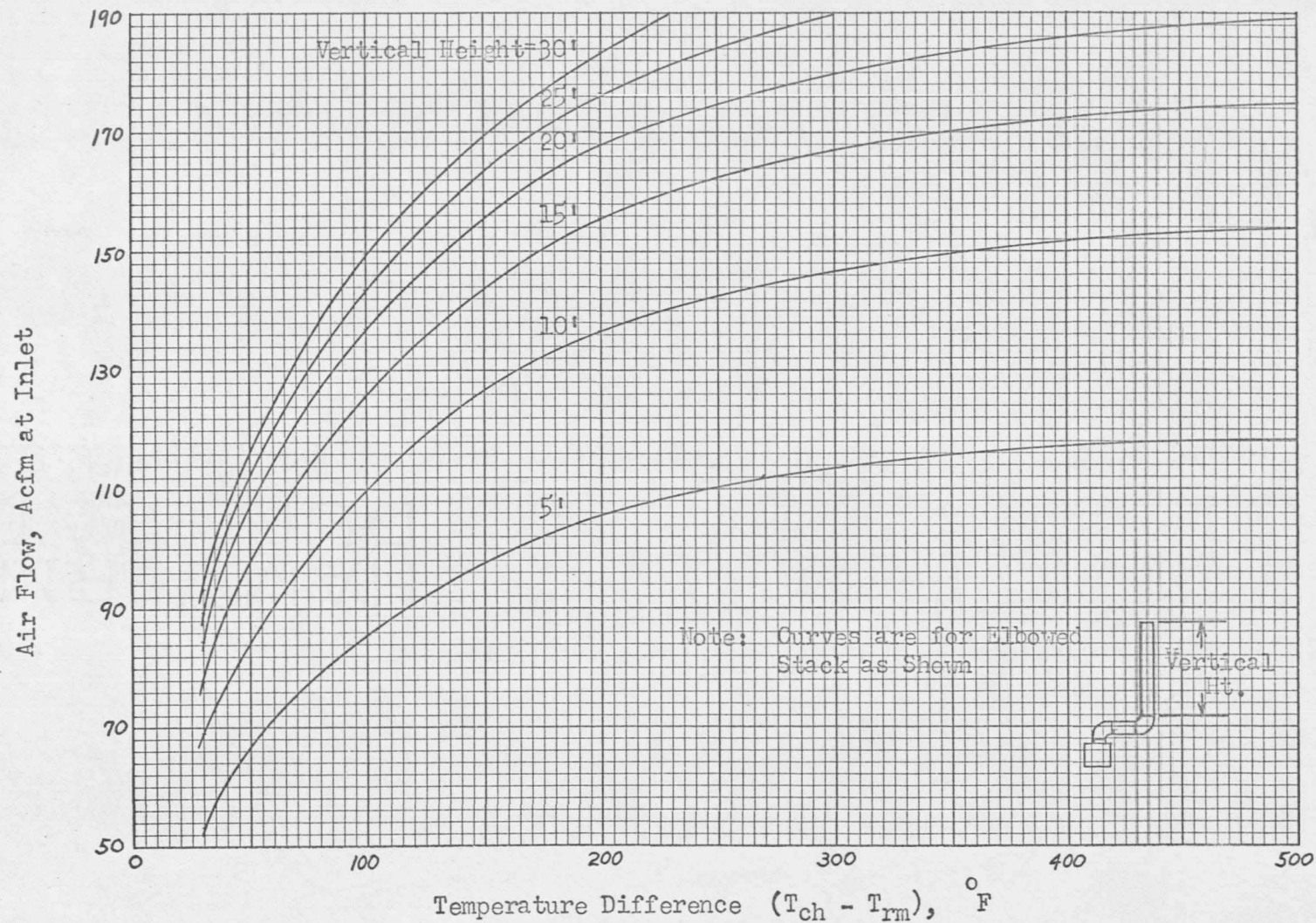


Figure 20. Air Flow vs. Temperature Difference for 8" Diameter Stack--Open Door

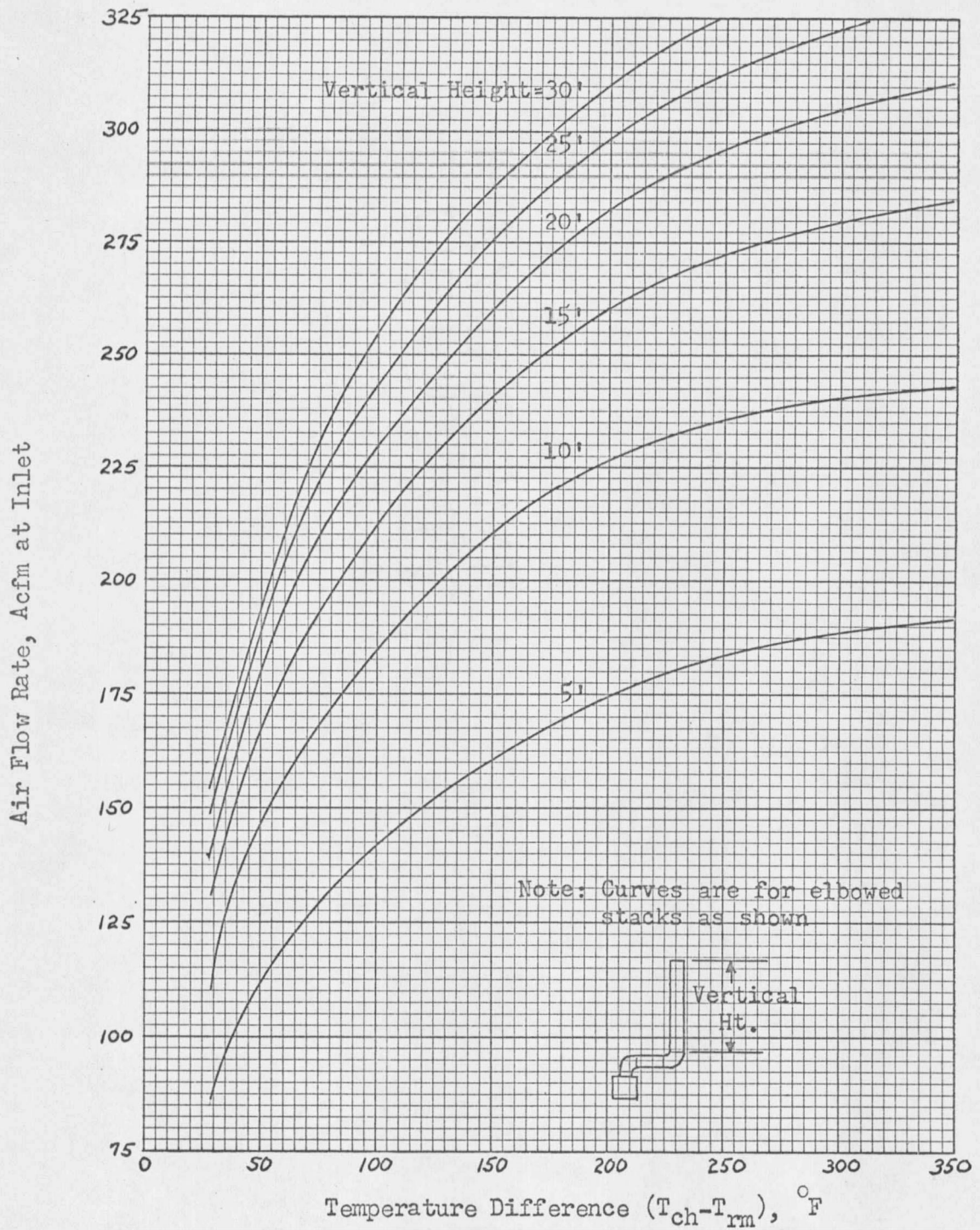


Figure 21. Air Flow vs. Temperature Difference for 10" Diameter Stack -- Open Door

Air Flow, Acfm at Inlet

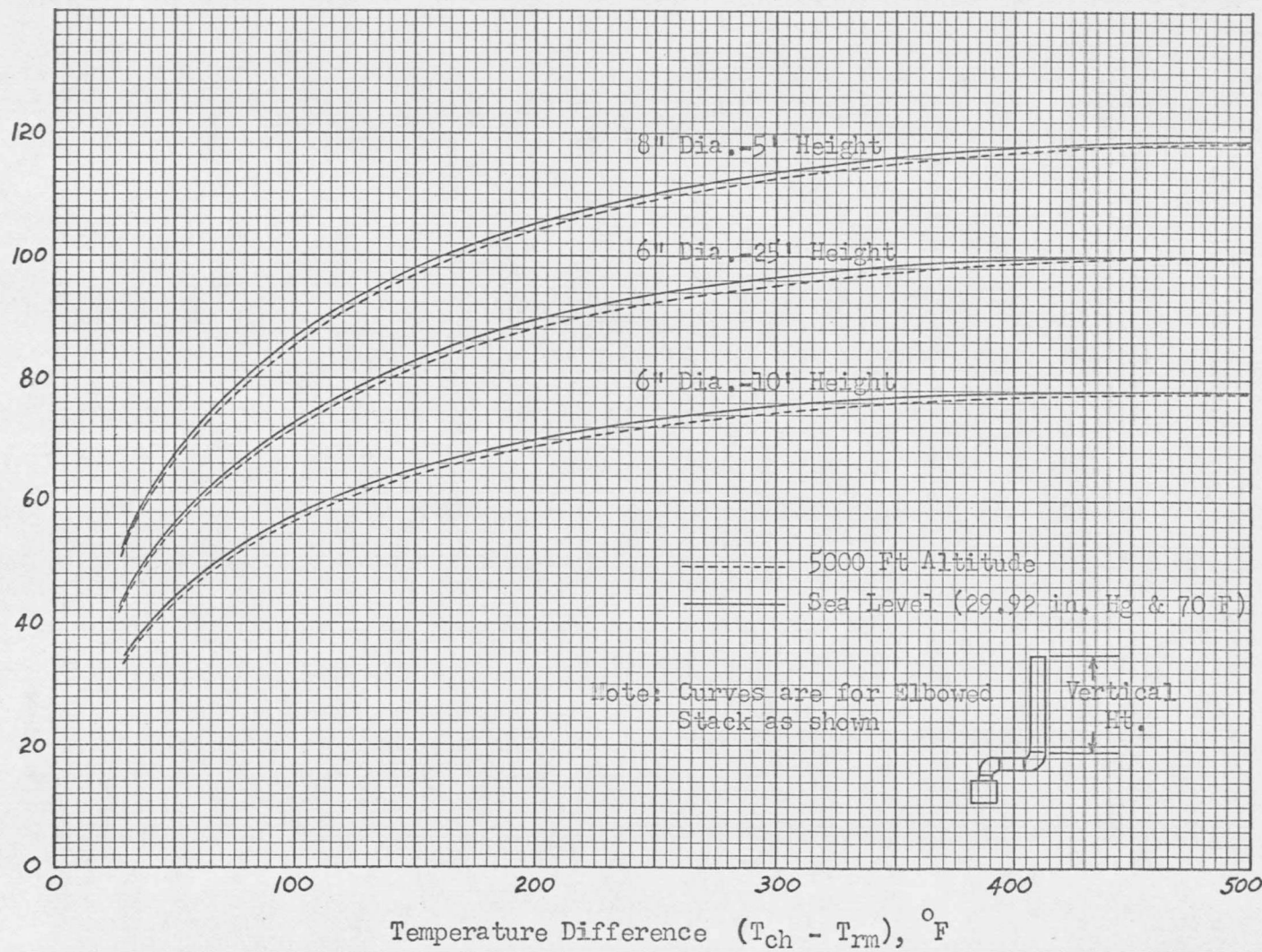


Figure 22. Effect of Altitude on Air Flow

5000 ft altitude curves varied as follows: barometric pressure--24.59 to 25.22 in. Hg; inlet temperature--72 to 78 F; and ambient temperature around stack--76 to 82 F. The sea level curves are for the standard conditions of 29.92 in. Hg barometer and 70 F temperature.

The decrease of standard air flow at 5000 ft altitude is less than 1.4 percent for any given temperature difference. This means that negligible error is introduced when air flow-temperature difference values from the curves are used for altitudes of zero to 5000 ft.

An analysis of the fuel data taken during testing is presented in graphical form in Figure 23 on page 61. Although the data presented in the graph is for a closed door arrangement, which was ruled out in previous discussion, the fact that there can be fuel savings by the use of insulation was pointed out by this test.

The two curves for 6 in. stacks illustrate the effectiveness of insulation placed on the hood, elbows, and horizontal length of stack. For any given air flow covered by the two curves, the fuel (kerosene) consumption for the uninsulated stack is decreased by about 15 percent by the addition of 1 in. of fiber glass insulation. This percentage decrease in fuel rate is seen to apply over the entire flow range for the 6 in. stack condition described. Over an extended shelter occupancy, this saving could mean a considerable amount, not only in volume of fuel, but also in cost. Also seen from Figure 23 is the relative effectiveness of the "three-holer" lamp and the dual space-heater burner.

The curves in Figure 24 illustrate the effect on air flow of two types of ventilator stack caps. The Artis cap, from which air emerges

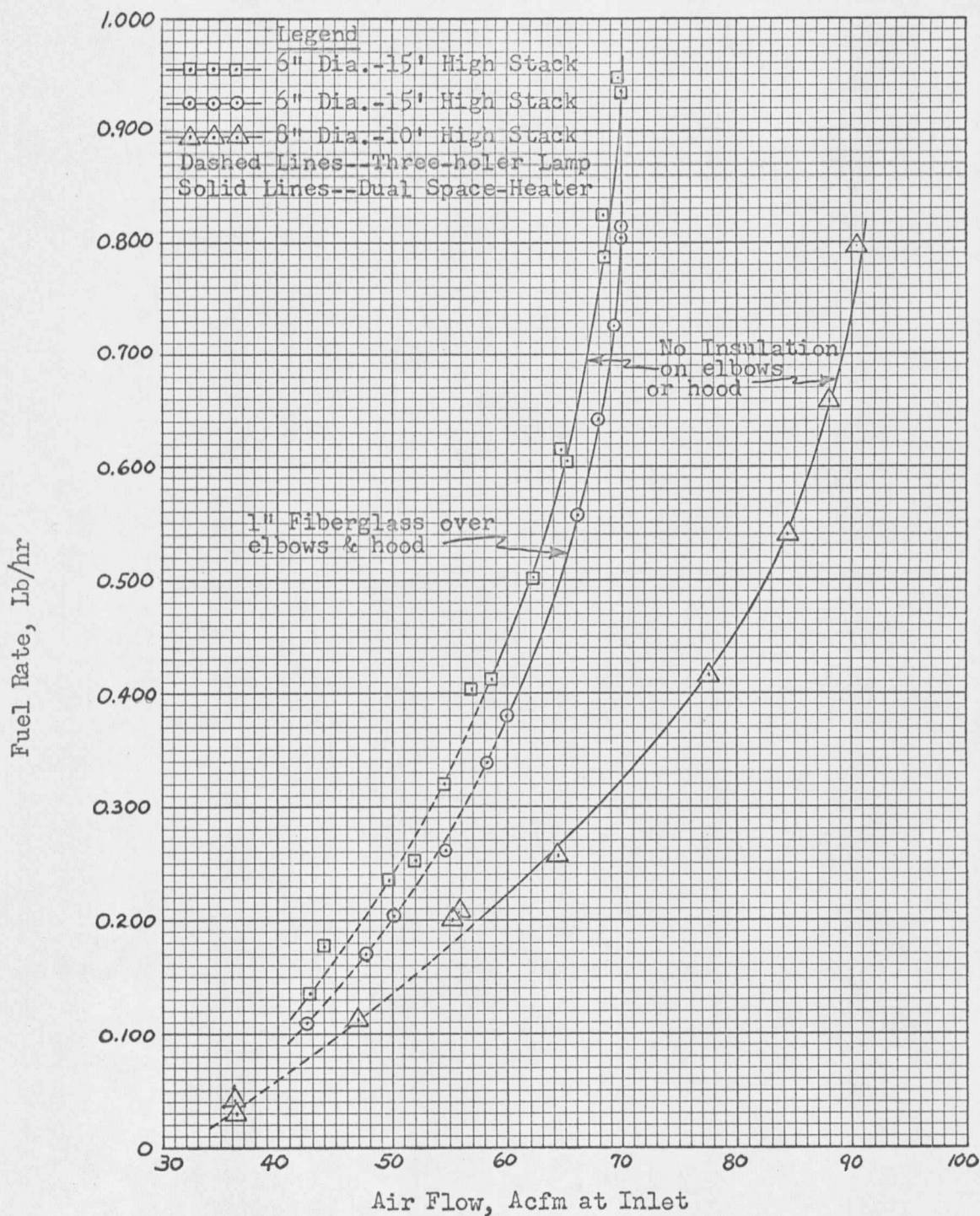


Figure 23. Fuel Rate vs. Air Flow -- Closed Door Test

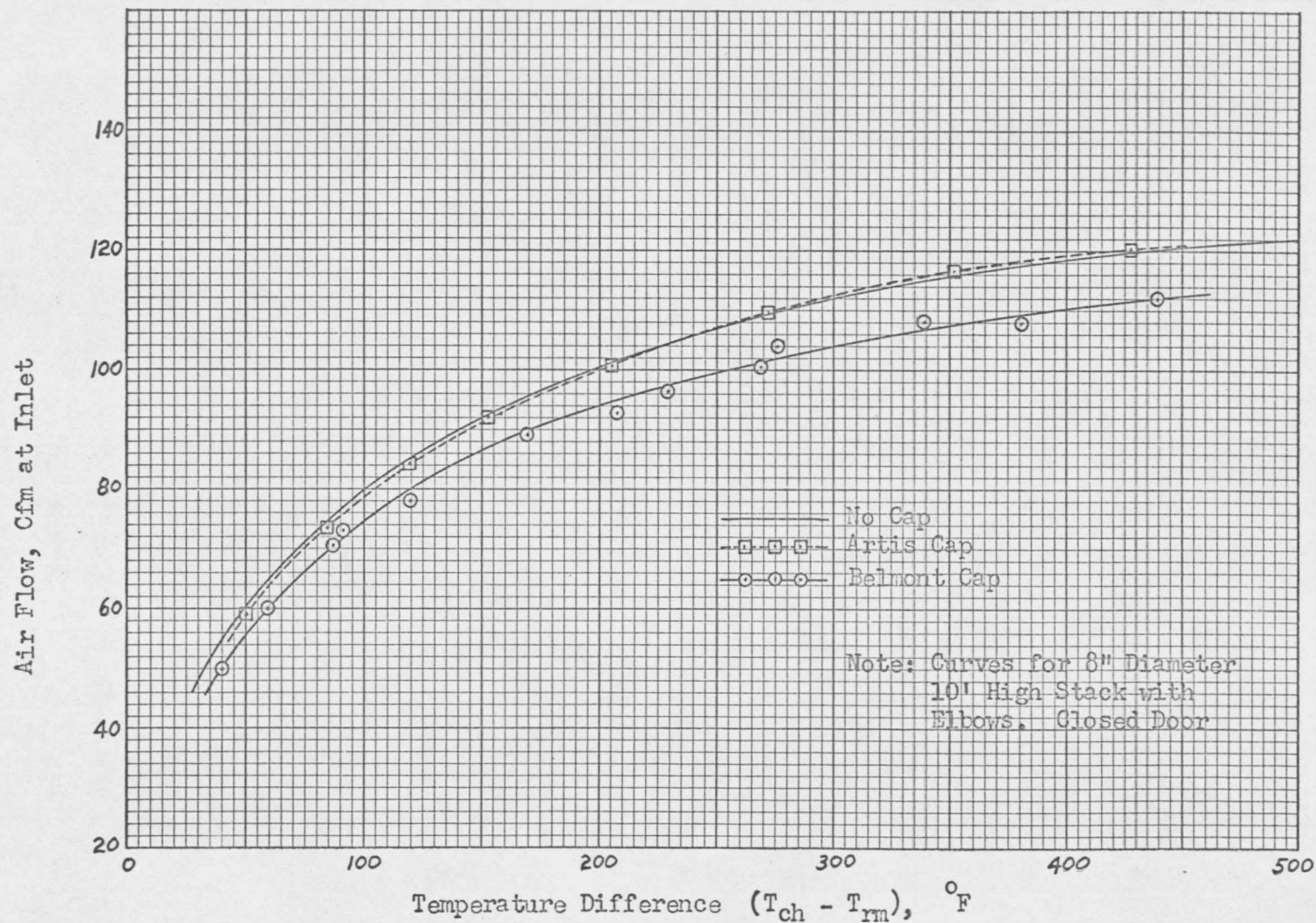


Figure 24. Effect of Ventilator Caps on Air Flow

horizontally at two sides only, does not produce any noticeable change in air flow from that of the uncapped-stack condition. With the Belmont stack cap, air emerges from all sides in a downward direction, i.e., air flowing up the stack must reverse direction in order to leave the cap. This motion reduces the uncapped-stack flow by about 8 percent as shown in the curve. This flow resistance amounts to approximately 5.5 ft equivalent stack length as calculated from equation (11). Similar results were obtained from a straight stack test (not shown).

It should be remembered that the ventilator cap tests were conducted in the absence of wind. In actual shelter applications where wind conditions are present, the use of a Belmont cap or a type L Breidert Air-X-Hauster cap (built specifically for fallout shelters) will have less dampening effect on air flow than that shown by the curves. These caps are built so that the outflow of air, regardless of the wind direction, is actually increased by an aspirating effect of the wind, i.e., the wind produces a suction at the cap outlet due to a partial vacuum. Thus, in general, a good ventilator stack cap can be considered to offer little, if any restriction to air flow at a shelter exhaust when used with the induced method of ventilation.

Figure 25 shows the results of a test conducted with different heating equipment. The maximum ventilation rate obtained with each lamp or burner combination is given versus temperature difference between inside and outside of the stack. With this 6 in. diameter 15 ft high elbowed stack arrangement, the maximum air flow with the "three-holer" kerosene lamp was about 59 cfm. Maximum flow with the dual space-heater unit was about

Air Flow, cfm at Inlet

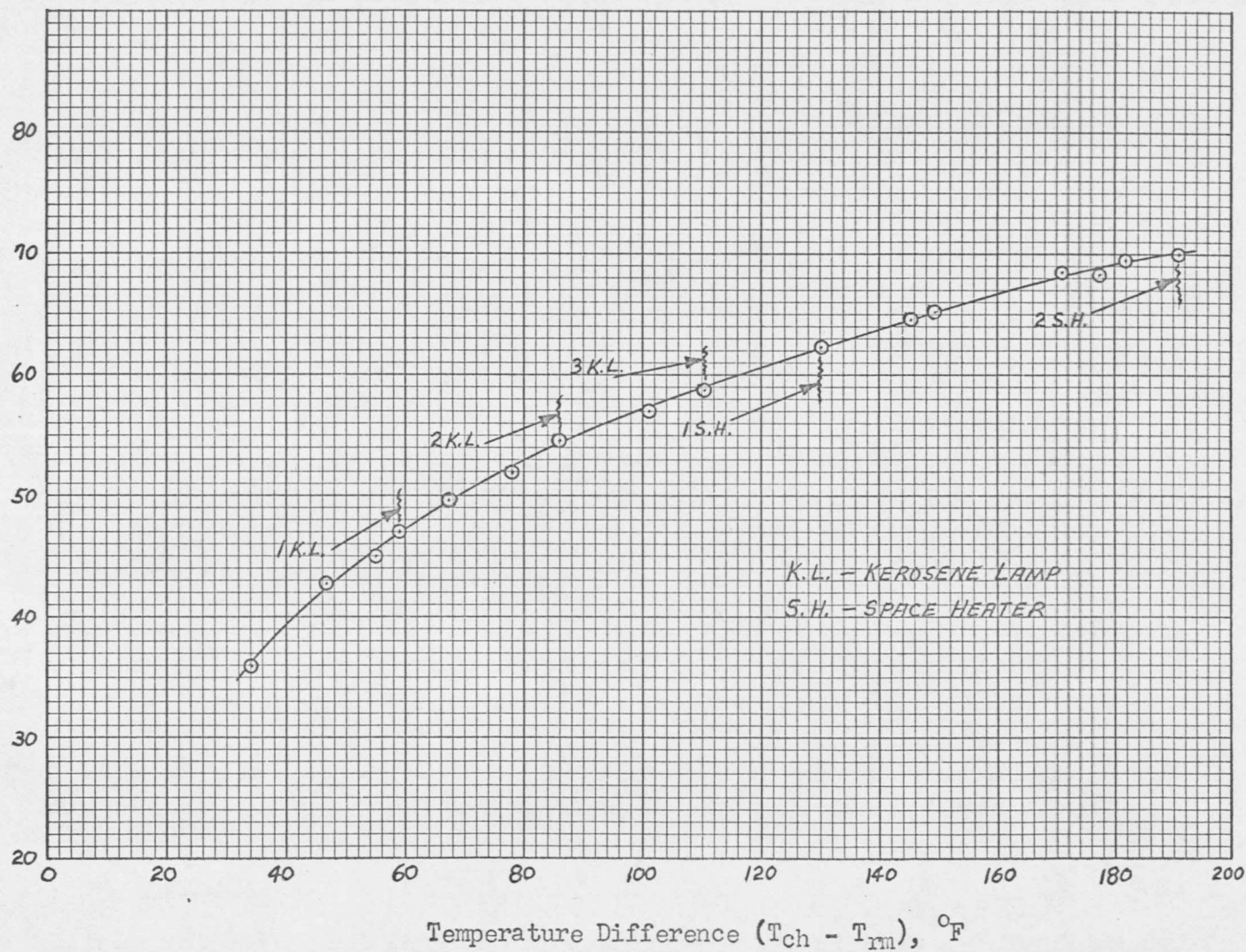


Figure 25. Air Flow vs. Temperature Difference for Different Heating Methods. Closed Door Test with a 6" Dia.-15' Elbowed Stack

70 cfm. Although this test was made with a closed door, it revealed the relative usefulness of each of the heating methods.

CHAPTER 9

EXPLANATION OF PHOTOGRAPHS

Photographs of much of the basic equipment used in the fallout shelter analysis are grouped and presented on the following pages. For detailed descriptions, one is referred to previous chapters which relate to the specific type of equipment shown.

Figure 26 illustrates the manner in which the Metalbestos stacks are taken out of the shelter through the shelter ceiling (power laboratory floor). Two thermocouple probes can be seen protruding from the stack wall. The short cylinder at the bottom and to the right of the stack is the duct to which the laminar flow element was connected when the air anemometers were calibrated. The old engine and other equipment in the background are not part of the test apparatus.

Figure 27 shows the "three-holer" kerosene lamp with the fuel tank that is common to all lamps.

The air intake duct arrangement can be seen in Figure 28 as it appears from the inside of the shelter. Only the center duct was used because of the methods of air flow measurement that were employed. Upon close surveillance of the center duct, a thermocouple probe with its electrical lead wires is seen protruding from the top side near the duct center. The tube extending across the duct is an unused pressure tap. Also seen in this photograph are parts of the shelter door and partition.

The micromanometer for measuring stack and intake pressure drops is shown in Figure 29. The two Tygon tubes from the meter are shown extending upward toward a pressure manifold (not shown).

Figure 30 shows the propeller or vane-type anemometer in its position

at the outlet end of the intake duct.

The dual space-heater unit is pictured in Figure 31 as it was used during a ventilation test. Short individual heater chimneys are seen extending into the bottom of the stack intake hood. Also seen in this picture are a scale balance and the burner support stand.

Figure 32 shows the elbowed stack configuration and the manner in which it is fastened within the shelter. Another view of the entrance hood is also shown.

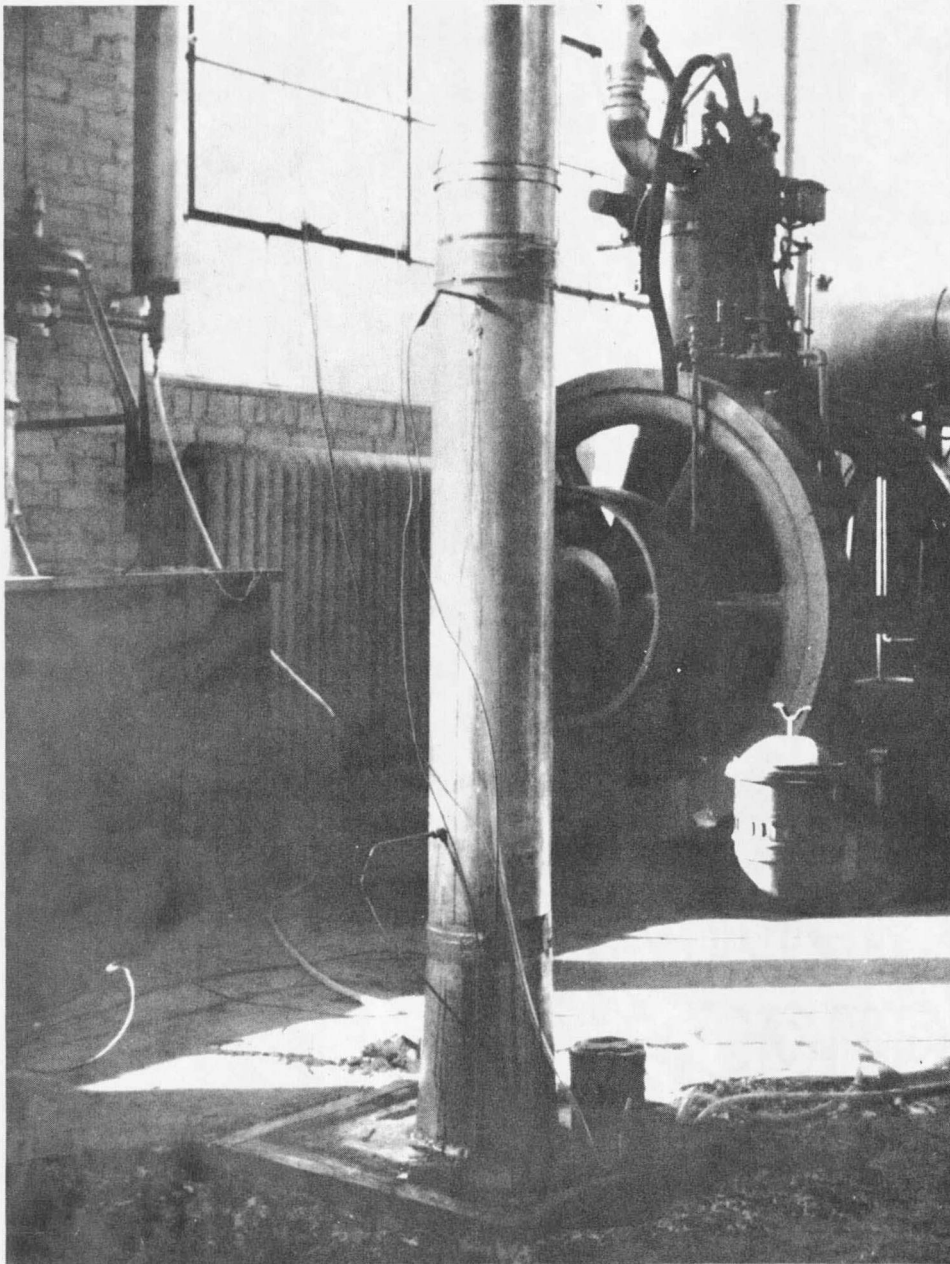


Figure 26. Stack Extending from Shelter

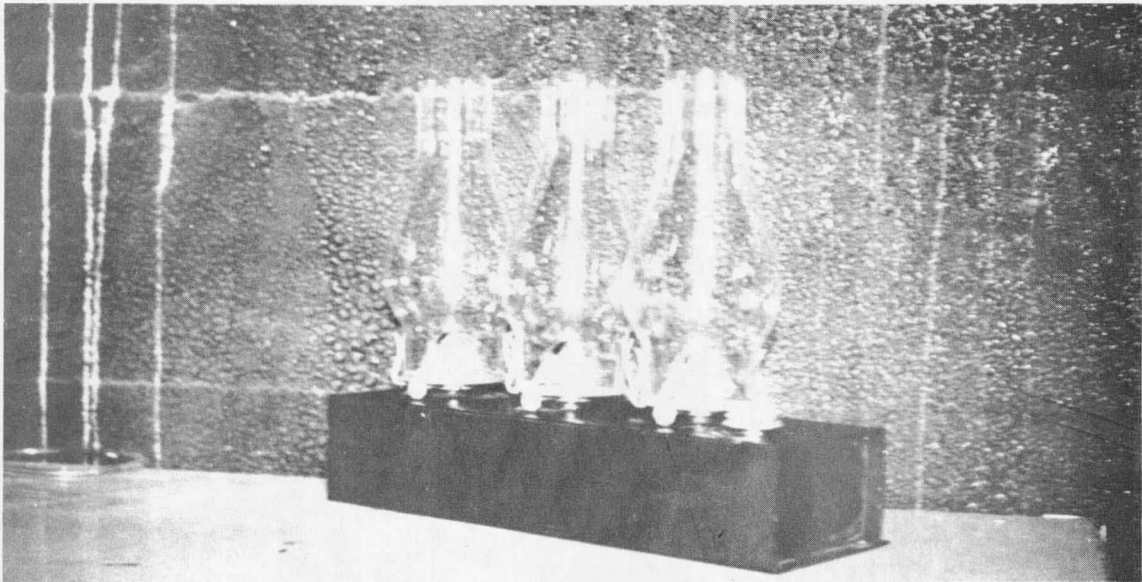


Figure 27. "Three-Holer" Kerosene Lamp

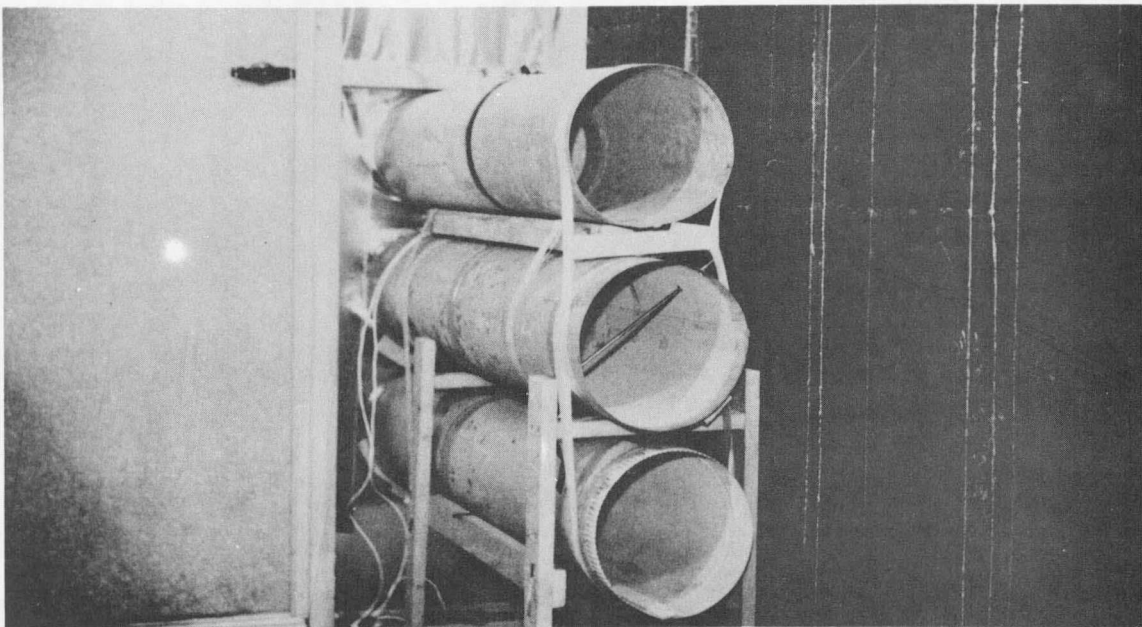


Figure 28. Shelter Air Intake Ducts

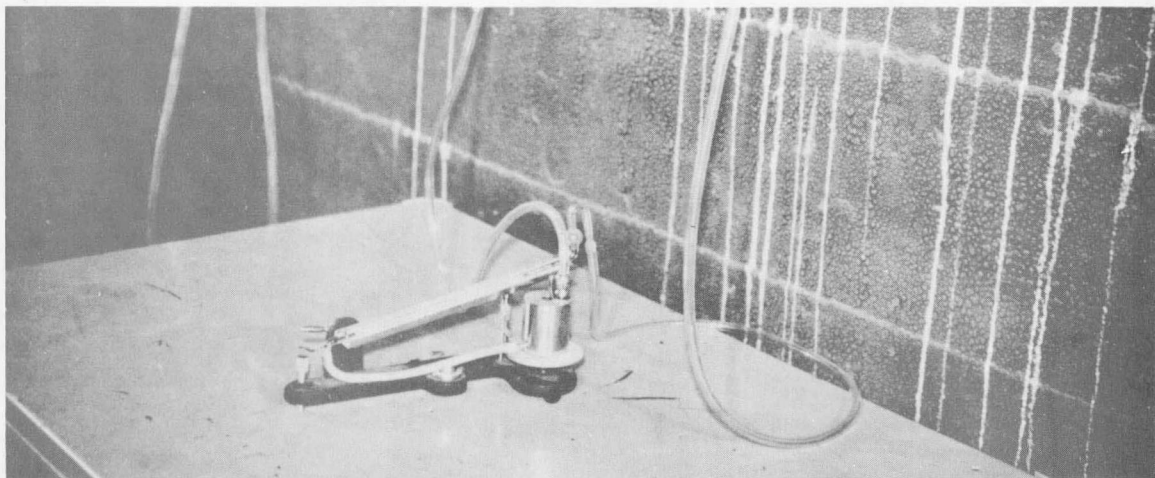


Figure 29. Pressure Measuring Device (Micromanometer)

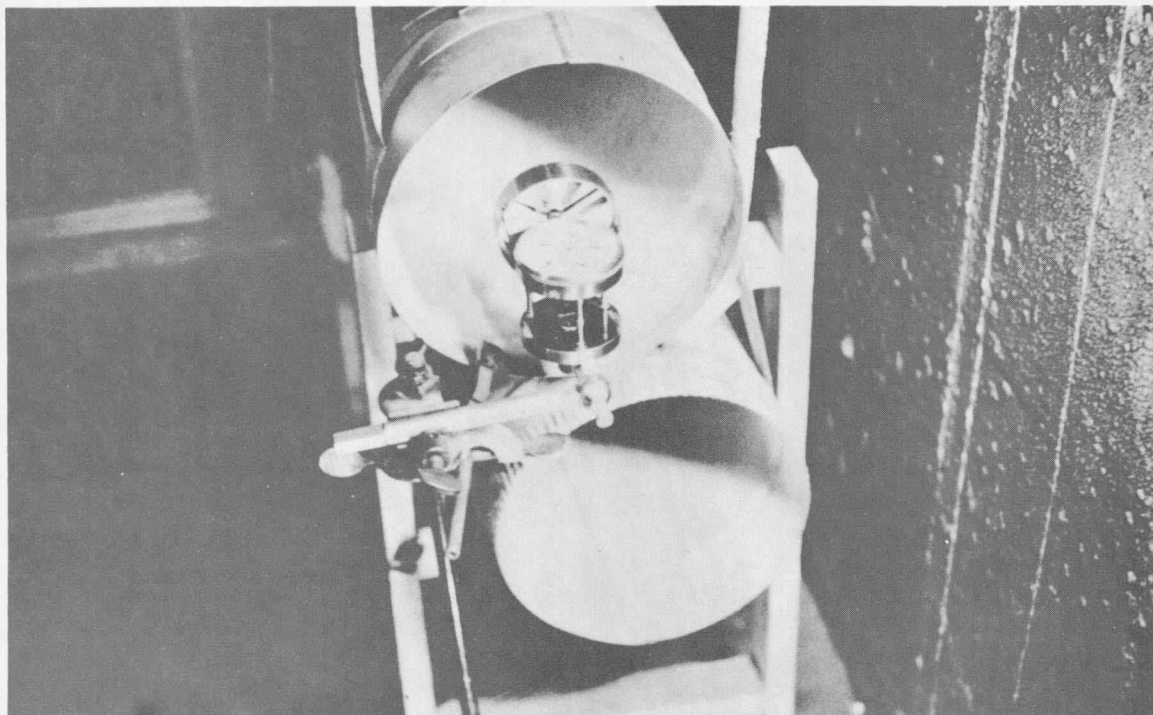


Figure 30. Propeller-Type Anemometer

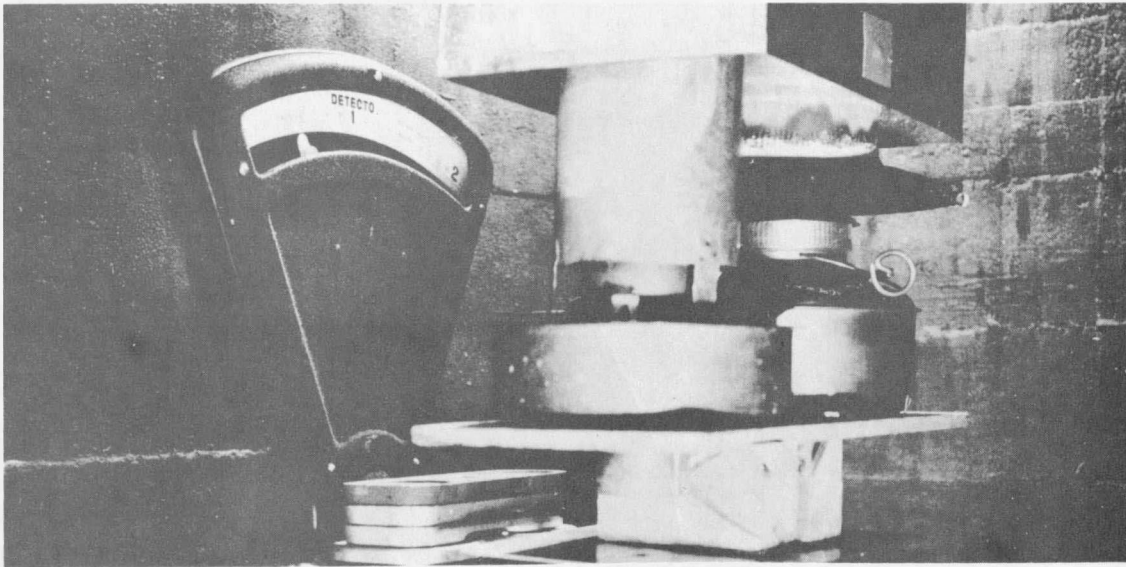


Figure 31. Dual Space-Heater Equipment

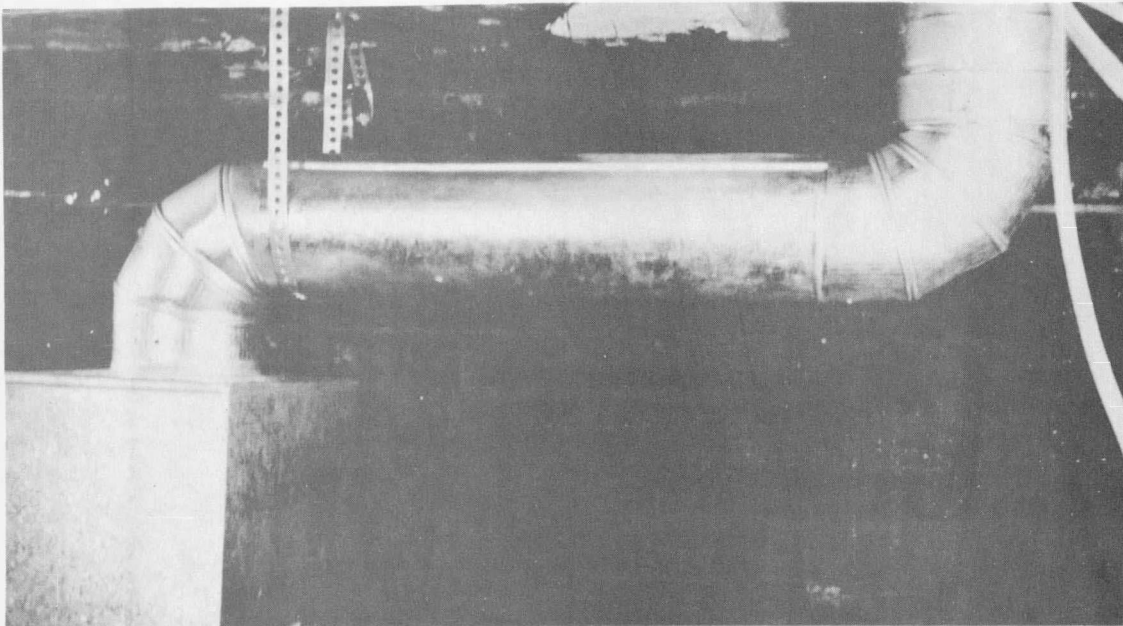


Figure 32. Elbowed Stack Configuration and Entrance Hood

CHAPTER 10

CONCLUSIONS AND RECOMMENDATIONS

As stated previously, the fallout shelter tested has an occupant capacity of 18 people and thus requires, as a bare minimum, at least 54 cfm of fresh ventilating air. It is concluded from the research work performed that this minimum ventilation rate can be obtained by burning kerosene in the base of a chimney or stack. One set of conditions which satisfies the above requirements is: stack--6 in. diameter, 15 ft height, two elbows, a 2 ft horizontal section, and a hood; shelter intake--essentially unrestricted or open door; temperature difference between inside and outside of stack--approximately 80 F (mean stack temperature of about 150 F); heat generator--either the "three-holer" kerosene lamp or the dual space-heater unit; and fuel rate--about 0.30 lb kerosene per hour (equivalent to 1.05 gallons per 24 hour day). It should be kept in mind that there are many possible combinations of stack parameters, temperatures, fuel rates, etc. that will produce this same ventilation rate.

Results of this ventilation investigation indicate that the maximum air temperature difference between the inside and outside of the stack ($T_{ch} - T_{rm}$) that is of benefit for home shelters is around 300 F. With an ambient temperature of 70 F, this means that the maximum mean stack temperature of benefit is about 370 F. The return for higher stack gas temperatures is not significant.

The highest stack temperatures obtainable with the kerosene wick-type burners used in the test are about 270 F. It is feasible that a wick-type burner could be designed to produce a maximum stack temperature of 370 F. Wick-type burners are desirable because they require no pressure drop for

efficient combustion and they can be used to light the shelter.

The lowest fuel consumption rate is obtainable when the intake hood and the portion of the stack that is within the shelter is insulated. A minimum of one inch of common fiberglass insulation is recommended.

Family-type shelters in general can be adequately ventilated by the induced draft method if certain qualifications are observed. The shelter should be located in the basement of the house so that air can be drawn from the main part of the house. In this manner the house will serve as the radiation-particle filter for the shelter inlet. Outside air will infiltrate naturally into the house through window cracks, under doors, etc. in sufficient quantities to be available to ventilate the shelter. Filters should not be used at the shelter inlet because they offer a large restriction to air flow and reduce ventilation rates to below minimum values. If an intake duct is used, its cross-sectional area should be at least twice that of the stack. An open door is recommended. The stack should contain two elbows and a short horizontal length of duct. This is essential for providing shielding from fallout radiation rays. A good ventilator stack cap should be used if the stack exhausts directly to the atmosphere to insure that fallout particles, rain, snow, and other debris do not enter the shelter through the stack. If a stack cap is not used, the stack should exhaust to a ventilated attic. The attic will minimize wind effects, eliminate fallout particles from entering stack, and prevent vitiated air from recirculating through the house and shelter.

The induced draft ventilation method creates a slight negative pressure in the shelter. This eliminates the possibility of using this system

in multiple for community-type shelters. Civil Defense authorities have stipulated that community shelters will have positive pressure in the shelter.

Although kerosene wick-type burners are recommended for induced draft ventilation, it seems feasible that in an extreme emergency situation any flame from whatever equipment is available--such as a gasoline camp-cook stove--could be used to induce the draft. Ventilation efficiency and safety aspects would be reduced, but perhaps, and most important, human lives could be saved.

There are several facets of family-type shelter ventilation, by the induced draft method, which warrant further investigation. Future research should center around the following objectives:

1. Further investigation of kerosene wick-type burner design with the intent of reaching 370 F stack temperatures.
2. Continuation of fuel investigation including storage, consumption, and safety controls.
3. Economic feasibility of all phases of the induced draft method of shelter ventilation.
4. Construction of a design nomograph relating the parameters of vertical stack height, equivalent stack length, stack diameter, stack temperatures, and cfm of air flow to be used for the design of family fallout shelter induced draft systems.

The last item mentioned, a design nomograph, could conceivably be constructed from the data collected during the research work already performed. Construction of such a nomograph represents more time than was available in this investigation.

APPENDIX

THERMISTOR ANEMOMETRY

Extensive investigation was conducted concerning the possibility of using a bead-type thermistor as an air anemometer. The attempts were unsuccessful and the results useless as far as ventilation was concerned, but the research work was important and is recorded here for posterity.

Although nearly four and one-half months were spent investigating thermistors with no apparent useful results, the author believes that the time was well justified for three principal reasons. First, thermistors, if applicable, would allow measurement of extremely low velocity air flows (starting at zero velocity) as well as high flows. No other method of flow measurement known to the author permits, with sufficient accuracy, measurement of flows with a velocity lower than about 40 ft per min (nearly 15 cfm in an 8 in. duct). Thermistors would have allowed measurement of the natural ventilation through the shelter. Secondly, a permanent record of output voltage and thus of flow would be available with this method. This is desirable because it establishes a reference and affords one the opportunity to recheck measurements later on if the need arises. Thirdly, it is possible that thermistors other than the bead-type and circuits other than the bridge-type may be used successfully in air flow measurements. No more time could be spent on thermistor investigation, but the knowledge gained from this experience should be useful as a starting point in further thermistor research.

Thermistors are "thermal resistors" or resistors with a high negative temperature coefficient of resistance. As the temperature increases, the resistance decreases and as the temperature decreases, the resistance increases. This is opposite to the effect of temperature changes on metals.

Thermistors are semi-conductors of ceramic material made by sintering mixtures of metallic oxides such as manganese, nickel, cobalt, copper, iron, and uranium.

Thermistors of the bead-type were used for the air anemometer. Beads are made by forming small ellipsoids of thermistor material on two fine wires tight and parallel about 0.010 in. apart. The material is sintered at high temperature and the leads become embedded tightly in the beads making good electrical contact inside the thermistors. Beads can be obtained in sizes ranging from 0.006 in. to 0.100 in. in diameter.

For use as an air measuring device, the thermistor was built into probe form. Two small parallel holes were drilled through the length of a two-inch long, 1/2 in. diameter nylon rod which is pointed on one end. The fine wires of the bead thermistor were inserted through the holes in the rod and cemented in place. Only the bead protruded from the pointed end of the nylon rod. After wire leads were soldered to the bead wires, the rod was fastened into the end of a 5 in. long piece of 9/16 in. diameter stiff cardboard cylinder, the bead remaining exposed. The entire probe was stuck through a one-inch diameter rubber cork. When the assembly was inserted into a hole in the side of the 8 inch intake duct, the bead was positioned in the air stream by adjusting the length of probe extending beyond the cork. The probe was held rigidly in place due to a tight fit between the rubber cork and the hole in the side of the intake duct.

Figure No. 33 on page 78 shows the original bridge circuit designed and built for the thermistor air anemometer. The circuit consists

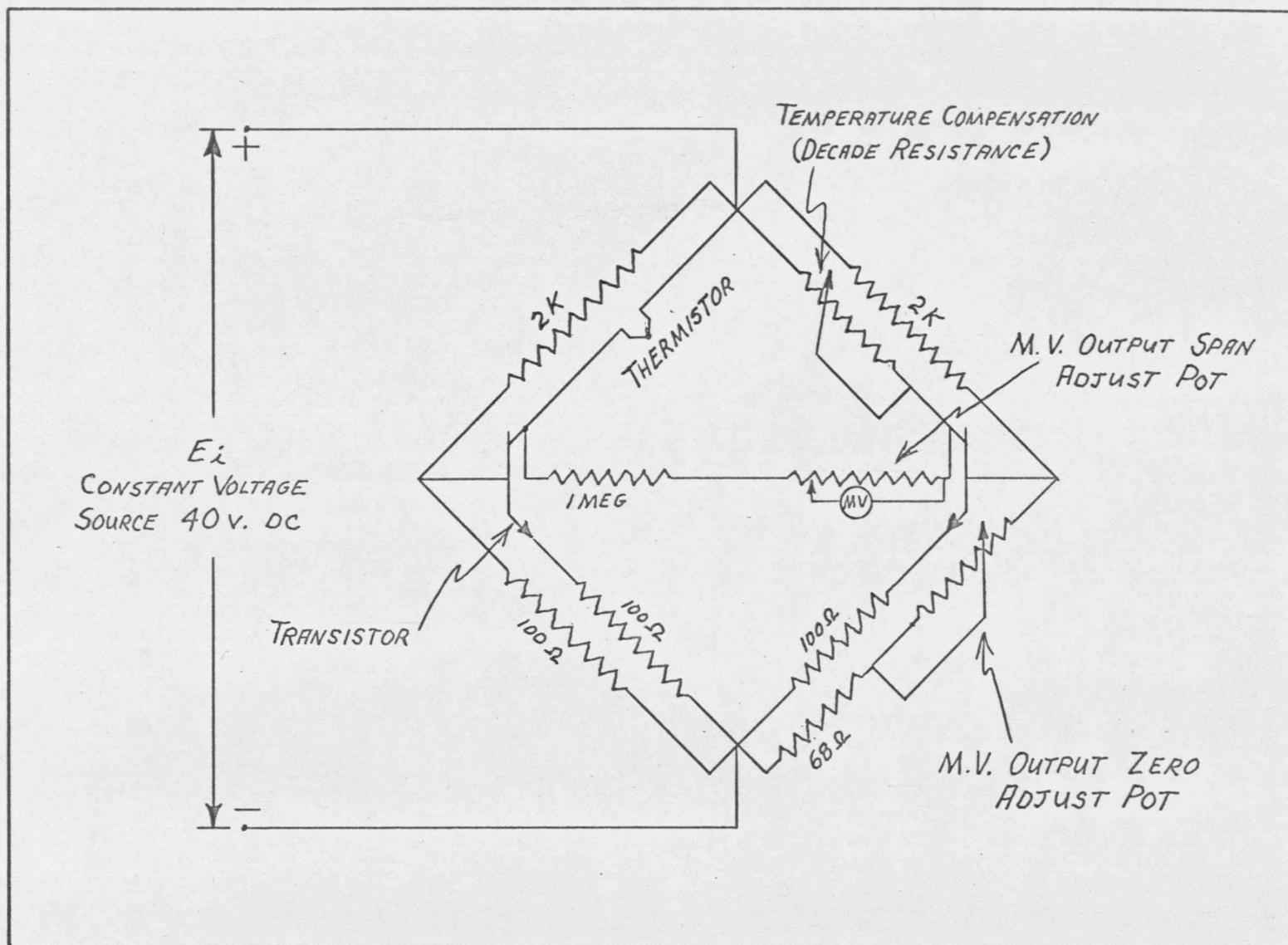


FIGURE 33. THERMISTOR BRIDGE CIRCUIT

basically of a Wheatstone-bridge with the thermistor as one resistance leg. In a leg adjacent to the thermistor is a decade resistance which permits compensating for thermistor bead temperature changes caused by intake air temperature changes. In the leg opposite to the thermistor is an adjustable resistance used to zero the voltage output of the bridge. Two transistors, connected together thermally, are included to provide a constant current through the thermistor when a constant 40 volts is supplied to the bridge circuit.

Output from the thermistor bridge was connected to a Leeds & Northrup Speedomax 0-10 millivolt recorder. Thermistor output is sensitive to air velocity, and because the cubic feet per minute of air flowing in a fixed-size duct depends directly on the velocity, the output can be plotted against air flow rate. Thus, the recorder provided a direct indication and a permanent record of air flow.

During calibration of the thermistor, a laminar flow element, produced by the Meriam Instrument Company, was used with an accompanying centrifugal blower as the air flow standard. This laminar flow device, applicable to air flows from 0 to 200 standard cfm (at 29.92 in. Hg and 70° F), consists of parallel, capillary-size tubes through which the air flows. The pressure drop across the tubes has a straight-line relationship with the mass air flow. A linear calibration curve of this relationship, showing pressure drop across the tubes (in. of water) versus standard air flow (lb per min) is supplied with the element.

Equipment required to operate the thermistor included a 40 volt d.c. power supply, a voltmeter, two decade resistances for use in the bridge

circuit, a millivolt recorder for recording thermistor output, and a thermocouple temperature recorder for determining temperature of air in the vicinity of the thermistor probe. This equipment, with the exception of the temperature recorder, is shown in Figure 34 on page 81 as it was arranged and used in the shelter.

To calibrate the thermistor, the blower and laminar flow element were connected at the exhaust of the shelter so air could be pulled through the intake duct and shelter space. This permitted an accurate calibration because it included any air leaks that may have existed in the shelter. The calibration procedure for the thermistor probe located at the center of the eight-inch intake duct was as follows:

1. With probe in still air (cap ends of the 8 in. duct), set the output zero adjust on the bridge box until the millivolt output on recorder is zero.

2. Note temperature of air in duct, read the milliamps and volts through probe, and calculate by ohms law the resistance to be set on the temperature compensation decade box.

3. Uncap duct and set laminar flow element at a manometer reading which corresponds to the maximum flow desired. Set output span adjust on bridge box until about 10 mv show on recorder. Once output adjust is initially set, do not change.

4. Take readings at various points between zero and maximum flow. At each flow setting repeat step 2 and reset the temperature compensation decade resistance if temperature changes. At each reading record inches of water at manometer, duct temperature, and millivolt output. The

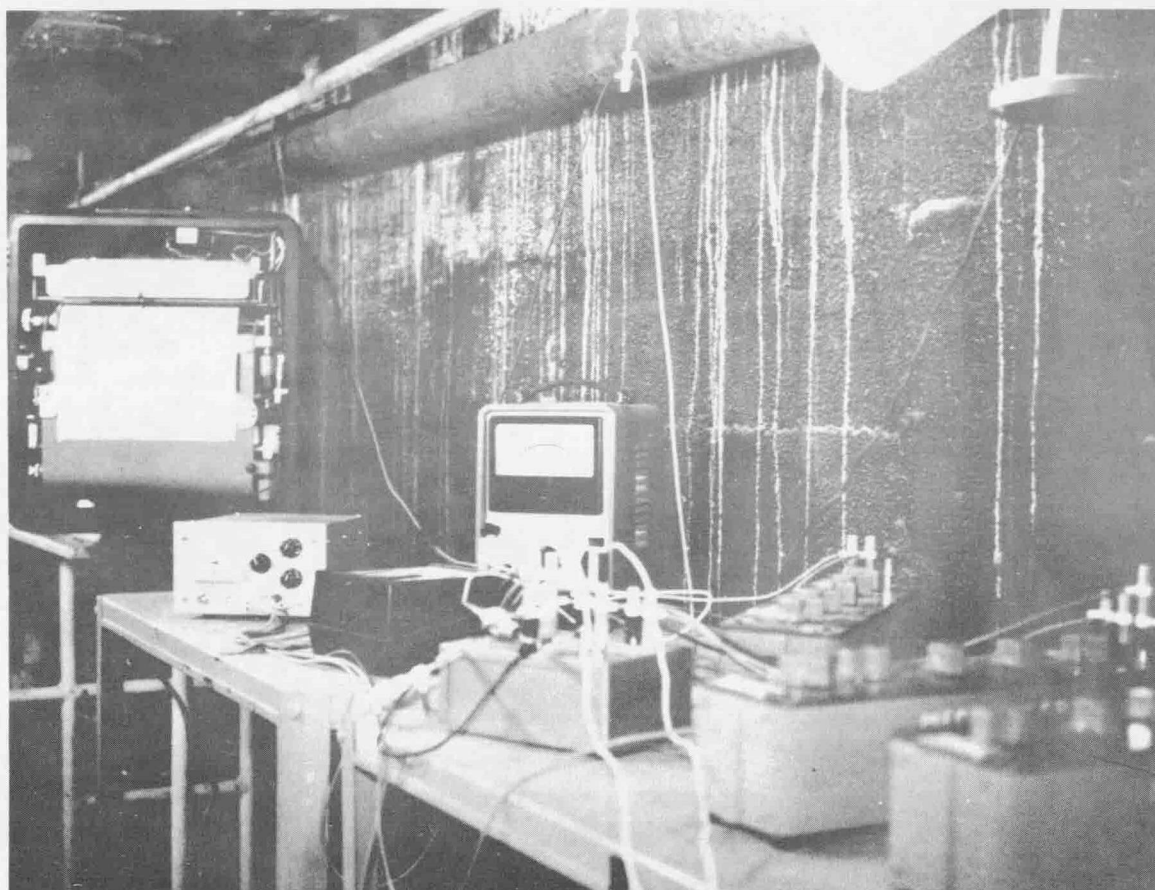


Figure 34. Equipment Used with Thermistor Anemometer

barometric pressure should be recorded at start of calibration.

5. From the calibration curve supplied with the Meriam Instruction manual, read the standard air flow (pounds per minute at standard conditions of 29.92 in. Hg and 70°F) corresponding to manometer reading. Multiply standard pounds per minute by the barometric pressure correction factor and temperature-viscosity correction factor from instruction manual to obtain the real pounds per minute flowing at actual conditions of pressure and temperature. Find the actual cubic feet per minute (acfm) by applying the perfect gas law.

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.

After collecting data as outlined above, a thermistor calibration curve of probe output (millivolts) versus flow (actual cubic feet per minute - acfm) was plotted. The curve, nearly linear and also approaching verticality, accommodated flows from zero to 150 acfm. Also drawn was a curve of resistance (ohms) versus intake temperature (°F) for use with the temperature compensation decade resistance box.

Two months were spent in conducting preliminary ventilation tests and becoming acquainted with the operation of various combustion equipment, temperature and pressure equipment, and the thermistor flow device. During this period of time results of flow measurement had been fairly consistent and reproducible. Then for several days the zero setting of the thermistor in still air seemed to drift from conditions dictated by the

temperature versus resistance calibration curve. Previous examination of literature on thermistors and comments by Electronics Research personnel indicated the possibility that thermistors may lose their reliability with use and time due to loss of calibration. No definite reasons for a possible calibration loss were cited except that perhaps the thermistor bead deteriorates enough to change the heat transfer characteristics of its surface.

To check out the above theory, it was decided to recalibrate the thermistor device. Two or three calibration runs, conducted according to the procedure outlined previously, covering the entire flow range from 0-150 acfm produced completely different and disassociated thermistor output readings. No sensible relationship between output and flow could be found. At this point we decided that the thermistor was completely deteriorated or "burned up". The surface of the bead appeared brown and somewhat charred.

Another thermistor bead similar to the first was acquired from the electronics laboratory. The only difference in the two thermistors was in the so-called response time. The second bead had a response of 25 seconds as compared to 2 seconds for the original bead. It was anticipated that the longer response time would decrease sensitivity enough to provide a more constant recording of output on the millivolt recorder, thus making output easier to read.

In probe form the second thermistor had an aluminum shield around the bead. The bead was located in the path of a diametral hole which ran through the hollow cylindrical shield. With the hole parallel to the air

flow path, straighter flow lines around the bead than previously obtained were expected at all rates of flow.

Several calibration runs were conducted with the shielded probe installed in the intake duct. The millivolt recording deviated from a straight line by as much as three-fourths of a millivolt but readings were averaged with a polar planimeter. The flow-millivolt curve had many erratic points; enough that the results could not be relied on.

It was again decided to use another thermistor. The third thermistor bead chosen required the same design voltage and current and had the same response time (2 seconds) as the original one. In order to retard the response time slightly, epoxy was added to the thermistor in the form of a thin coating over the bead.

Before starting to calibrate the new thermistor, further investigation of bridge circuits and instruments was carried out. It was discovered that current through the probe was not constant as had been originally planned during circuit design. It was strongly believed that this was the main cause for the wide variations in millivolt output from thermistor.

The designer of the original circuit was again contacted and the problems were discussed. As a possible solution, a new circuit which completely eliminated the thermistor bridge box was attempted. A number of decade resistances replaced the bridge, the idea being to obtain closer control of probe current. A calibration run proved, however, that the contrary was true. Current and voltage control through the probe was not only difficult to control but also unwieldy and inconvenient. Due to this fact, the third thermistor was overburdened with a power surge and burned

out.

The train of thought returned to the original thermistor bridge circuit. After lengthy counseling with Professor Drummond of the Mechanical Engineering Department, it was agreed that the bridge should work satisfactorily if a few minor changes in operational methods were made. According to Fenwal Electronics, Inc., Framingham, Massachusetts, in their "Thermistor Manual" brochure, bead thermistors should be able to dissipate a maximum of about 850 milliwatts of power at a maximum safe continuous current of nearly 18 milliamps. With the first three thermistors operating with 40 volts at the supply, the probe current (about 12 m.a.) was well under the maximum allowable value. This indicated that previous burn-outs were caused either by power surges or by allowing the thermistor to set too long in still air with full power applied.

For the fourth thermistor probe the bridge supply voltage was increased to 47 volts which allowed nearly 15 milliamps through probe. This increased the possibility of bead "burn-out", but according to the thermistor characteristic curves, these power and current levels were tolerable. The main object of the power increase was to try to produce a more nearly horizontal calibration curve than was obtained originally.

Previous experience indicated that the method for calibrating the temperature compensation resistance or so-called zeroing of the millivolt output scale was deficient. The still air method originally used was changed somewhat by sealing the probe and a thermocouple in a pyrex beaker which in turn was immersed in a larger container of water. The temperature inside the beaker containing the thermistor was varied by

changing the temperature of the water in the outer container. Readings of temperature and thermistor resistance were taken when conditions in the flask reached equilibrium. In this manner a calibration curve for temperature compensation was obtained for air temperatures ranging between 60° F and 100° F. Results of later spot checks revealed that points were reproducible within two percent of those on the curve.

Calibration of the thermistor for air flow measurement was again accomplished using the laminar flow element as before. Flow tests were again resumed using a flame in the stack. After one day of testing, the compensation resistance required to zero the output scale had drifted to the point where it was 30 ohms (about 20 percent) above what the calibration curve dictated. We checked the thermistor bead only to find that it too had burned up or at least disintegrated beyond useful purposes.

In an attempt to resolve the thermistor burn-out problem, it was discovered that the epoxy being used to coat the thermistor beads begins to deteriorate above a temperature of about 270° F. The thermistors had been operating at a temperature of nearly 350° F. Perhaps as the epoxy decays, the heat transfer characteristics of the unit change enough to adversely affect the resistance-temperature relationship of the thermistor. This phenomenon may account for the zero shift in output and add to the deterioration rate of the thermistor itself.

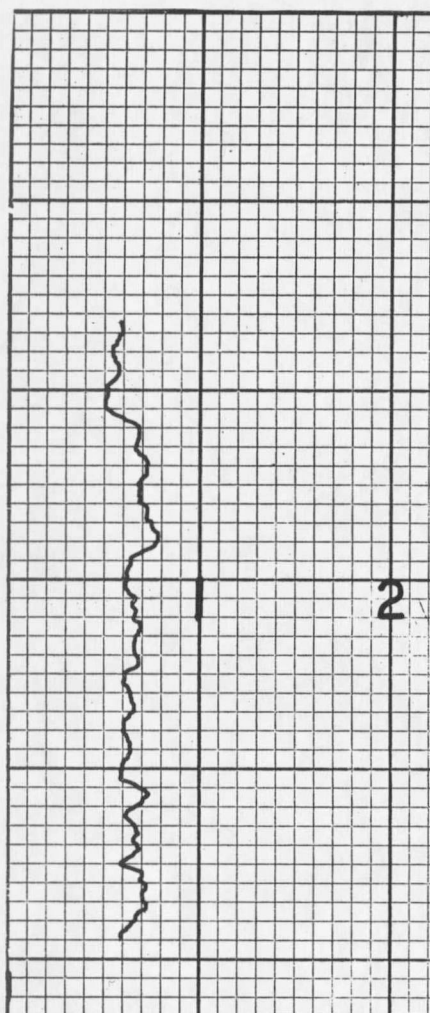
It was believed that the only way to check this theory was to try another thermistor without an epoxy coating. A probe of this type was built and installed in the intake duct. In order to decrease the possibility of further burn-up, the power through the bead was limited to 100

milliwatts which was 30 percent below the maximum tolerable level. A resistance of 2550 ohms, found by trial and error and connected in parallel with the thermistor, allowed operation at about 83 milliwatts with the original 40 volt bridge supply.

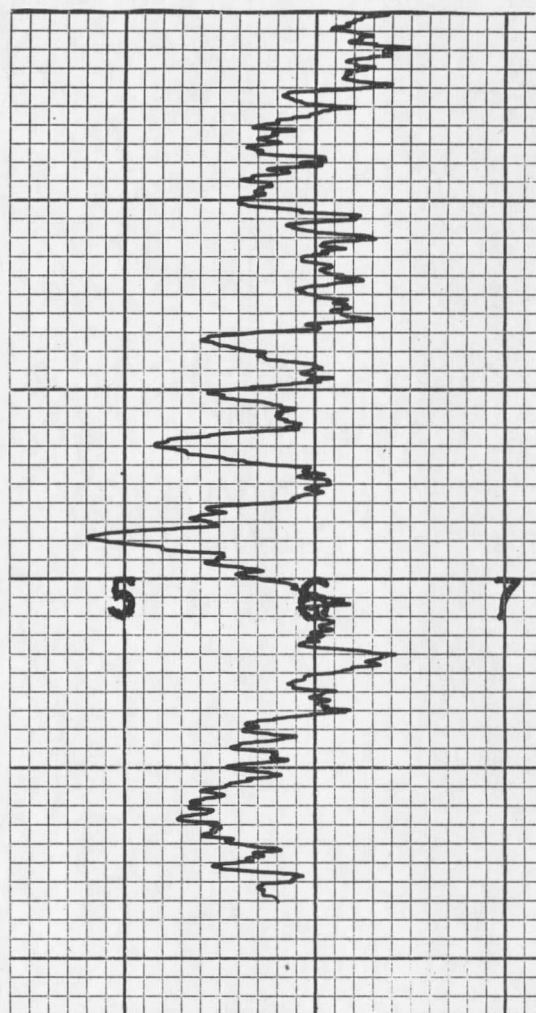
A temperature compensation calibration curve was again obtained by the previously described water immersion method. A flow check revealed, as had been anticipated, that the bare thermistor bead was too sensitive to air velocity changes. This was apparent from the millivolt output recording. Recorded millivolt values at flows above 80 cfm deviated as much as 15 percent from a straight line when plotted. Figure 35-B on page 88 shows the degree and rapidity of millivolt output deviation. The chart speed during this particular run was six inches per minute.

In an attempt to reduce the sensitivity of the probe to the higher velocity air flows, two more 8 in. intake ducts, also about 4 ft long, were installed immediately above and below the first one. Still using only the single probe in the original duct, a complete calibration run with all ducts open was started. With the triple intake arrangement, the velocity past the probe should have been about one-third of the value previously obtained with a single duct. A spot test at about 100 acfm revealed that the output recording was as erratic with three ducts as with one. This is still not understood.

Another attempt to linearize the thermistor output consisted of filling the end of the center duct with one inch diameter aluminum foil tubes, twelve inches long, placed with their longitudinal axis parallel to the longitudinal axis of the main duct. The tubular "grid" ends about four



A. Low Flow. Output reading is 0.65 mv at 18.5 acfm.



B. High Flow. Output reading is about 5.8 mv at 86 acfm.

Figure 35. Comparison of Millivolt Output from Thermistor at Low and High Flows

inches from the thermistor. We found that the tube arrangement straightened the air flow path before it passed by the probe to the extent that the maximum millivolt output variation was under 7 percent at the highest air flow rate of 150 acfm. This was within tolerable limits because the output readings were averaged anyway by taking planimeter readings.

With some hope in the thermistor method of flow measurement restored, calibration was continued. These hopes were again shattered near the end of the run when the temperature compensation resistance setting began drifting as it had with the epoxy-covered beads. From this behavior it was concluded that temperature compensation is not practical with this type of thermistor bridge circuit. A possible explanation is that the rate of heat transfer from the thermistor differs at different air temperatures and thus the thermistor resistance is perhaps not a constant for any particular temperature.

Endeavoring to find a solution, the compensating resistor of the bridge was arbitrarily fixed at 725 ohms and calibration tests conducted at several different but constant intake air temperatures. The final result was a family of linear calibration curves of millivolts versus flow, one curve for each temperature. Nearly all points for each curve were reproducible within $\pm 5\%$ which amounts to a maximum flow deviation of about 7 cfm. It was decided to proceed with ventilating tests and rely on this arrangement of air flow measurements, keeping in mind the accuracy of calibration.

After several days of testing, using kerosene lamps as the heat source at the stack, the millivolt recorder failed to operate. Several

parts including tubes, fuses, a rectifier, and transformer had burned out. Another recorder was borrowed from the Physics Department, Montana State College, which necessitated another calibration run. After this had been done, more test data collected, and graphs plotted of flow versus stack temperature, it was very evident that the thermistor method of measuring air flow, as it was arranged, was by no means accurate enough for practical purposes. The points on the experimental flow-temperature graph were so widely scattered that no reasonable curve could be drawn through them. Because stack temperature measurements were reasonably accurate, the major difficulty was assumed to arise from the thermistor and its related circuitry. Thus, at this point it was decided to abandon the idea of measuring air flow with bead thermistors.

The difficulties and problems encountered as outlined in this section have sparked additional people to investigate the possibility of thermistor air flow measurement. If the reader desires further information on thermistors, their applications, and factors influencing flow measurement via thermistors, he is referred to a thesis by Mr. Bill Cousineau, Montana State College, entitled "Thermistor Anemometry Including Effects of Ambient Pressure and Humidity".

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