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Outdoor-Air Supply and Ventilation of Furnace Closet Used with a Warm-Air Heating System

by

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A REPORT OF AN INVESTIGATION

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Outdoor-Air Supply and Ventilation of Furnace Closet Used with a Warm-Air Heating System

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ABSTRACT

This bulletin is the second report of results obtained in Warm-Air Heating Research Residence No. 2 and includes studies dealing with the introduction of outdoor air into the Residence and the ventilation of a closet-furnace installation.

Outdoor air was introduced into the return-air inlet of the furnace, and positively discharged into all of the conditioned spaces. The introduction of outdoor air was equivalent to an air-flow rate corresponding to 0.44, 0.89, and 1.33 air changes of room volume per hour. The indoor-air pressure was found to be greater than atmospheric pressure with some wind directions even when no outdoor air was introduced. The magnitude of this pressure increase was dependent upon both the wind velocity and wind direction. No evidence was obtained that any appreciable reduction in normal infiltration resulted from the introduction of sizable rates of outdoor air. Admission of outdoor air did result in an appreciable increase in fuel consumption. The percentage increase in fuel consumption due to the introduction of outdoor air was found to correlate with the calculated increases in heat loss when the entire structure was taken into consideration, but not when the first-story rooms alone were considered. The introduction of outdoor air resulted in a marked reduction in the indoor relative humidity, under some conditions to an undesirably low value since no artificial humidification was used. No difficulties were experienced in temperature control or plant operation over the range of outdoor flow rates investigated.

Studies were made with a furnace-closet installation in which the furnace was located in a closet isolated from the remainder of the house; it was barely large enough to accommodate the furnace. Two ducts connected the closet space either to the attic or to the outdoors for the purpose of providing air for the combustion process as well as for ventilation of the closet. Twenty arrangements of opening areas of the two ducts, as well as for the flue pipe, were studied. The direction of air flow in the vertical ventilating duct was apparently dependent upon the relative magnitudes of the stack effect of the ventilating duct and the draft effect of the flue pipe and chimney. The horizontal arrangement of the two ventilating ducts was found to be more critical than the vertical arrangement, particularly with a blocked flue. The recommendations were made that each of the vertical ducts should have an opening area of at least \( \frac{1}{2} \) sq in. per 1000 Btu per hr of rated input; while each of the horizontal ducts should have an opening area of at least 1 sq in. per 1000 Btu per hr of rated input.

With the furnace installed in an open basement and under conditions of low chimney draft, the exhaust action of a fan resulted in a reversal of chimney venting action and the spillage of combustion products into the house.
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I. INTRODUCTION

1. Preliminary Statement

This bulletin is the second report of results obtained in Warm-Air Heating Research Residence No. 2, and includes the studies conducted during the 1949-50 and 1950-51 heating seasons. This Residence was completed in June 1947 and was built, furnished, and completely equipped specifically for research in warm-air heating by the National Warm Air Heating and Air Conditioning Association. It replaced the original Warm-Air Heating Research Residence\(^1\) in which investigations of warm-air heating systems and summer-cooling systems were conducted from 1924 to 1946.

This investigation was conducted under the terms of a cooperative agreement made in 1918 between the Association and the Engineering Experiment Station of the University of Illinois. In this cooperative research agreement the Association is represented by its Research Advisory Committee. During the period of investigation reported, the committee consisted of fifteen men:

J. B. Burrowes, Lau Blower Co., Dayton, Ohio.
K. T. Davis, Bryant Heater Division of Affiliated Gas Equipment, Inc., Cleveland, Ohio.
G. W. Denges, Williamson Heater Co., Cincinnati, Ohio.
R. S. Dill, National Bureau of Standards, Department of Commerce, Washington, D.C.
R. A. Galieck, May-Fiebeger Co., Newark, Ohio.

* Numbers refer to corresponding entries in References.

F. W. Taylor, National Warm Air Heating and Air Conditioning Association, Canadian Chapter, Toronto, Ontario.
H. Weyenberg, Holland Furnace Company, Holland, Michigan.

The investigation was sponsored in part by the American Gas Association through its Committee on Domestic Gas Research as a PAR plan activity of the Association (AGA Project No. DGR-7-CH). Guidance was given by the Technical Advisory Group for Heating and Air Conditioning Research which was composed of:

S. J. Levine, Chairman, General Electric Company, Bloomfield, New Jersey.
W. L. Amann, Houston Natural Gas Company, Houston, Texas.
C. E. Blome, William Wallace Company, Dallas, Texas.
K. T. Davis, Bryant Heater Division, Affiliated Gas Equipment, Inc., Cleveland, Ohio.
W. A. Marshall, Dearborn Stove Company, Dallas, Texas.
R. D. McNeice, Public Service Electric & Gas Company, Newark, New Jersey.
W. M. Myler, Jr., Surface Combustion Corp., Columbus, Ohio.
W. J. Riester, Peerless Manufacturing Company, Louisville, Kentucky.
2. Objectives of Investigation
The over-all objective of the investigations conducted in Research Residence No. 2 is to make studies of the performance of warm-air heating systems, with emphasis placed upon the evaluation of the comfort produced by these systems and of the cost of producing that comfort.

The objectives of this particular investigation were as follows:
(a) To study the performance of a conventional forced warm-air system over a wide range of weather conditions. No outdoor air was introduced directly into the return-air duct, and only room-air was circulated. The results from this study serve as a basis for comparison with those from other studies. The calculated infiltration rate, as determined by conventional procedures, was equivalent to 0.95 air changes per hr.

(b) To study the performance of the same system as in (a) except that outdoor air was introduced into the return-air side of the system at a rate equivalent to 0.44 air change of room-air volume per hr.

(c) Same as (b) except with 0.89 air change per hr.

(d) Same as (b) except with 1.33 air changes per hr.

(e) To study the performance of the same system as in (a) except that the blower-furnace combination was enclosed in a closet isolated from the rest of the basement. The combustion air for the furnace, as well as the ventilation air for the closet, was introduced from a vented attic space, or directly from the outdoors.

(f) To study the venting characteristics of the chimney when an exhaust fan was used, both with and without the introduction of outdoor air to the return-air side of the system.

3. Glossary
Air changes per hour — The number of changes of room-air volume per hr due to infiltration air leakage or introduction of outdoor air. Based on standard air density of 0.075 lb per cu ft. See “Air recirculations per hour.”

Air-flow rate — The rate of circulation of air in cu ft per min (cfm). Unless otherwise stated, all cfm values are for standard air density of 0.075 lb per cu ft.

Air recirculations per hour — The number of changes of room-air volume per hr due to recirculation of room air only. See “Air changes per hour.”

Blower — A centrifugal fan. The warm-air heating industry uses the term to distinguish centrifugal fans from propeller fans.

Blower cycle — One complete cycle of operation from the time the blower begins operation until it begins a second operation, following an off-period.

Bonnet capacity — The heat output of the furnace available at the bonnet, in Btu per hr for a specified air-temperature rise through the furnace.

Bonnet efficiency — The ratio of the bonnet capacity to the heat liberated in the furnace by the burner, also expressed as a percentage. For approved gas-fired forced-air furnaces, the rated bonnet efficiency is 80 percent.

Breathing-level temperature — Temperature of room air measured at a level 60 in. above floor.

Burner cycle — One complete cycle of operation from the time the burner begins operation until it begins a second operation, following an off-period.

Ceiling-level temperature — Temperature of room air measured at a level 4 in. below ceiling.

Design heat loss — The calculated heat loss for a given space based on outdoor design conditions for the locality. In the text the outdoor design conditions are assumed as -10 F and 15 mph wind velocity.

Draft effect — The effect produced on indoor pressure by the chimney draft.

Duct transmission efficiency — The ratio of the register delivery to bonnet capacity; also expressed as a percentage.

Extended plenum duct — A trunk duct which is of uniform size along its entire length.

Floor-level temperature — Temperature of room air measured at a level 4 in. above floor.

Flue-gas temperature — Temperature of flue gas measured at the exit of the flue pipe from the furnace and below draft hood. CO₂ content of flue gas was also determined at the same location.

Fuel consumption — The consumption of fuel per 24 hr. For gas-fired equipment the units are in terms of cu ft of gas per 24 hr. (For these studies,
the calorific value of the gas was 1000 Btu per cu ft.)

Fuel-input rate — The rate of heat liberation in the furnace by the burner assuming complete combustion expressed in Btu per hr.

Furnace bonnet — A central plenum, or collecting chamber, usually located above the furnace, in which the heated air is mixed before distribution to the duct system.

Furnace casing — The jacket or enclosure surrounding the furnace. In forced-air furnaces, the casing is usually insulated.

Intermittent blower operation — A method of blower operation in which the blower cycles frequently during average winter weather and occasionally even with severe weather.

Living zone — The space in a room between the floor level and the breathing level.

Panel effect — A heat transfer effect similar to that obtained from a panel heating system, in which warmed surfaces transmit heat by radiation to cooler surfaces and by convection to cooler air next to the panel surfaces.

Register delivery — The heat available at the registers, in Btu per hr. This is based on the airflow rate through the register and the difference between register-air temperature and the air temperature at the return-air intake.

Sitting-level temperature — Temperature of room air measured at a level 30 in. above floor.

Stack effect — The "chimney action" in a stack, or duct, resulting from the difference in air densities inside and outside of a stack.

Temperature differential, room-air — The difference in air temperature in a room at two elevations. Usually the sitting level, 30 in. from floor, is considered as the reference level. See "Temperature gradient."

Temperature gradient, room-air — A curve representing air temperatures existing at several elevations in a room at one station. See "Temperature differential."

Thermostat differential setting — An adjustable setting of the room thermostat which controls the amount of temperature change required to position the thermostat contacts from "cut in" to "cut out" points.

Total heat input rate — The sum of the fuel input rate and the rate of heat input from lights and other household appliances.
II. DESCRIPTION OF EQUIPMENT AND PROCEDURE

4. Research Residence No. 2

The Residence, the rear or south view of which is shown in Fig. 1, is a one-story structure of frame construction with a large amount of glass exposure and a full basement. A detailed description of the Residence has been presented\(^{(a)}\) in Engineering Experiment Station Bulletin 401. A summary of the construction data, room dimensions, and heat loss calculations are presented in Table 1. The heated space during these studies consisted of only the first-story rooms. The calculated heat loss, based upon an outdoor design temperature of \(-10^\circ F\), was 31,047 Btu per hr for the first-story rooms and the Residence was completely furnished and occupied by a family of two adults, and all observations were made under normal living conditions.

5. Furnace and Duct System

A first-story plan of the Residence is shown in Fig. 2. All registers in the first-story rooms were at the high-sidewall location, \(6\frac{1}{2}\) ft from the floor, with the exception of the baseboard register in the front hall. All of the five return-air intakes were located in the baseboard.

The duct system shown in Fig. 3 was of the extended plenum type\(^{(b)}\) having uniformly sized trunk ducts. The branch ducts were connected to the top or side of the trunk ducts, and were unchanged in size from the trunk take-off fitting to the register stackhead.

Although no heated air was introduced into the basement, even in the coldest weather, the heat regain from the furnace casing, furnace bonnet, duct system, and flue pipe as well as the heat from lights was sufficient in magnitude to maintain the basement-air temperature above 57 F when the furnace was not enclosed in the closet. A sizable heat loss occurring from the basement to the outdoors cannot be neglected in any consideration of the total utilization of fuel input to the furnace.

![Fig. 1. South View of Warm-Air Heating Research Residence No. 2](image-url)
tional units 8 in. in diameter. The total chimney height was 26 ft 11 in. above the basement floor; the 5-in. furnace flue connection was at an elevation of 6 ft 9½ in.; and the top of the chimney extended 2 ft 6 in. above the roof ridge. The draft hood was

raised 8 in. above the normal position to accommodate instrumentation for measuring the flue gas temperature and the CO₂ content in the flue gas. The change in the height of the draft hood, which resulted in an excessive air supply to the burner, necessitated a reduction in the flue passage. For the gas-fired water heater, a separate 4-in. diam chimney of cement-asbestos material was provided.

A room thermostat of the heat-anticipating type was located in the front hall near the living room at an elevation of 30 in. from the floor. This thermostat, which controlled the operation of the burner, was adjusted to provide a minimum temperature differential. The thermostat setting was maintained at 72 F throughout the 24 hr constituting a test period. The auxiliary control equip-

ment consisted of a fan switch and limit control, both located in the furnace bonnet. The blower was operated by means of the fan switch, which was adjusted to start the blower whenever the bonnet-air temperature attained 100 F and to stop the blower when the bonnet-air temperature decreased to 80 F. The limit control in the burner circuit was set to close the gas valve whenever the bonnet-air temperature exceeded the 200 F setting of the control.

Natural gas having a calorific value of 1000 Btu per cu ft was used as the fuel. The furnace, which had a normal rated input value of 90,000 Btu per hr and a rated bonnet capacity of 72,000 Btu per hr, had been selected originally (2) to handle the total design heat loss of 60,780 Btu per hr for the first story and heated basement. However, for the studies reported in this bulletin, the basement was unheated and only the heat loss of the first story (31,047 Btu per hr) was considered. The desired input rate was determined by dividing the first-story heat loss by the assumed bonnet efficiency of the furnace and the assumed duct transmission efficiency.

\[
\frac{31,047}{0.8 \times 0.85} = 45,600 \text{ Btu per hr}
\]

Except in those studies where it is otherwise specified, the fuel input rate was reduced from the normal rated value to the desired value of 46,000 Btu per hr by adjusting the flow rate at the meter. Despite the large reduction of input rate, no trouble was experienced from flame failure during normal operation. The reduction of the fuel input rate by about one-half resulted in an increase in excess air supplied to the burner. A reasonable restriction was made in the flue passage at the furnace outlet in order to reduce the amount of excess air, but since any further reduction might have resulted in faulty operation of the burner, additional corrective measures were not employed. The net result was that the CO₂ content in the flue gas was decreased from about 8.5 percent with rated input to about 5.5 percent. It was not possible to further raise the CO₂ content materially without closing the secondary air openings into the furnace, a practice which is not recommended.

For the desired air-temperature rise of 100 F between the furnace inlet and discharge, the fuel input rate of 46,000 Btu per hr required a total air-flow rate of 340 cfm. As will be discussed later, this
flow rate was maintained for the basic study in which all of the air was being recirculated from the first-story rooms, and no outdoor air was introduced into the return-air duct system. In order to provide a fair basis for comparison, this total flow rate of 340 cfm was also maintained in all the studies during which three different amounts of outdoor air were introduced.

6. Outdoor-Air Duct

For the investigation dealing with the effects of introducing outdoor air, a special 9-in. diam duct was installed as shown in Fig. 4. The outdoor air entered this duct through a screened opening in the north window well which was located to the east of the front door. The size of the duct was selected as 9 in. on a basis of an available energy head of 0.10 in. of water, a maximum flow rate of 180 cfm, and the equivalent lengths corresponding to the duct system shown in Fig. 4. The air-flow rate was determined by means of a calibrated venturi-meter, having a throat diameter of 4 in. In the calibration of this meter, the flow rate was correlated with the differences in static pressures existing at the upstream and throat sections. For the purpose of obtaining a continuous record of the pressure differential across the venturi-meter, a 24-hr recording draft gauge was connected to the two pressure stations.

By suitable adjustments of the orifice openings near the furnace and the setting of the return-air damper in the vertical return-air duct, it was possible to obtain any desired combination of flow rates for outdoor air and recirculated room air. The calibrated vane anemometer(2) installed in the vertical return-air duct provided a means of measuring the air-flow rates for the recirculation air alone.

For the study of the effect of outdoor-air supply, rates corresponding to 0, 0.44, 0.89, and 1.33 air
changes of room volume per hr were used, and the test series were designated as A-111, C-0.5, C-1.0, and C-1.5, respectively. These air changes were based on a house volume determined from internal room dimensions, without deducting the volumes for closets and cabinets. If the volumes for closets to prevent any pressure effect of the blower from acting on the closet space.

c) An inclined flue pipe 5 in. in diam connecting the draft hood to the chimney located outside of the closet. Since the draft hood was located above the furnace and within the closet space, both com-

and cabinets were accounted for, these air changes would be equivalent to 0, 1/2, 1, and 1 1/2 air changes, respectively.

7. Furnace-Closet Installation

For the studies relating to the ventilation of a furnace closet, the furnace in the basement was enclosed in a tightly built closet approximately 4 ft x 4 ft x 7 ft high, as shown in Figs. 5 and 6. The enclosure was constructed of plywood and cement-asbestos board, and all joints were sealed with tape. An access door to the front of the furnace was attached to the closet by means of wood screws, and the joints were also sealed. Seven openings into the closet provided for:

(a) A vertical return-air duct connection to the blower inlet. All joints were sealed to prevent any suction effect of the blower from acting on the closet space.

(b) A furnace bonnet connection to the discharge opening of the furnace. All joints were sealed

bustion products and dilution air were exhausted through the flue pipe.

(d) A vertical 7-in. square ventilation duct connected to the top of the closet and leading to a point 2 ft 6 in. above the joists in the attic space. This duct is referred to hereafter as duct V. The intended purpose of duct V was to permit circulation into the attic of any heated air accumulated at the ceiling of the closet.

(e) A vertical 7-in. square duct for the purpose of supplying cool attic air for ventilation as well as for the combustion process. This duct extending from near the closet floor to a point 2 in. above the joists in the attic will be referred to hereafter as duct CV.

(f) A horizontal 7-in. square ventilation duct connected near the top of the closet, as indicated by the broken lines in Figs. 5 and 6. This duct, referred to as duct V-H, was extended into the window well.
(g) A horizontal 7-in. square duct for the purpose of supplying air for ventilation as well as for combustion was installed about 4 ft above the floor. This duct, referred to as duct CV-H, was also extended into the window well. The two horizontal ducts were studied separately from the two vertical ducts.

For any given study, therefore, the closet space was provided with three separate ducts, one of which was the flue pipe; and the other two led either to the attic or to the window well. Since the pressures and temperatures existing in the attic space were related to the outdoor weather conditions, the provisions for ventilating the attic are of interest. Screened and louvered openings of 1.6-sq ft area were provided at each of the east and west walls of the attic. In addition, a 1-in. air space under the eaves on two sides of the Residence pro-
Fig. 5. Schematic Diagram of Furnace-Closet Installation
vided a 3-sq ft opening. For the studies conducted using the vertical ducts, the test series were denoted by the prefix “D”; for studies using the horizontal ducts, the prefix “E” was used.

Special instrumentation was provided for these studies as indicated in Fig. 5. Four thermocouples connected in a parallel-grid circuit were installed near each end of ducts V, CV, V-H and CV-H. For the purpose of measuring the extremely low velocities of the air passing through these ducts, four anemometer measuring stations were provided. The anemometers were of the heated-thermocouple type, developed by Hershey and Engdahl.\textsuperscript{4,5} At each station four of the anemometers were uniformly spaced across the duct to obtain a weighted average of velocities and were connected in series to give a magnification of cmf developed. The pair of lead wires from the anemometer was connected to a recording potentiometer. Provision was made to record the pressure differentials existing between the furnace closet and attic and between the closet and first-story rooms. A standard was located within the closet, as indicated in Fig. 6, for supporting seven thermocouples located at 1-ft intervals from floor to ceiling. Thermocouples S-2, S-6, and S-9 were installed in the ceiling surface of the closet.

8. Instrumentation

For the purpose of determining temperatures, approximately 200 thermocouples of 24-gauge copper-constantan wire were installed in the Residence. Thermocouples were installed at four different levels on a standard located near the center of each of the first-story rooms as shown in Fig. 2, at three stations in the basement, on the ceiling and floor surfaces, in the attic, and in the duct system, and at other desired points inside and outside the Residence. All thermocouples were connected to two central switchboards on the instrument panel in the basement, and each of the switchboards was connected to an indicating potentiometer. By means of a recording potentiometer to which any twelve of the 200 thermocouples could be connected, it was possible to obtain a continuous record of the temperature at any twelve stations.

Resistance thermometers having greater sensitivity than thermocouples were installed at the 30-in. level on the thermocouple standards in five of the first-story rooms and were connected to a sensitive recording potentiometer. The temperature of the air in the trunk ducts was measured by means of two thermocouple grids, each consisting of six thermocouples, connected in parallel for the purpose of obtaining average temperatures at each station. These grids, which were located in the trunk ducts close to the furnace bonnet, were not affected by direct radiation from the furnace so that the measured temperatures also represented the bonnet-air temperatures.

A recorder located in the living room was used to measure the relative humidity of the room air, frequent calibrations of the recorder being made by means of a sling psychrometer. The flue gas temperature below the draft hood was measured by means of a recording thermometer, and continuous records of the CO\textsubscript{2} content in the flue gases were made. The electrical inputs to the burner and blower were measured by means of watt-hour meters. Self-starting electric clocks connected across each of the burner and blower motor circuits provided means for obtaining the total times of operation. The fuel input to the furnace was measured by the gas meter, while the gas used for cooking and water heating was measured by a separate meter.

For the purpose of obtaining the pressure differential between the outdoor and indoor air, an outdoor static pressure tube of the weather-vane type was developed, as discussed in the Appendix. The weather station, located about 40 ft north of the Residence, was provided with a cup-type anemome-
ter for recording the instantaneous wind velocities, a weather-vane for indicating the wind direction, and an Eppley Pyrheliometer for recording the solar intensity.

9. General Procedure

Either periodic or continuous records were made of all significant temperatures. Complete daily records were made of the operating time, the number of cycles of operation, the electrical consumptions of the gas valve and the blower motor, and the total electrical consumption for the Residence. Daily observations were made of the amounts of gas consumed in the furnace and for household purposes.

For the purpose of making comparisons, studies were conducted alternately for a period of a few weeks with each separate series. It was possible, therefore, to obtain performance characteristics of each of the various arrangements over a wide range of weather conditions.
III. WIND VARIABILITY, INFILTRATION, AND PRESSURE DIFFERENTIAL BETWEEN INDOORS AND OUTDOORS

In addition to the objectives listed in Section 2, information concerning the following factors which affect infiltration was considered essential for an understanding of wind action on a house:

(a) variability in velocity and direction of wind
(b) pressure differential between indoors and outdoors
(c) wind effects on outdoor-air supply.

10. Factors which Affect Infiltration

In the calculation of heat losses from a structure, the largest indeterminate factor is the air infiltration through crackages. Two methods for calculation of this infiltration are commonly employed in practice. The simplest method, which involves arbitrary selections of the number of changes of room-air volume that might occur in 1 hr due to infiltration air leakage, is referred to as "air-change" method. The second method, which is applicable to a wider range of building construction, is referred to as the "crackage method" for determining infiltration.

Experimental factors for use with the crackage method are given in the A.S.H.V.E. Guide[(5)] and were based upon laboratory studies by Houghten,[(7)] Larson,[(8)] and others. Other studies have also been reported by Houghten,[(9)] Emswiler,[(10)] and Larson[11] in which determinations were made of the infiltration through windows located in actual buildings. More recent studies have been reported by English investigators[12] in which infiltration rates as affected by wind velocity were studied in an experimental house. The results from this study, as well as those made in California,[13] are for relatively mild climates and for houses which are not as tightly built as are the prevailing well-built homes in the colder midwestern states.

Any infiltration of outdoor air which dilutes the concentrations of odor, dust, and moisture can be considered to be beneficial in improving the quality of the indoor air. Actually, however, such infiltration air may carry dust and create drafty regions near the windows and doors. For these reasons, an emphasis[14] has been placed on tighter construction by means of storm doors, storm sash, weatherstripping and vapor barriers. In many cases, particularly in better constructed homes, the reduction of infiltration has given rise to other complaints of stuffiness, odors, and a lack of "freshness." These complaints have been more frequent in recent years, probably as a result of the increase in smoking habits.

Granting that the introduction of outdoor air is desirable, but that it should be by other means than through window and door crackages, the idea has been proposed that infiltration could be largely eliminated by the positive introduction of outdoor air into a structure. In this case, the outdoor air should be tempered, filtered, and then introduced through the duct system and registers into the rooms to be conditioned. In some design procedures the assumption has been made that when the outdoor-air supply is equal to or greater than the normal infiltration supply, the pressure within the conditioned space will be greater than that outdoors, and normal infiltration would be reduced to a negligible quantity. However, no proof exists that the introduction of moderate quantities of outdoor air into the structure would preclude infiltration.

11. Variability in Velocity and Direction of Wind

Preliminary observations indicated that neither the rate of outdoor-air input to the Residence nor the pressure differential between indoors and outdoors was constant. Both items were greatly affected by the velocity and direction of the wind. In view of the specific relationship that existed between the Residence and the wind characteristics, the quantitative results reported in this bulletin cannot be considered as strictly applicable to all structures. Nevertheless, as a qualitative representation of the manner in which the many variables affect the performance of a given heating system in a structure, these studies may be of great interest.

The extreme variability that was experienced in wind directions and velocities, which are probably
typical of many other localities, is shown in Fig. 7. The analysis was based on hourly records of wind direction and wind velocities for the month of February 1950. Although the prevailing wind direction was northwest, at some time during the month the wind came from every other point of the compass. Similarly, although a wind velocity of 4 mph was most frequently recorded, the range of average hourly velocities during the month was from 0 mph to 16 mph. The average hourly velocities did not include the temporary gusts and lulls, characteristic of wind action.

12. Indoor-Outdoor Pressure Differential

A typical example of the variability of wind velocity, and the resulting fluctuations in the indoor-outdoor pressure differential, is shown in Fig. 8. The increase in wind velocity from 3 mph at 9 p.m. to an average of 13 mph at 1 a.m. was directly reflected as an increase in the pressure differential, a record of which is shown on the circular chart. The pressure within the structure was slightly below atmospheric pressure between 9 p.m. and 10 p.m.; practically the same between 10 p.m. and 11 p.m.; and distinctly above from 11 p.m. to 1 a.m. It may be noted in Fig. 8 that the indoor pressure was greater than outdoor pressure (positive pressure differential) for westerly winds at high velocity.

The averages for hundreds of separate observations of pressures, velocities, and directions were plotted, and the final faired curves redrawn on the circular graph shown in Fig. 9. In general, the pressure differential was greatest with a westerly or southwesterly wind, and least with an easterly wind. The reduced pressures observed for easterly winds could not be accounted for by the presence of adjacent structures, since the Residence was fairly remote from trees and buildings; the nearest structures were a one-story weather station about 40 ft to the north, a one-story bungalow about 80 ft northeast, and a row of one-story flat-roofed bungalows about 120 ft to the south.

During the first season's observations (1949-50) the outdoor pressure tube was located 3 ft 5 in. above the roof peak. The maximum pressure differential was obtained with a west wind, and the magnitudes were greater than those shown in Fig. 9. The 1949-50 data were not used, however, since some doubt existed as to whether the outdoor pressure tube was located in the free stream flowing over the roof. In the second season the tube was raised to an elevation of 6 ft 2 in. above the roof peak, and subsequently two additional years were devoted to the study. The results summarized in Fig. 9 are, therefore, the recorded pressure differentials for two seasons during which the furnace closet was removed from the basement. Data were also recorded during the summer when cooling studies were conducted in the Residence.

The negative pressure differentials observed during the low velocity winds, indicating that the indoor pressure was less than that outdoors, require some explanation. In the first place, the 8-in. diam and 4-in. diam chimneys, both of which were actively venting the flue gases from the furnace and water heater, provided a chimney draft effect which evacuated air from the house to the outdoors. Under such conditions it is reasonable to expect that the indoor pressure would tend to be less than that outdoors, especially if outdoor air could not leak into the house. Furthermore, as the wind velocity increased, the chimney draft would also increase, so that in air-tight structures even lower indoor pressures would be expected with increased wind velocities.

In addition to chimney draft effect, another influence would tend to give negative pressure differ-
Fig. 8. Typical Records of Wind Velocities and Indoor-Outdoor Pressure Differentials.
Since the tubing connecting the recording pressure gauge in the basement with the outdoor static pressure tube was installed on the outside of the north wall and below the roof rafters, about 22 ft of tubing was exposed essentially to outdoor air temperature. Calculations of the correction for the unequal air columns on the two sides of the recording gauge indicated that the magnitude of the correction could be as large as 0.020 in. of water. In other words, any correction applied to the observed readings shown in Fig. 9 would tend to give larger positive pressure differentials than those indicated. Since the magnitude of the corrections could not be exactly determined for each of the hundreds of readings, and also because the corrections were of the same order of magnitude as the deviations due to wind pulsations and lag effects, no attempt was made to correct for possible temperature effects on the two legs of the recording gauge. During the summer a confirmation was obtained for the fact that the negative pressures shown in Fig. 9 were due to cold weather effects (on the chimney draft and on the tubing connections of the gauge). All differential pressure readings were positive regardless of wind velocity or wind direction during summer operation.

The fact that under some wind directions and intensities the indoor pressure in winter was greater than the outdoor pressure can be accounted for by the relative sizes of openings on the windward and leeward sides of the building. The relationship between pressures and opening sizes can be illustrated by means of a small hollow cube, one face of which is normal to a given direction of air flow. If this en-
closure is made air tight except for a small orifice on the windward side, the velocity pressure corresponding to the air velocity approaching the cube will be converted to a static pressure acting on the surface of the cube. The resulting static pressure inside of the box will approach that of the velocity pressure. If the resulting conversion is complete and no losses occur, wind velocities of 5 mph, 10 mph, and 15 mph would result in internal static pressures of 0.012 in., 0.048 in., and 0.108 in. of water, respectively. Now if a separate small orifice is placed on the leeward side, which is subjected to a sub-atmospheric pressure, the static pressure inside of the cube will be smaller. That is, the magnitude of the internal static pressure will depend upon the relative size of the openings on the opposite sides of the cube.

The Residence, as well as all other buildings, is a more complicated geometrical shape than the simple cube of this illustration. Nevertheless, the pressure phenomena in the Residence due to wind effects can be expected to bear some resemblance to those for the small cube. At least, a qualitative analysis of the various factors should indicate whether the actual results shown in Fig. 9 are consistent with the preceding explanation. In this connection, the total equivalent crackages around doors and windows in the first-story and basement rooms were 99 linear feet for the west wall, 147 ft for the north wall, 60 ft for the east wall, and 163 ft for the south wall. Hence, the largest ratios of crackages on opposite sides of the Residence were for the west-east plane (99/60 = 1.65) and the southwest-northeast plane (262/207 = 1.26). Assuming that for a west wind the velocity pressure is converted to static pressure acting on the west wall and that a suction pressure exists on the east wall, it is reasonable to expect that the internal static pressure will be higher than that for any other wind direction, since the ratio of crackages is largest for the west-east plane. Conversely, the least value of internal static pressure, and hence indoor-outdoor pressure differential, can be expected with an easterly wind since the ratio of crackages is smallest for the east-west plane.

A comparison of the velocity pressures at different wind velocities and the measured indoor-outdoor pressure differentials reveals an interesting correlation of magnitudes, as shown in Table 2. The negative differential pressure observed on a calm day has been discussed earlier. The fact that the values in Column (4) were always less than those

---

Table 2: Velocity Pressures and Pressure Differentials Created by West Wind

<table>
<thead>
<tr>
<th>Velocity Pressure, mph</th>
<th>Wind Velocity, mph</th>
<th>Velocity Pressure Corresponding to Wind Velocity, in. of Water</th>
<th>Observed Indoor-Outdoor Pressure Differential for West Wind, in. of Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>5</td>
<td>0.012</td>
<td>0.027</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>0.048</td>
<td>0.018</td>
</tr>
<tr>
<td>15</td>
<td>15</td>
<td>0.108</td>
<td>0.027</td>
</tr>
</tbody>
</table>

---

in Column (3), and that the discrepancy in values became larger with increased wind velocity, is consistent with the explanation offered for the draft effect of the chimneys.

The relationships shown in Fig. 9 were obtained under conditions when no outdoor air was admitted into the Residence through the outdoor air duct. Repeated observations made with various rates of outdoor air introduction into the Residence revealed negligible additions to the indoor-outdoor pressure differential. The increase in internal pressure was barely discernible with the draft gauges used, of the order of 0.002 in. of water gauge. This indicates that even higher rates of outdoor-air supply would be required to cause a significant increase in internal pressure. This small increase in indoor pressure leads to the suspicion that the Residence cannot be classified as an air-tight structure, although it was as tightly constructed as any well-built residence. As shown in Table 3, the tabulation of the crackages existing around windows and doors and the two flue openings reveals a surprisingly large equivalent opening area. Based upon the data given in the A.S.H.V.E. Guide, the total window and door leakage area is calculated to be about 1.56 sq ft and the cross-sectional areas of the two chimneys amounts to 0.44 sq ft or a total of 2 sq ft of equivalent opening. This approximation is probably on the low side since no account was taken of the possible leakages through the walls and ceiling. Whether the small increase in internal pressure of 0.002 in. of water was effective in reducing the normal infiltration through windows and door crackages could not be determined from direct observations. An indirect indication of this possible effect was reflected in the fuel consumption, as discussed in Section 15.

13. Wind Effects on Outdoor-Air Supply

The values of 60 cfm, 120 cfm, and 180 cfm for outdoor-air supply listed in Table 4 for Series C-0.5, C-1.0, and C-1.5, respectively, represent the air-flow rates during an average winter day for which the indoor-outdoor temperature difference
was approximately 40 F and wind velocity was less than 3 mph. The flow rate was obtained by means of the venturi-meter and recorded continuously as shown on the circular chart in Fig. 10 (p. 26). When the blower was not in operation, as much as 25 cfm entered the return-air duct system as a result of wind effect alone, as shown for the period between 10 a.m. and 11:30 a.m. on the circular chart. After 3 p.m. the venturi pressure difference increased from about +0.07 in. to +0.10 in. of water at midnight, corresponding to outdoor air-flow rates from 120 cfm to 135 cfm. The large fluctuations in the venturi-meter readings between 11:30 a.m. and 1:30 p.m. coincided with gusty wind conditions.

In connection with the problem of locating the outdoor-air intake, the conclusions of Hite and Bray (12) from a study of models are of interest. "No single intake location could be found for which the pressure was unaffected by wind direction. The nearest approach to an intake to meet the requirements of constancy of pressure for all wind directions and having a pressure greater than exhaust or outlet vent pressure was found to be: a narrow band about 3 in. in width completely encircling the building about 0.2 of the height of the vertical sidewall above the ground." (13) From a practical standpoint, this optimum location is not feasible.

A study was conducted over a wide range of outdoor-air temperatures and wind velocities to determine the principal factors affecting the outdoor air-flow rate. Due to the fact that an infinite number of combinations of outdoor temperatures, wind direction, and wind velocity existed, great difficulty was experienced in showing sharply defined trends. Nevertheless some interesting tendencies were observed as shown in Table 5. Lower air-flow rates for the outdoor-air supply were obtained when southerly winds prevailed than when westerly or northwest winds occurred. The southerly winds usually accompanied milder weather and the northwesterly winds prevailed in colder weather. Since the outdoor-air intake was located in the window well outside of the north wall of the basement, any northwest or north winds tended to increase the flow rate, while south winds decreased the rate. Taking into consideration that the north location of the outdoor-air intake gave deviations in the ventilation air-flow rate of the order of only 15 cfm to 20 cfm (Table 5), the location of the intake might be considered to be of relatively minor importance.
Fig. 10. Typical records of outdoor air flow rates with varying wind conditions.
IV. PLANT PERFORMANCE WITH OUTDOOR-AIR SUPPLY

14. Performance of Furnace and Blower

For each of the four series of tests described in Section 6, complete performance curves were established, as shown by Fig. 11, which represents the data for Series C-1.5 only. No attempt was made to differentiate between the plotted points as to wind direction and wind velocity, so that the curves can be considered as representing average weather conditions. By extrapolating the curves to an indoor-outdoor temperature difference of 80 F, some interesting trends become evident. Under these conditions the maximum bonnet-air temperature would attain a value of 155 F, both the burner and blower would be operating continuously, and the fuel input rate would probably be insufficient to offset the design heat loss requirements, as evidenced by the required total hours of burner operation. Since many of the corresponding values for the other three series were approximately the same as those shown in Fig. 11, the complete set of four curves was not included.

For comparative purposes, values selected from the four separate performance curves, together with other pertinent data, have been listed in Table 6 for an indoor-outdoor temperature difference of 35 F corresponding to average weather conditions. A comparison of the data obtained for Series A-111, in which no outdoor air was introduced into the house, with the data for Series A-11, the results of which have been reported previously, indicated a confirmation of results.

The significant data concerning furnace and blower operation are listed as items (3) to (6) in Table 6. The values listed in items (3), (4) and (5) showed no major differences for the four series.

The blower operated about 22 hr per day and approached continuous operation in all series. No appreciable differences were noted in the maximum bonnet-air temperatures obtained during the burner cycle, even though it was expected that these temperatures would be lower for large outdoor-air rates. A special study was made of this apparent inconsistency by determining the rates of change of the bonnet-air temperature as affected by burner operation. Figure 12 shows the increase of bonnet-air temperature under prolonged operations of the burner in which both the fuel input rate and the outdoor-air temperature were constant. The steady-state temperatures attained after about 60 min of burner operation showed the lowest bonnet-air temperature for the largest outdoor-air rate. Since the air-flow rate at the inlet of the furnace was the sum of the recirculation air and the cooler outdoor air, the coolest mixture of air entering the furnace was obtained with the largest outdoor-air rate and was reflected as a lower bonnet-air temperature at the discharge of the furnace. In normal operation, however, the on-period of burner operation was much shorter than 60 min and varied from about 7 min to 30 min for indoor-outdoor temperature differences of 35 F and 60 F, respectively. An inspection of Fig. 12 reveals that the differences between the bonnet-air temperatures were smaller for these shorter on-periods. Furthermore, the time required for the bonnet-air temperature to reach a given value was not the same for the different outdoor-air rates. That is, for a given weather condition, the longer burner on-periods accompanied the larger outdoor-air rates. The net result was that about the same maximum bonnet-air temperatures were attained for all four series. A similar study of the rate of decrease of bonnet-air temperature as affected by the burner off-periods showed that the cooling rates were essentially the same for all four series. Therefore, the maximum and minimum bonnet-air temperatures for normal operation, which were plotted in Fig. 11, were not markedly different for the four series under consideration.

The same analysis applies to the maximum and minimum values of flue-gas temperatures measured during a normal cycle of burner operation. These values were substantially the same for all four series. The introduction of large outdoor-air rates also had no appreciable effect on the combustion process in the furnace, the CO₂ content
15. Fuel Consumption as Affected by Introduction of Outdoor Air

The curves for fuel consumption for each of the four series were separately established, as shown by the typical example at the bottom of Fig. 11. For purposes of comparison, the four curves were transposed to Fig. 13. The fuel consumption increased almost proportionally to the increase in outdoor-air rate. For example, for an indoor-outdoor temperature difference of 35 F, the fuel inputs to the furnace were 365 cu ft, 400 cu ft, 440 cu ft, and 485 cu ft per day for increasing outdoor-air rates. These values are also shown in Table 6, item (6c). As shown by item (6d), the percentage increases in fuel consumption over that for Series A-111, in which no outdoor air was introduced into the Residence, were 9.5 percent, 20.5 percent, and 33.0 percent, respectively, for outdoor-air rates of 0.44, 0.89 and 1.33 air changes per hr.

In Table 7 is shown a comparison of these actual increases in fuel consumption (Column 2) with calculated increases in heat loss from the Residence (Column 8). The items listed in Columns (4) and (5) are transmission and infiltration losses for the first-story rooms only for an indoor-outdoor temperature difference of 35 F. The infiltration loss was assumed to be the same in all four cases; that is, the introduction of outdoor air into the structure was considered as having no influence on the normal infiltration process. The calculated increases in heat loss (Column 8) were considerably greater than the observed increases in fuel consumption (Column 2). It was recognized that this comparison was not valid, since no account was taken of the effects of heat regain to the house and to the outdoor-air duct. Furthermore, in the calculation of heat loss designated as Method A, no consideration was given to the heat loss from the basement. The successive steps which were taken to obtain a better correlation between heat input and heat loss are indicated below.

In the first place, as indicated in Table 6, item (7c), the total heat input to the house includes the heat from appliances, but not from occupants or from the sun. On this basis, the increases in heat inputs were 8.0 percent, 17.4 percent, and 27.6 percent (Table 7, Column 3). These increases were only about one-half as large as the calculated increases (Column 8) based on Method A.

In the calculation for heat loss due to addition of outdoor air (Column 6), the assumption was made that the temperature of the outdoor air did not increase between the window well and furnace inlet. Actually the temperature of the outdoor air increased between the two stations. For an indoor-outdoor temperature difference of 35 F, the temperature rises were 10.5 F, 8.5 F, and 6.3 F, for Series C-0.5, C-1.0, and C-1.5, respectively. The adjusted increases in heat loss (Column 11) were also considerably greater than the corresponding increases in total heat input (Column 3).

As reported previously, the only logical comparison of heat losses and heat inputs can be made by considering the entire house including the base-
Table 6
Summary of Data Obtained with Outdoor-Air Supply

<table>
<thead>
<tr>
<th>Series</th>
<th>A-111</th>
<th>C-1.0</th>
<th>C-1.5</th>
<th>C-1.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Conditions: Air-Flow Rate, cfm</td>
<td>(a) Outdoor Air</td>
<td>0</td>
<td>60</td>
<td>120</td>
</tr>
<tr>
<td></td>
<td>(b) Recirculated</td>
<td>340</td>
<td>240</td>
<td>120</td>
</tr>
<tr>
<td></td>
<td>(c) Total</td>
<td>340</td>
<td>240</td>
<td>140</td>
</tr>
</tbody>
</table>

* Sees C 0.5

Summary of Values for an Indoor-Outdoor Temperature Difference of 35 F

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
<th>Value</th>
<th>Value</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1) Room-Air Temperature Differentials, F</td>
<td>(a) Floor to Br. Level</td>
<td>4.0</td>
<td>4.0</td>
<td>3.8</td>
</tr>
<tr>
<td></td>
<td>(b) Br. Level to Ceiling</td>
<td>3.8</td>
<td>3.7</td>
<td>3.7</td>
</tr>
<tr>
<td></td>
<td>(c) Floor to Ceiling</td>
<td>3.8</td>
<td>7.7</td>
<td>7.5</td>
</tr>
<tr>
<td>(2) Indoor Relative Humidity, percent</td>
<td>(a) Average</td>
<td>45</td>
<td>41</td>
<td>34</td>
</tr>
<tr>
<td></td>
<td>(b) Maximum</td>
<td>50</td>
<td>47</td>
<td>38</td>
</tr>
<tr>
<td></td>
<td>(c) Minimum</td>
<td>40</td>
<td>33</td>
<td>28</td>
</tr>
<tr>
<td>(3) Bonnet-Air Temperature, F</td>
<td>(a) Maximum</td>
<td>115</td>
<td>115</td>
<td>115</td>
</tr>
<tr>
<td></td>
<td>(b) Minimum</td>
<td>95</td>
<td>95</td>
<td>95</td>
</tr>
<tr>
<td>(4) Blower Operation</td>
<td>(a) No. of Operations per day</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>(b) Hr. of Operation per day</td>
<td>22</td>
<td>22</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>(c) Average Length of Off-period, hr</td>
<td>0.4</td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td></td>
<td>(d) Electric Input, watt hr</td>
<td>2700</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td></td>
<td>(e) Minimum</td>
<td>265</td>
<td>290</td>
<td>275</td>
</tr>
<tr>
<td>(5) Flue Gas Temperature, F</td>
<td>(a) Maximum</td>
<td>170</td>
<td>175</td>
<td>160</td>
</tr>
<tr>
<td></td>
<td>(b) Minimum</td>
<td>170</td>
<td>175</td>
<td>160</td>
</tr>
<tr>
<td>(6) Burner Operation</td>
<td>(a) No. of Operations per day</td>
<td>8</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>(b) Hr. of Operation per day</td>
<td>7</td>
<td>8</td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>(c) Fuel Input to Furnace per day, cu ft</td>
<td>365</td>
<td>400</td>
<td>440</td>
</tr>
<tr>
<td></td>
<td>(d) Increase in Fuel Input, %</td>
<td>9.5</td>
<td>20.5</td>
<td>30.0</td>
</tr>
<tr>
<td>(7) Heat Input, Btu per hr†</td>
<td>(a) Fuel Input to Furnace</td>
<td>15250</td>
<td>15670</td>
<td>18340</td>
</tr>
<tr>
<td></td>
<td>(b) Heat from Appliances</td>
<td>2920</td>
<td>2920</td>
<td>2920</td>
</tr>
<tr>
<td></td>
<td>(c) Total Heat Input</td>
<td>18170</td>
<td>18600</td>
<td>21260</td>
</tr>
<tr>
<td></td>
<td>(d) Increase in Input, %</td>
<td>8.0</td>
<td>17.4</td>
<td>27.6</td>
</tr>
</tbody>
</table>

* All air flow rates are expressed in terms of cfm at standard air density of 0.075 lb per cu ft.
† Item 7(a) was obtained from Item 6(e).

There is the sum of Items 7(a) and 7(b).

The outdoor-air temperature was in-

ment. Certainly the total heat input to the house is not confined to the first-story rooms alone, since the heat regain from furnace casing, furnace bonnet, ducts, flue pipe, and chimney serves to heat the basement to some extent. Hence, in Method C, the calculated heat loss for the basement was taken into account. In these calculations the value of 65 F for the basement-air temperature was used in order to conform with the observed temperatures. The total calculated heat losses (Column 15) were approximately 5800 Btu per hr greater than the total heat inputs listed in item (7c) of Table 6. A part of this discrepancy, which was practically constant for all four series, may be due to the fact that the total heat input did not include heat from occupants and from the sun. In any case, the values shown in (Column 15) of Table 7 represent the best estimate of the heat loss from the entire house for an outdoor-air temperature of 37 F and a wind velocity of 15 mph. The calculated increases in heat loss (Column 16) were 6.8 percent, 14.3 percent, and 23.6 percent, respectively, for series C-0.5, C-1.0, and C-1.5. The correlation between these values and the corresponding increase in total heat input (Column 3) were good in view of the numerous assumptions made.

A study was made of the possible increase in bonnet efficiency as the outdoor-air rate was increased and as the air temperature entering the furnace was lowered. A comparison of the values in Columns (3) and (16) indicated differences of 1.2, 3.1, and 4.0 percent for the three series. Since these differences were of the same order of magnitude, it was tentatively concluded that any change in bonnet efficiency was negligible. Additional studies
of bonnet capacities, fuel inputs, and bonnet efficiencies confirmed the fact that any differences in efficiency were small and within the range of experimental deviation.

The evidence presented in Table 7 indicates that the correlation between the total heat inputs and calculated heat losses should only be made by considering the entire Residence. Furthermore, the

![Fig. 13. Fuel Consumption with Various Outdoor Air-Flow Rates](image)

frequently made assumption that the introduction of outdoor air will result in a reduction of the normal infiltration rate does not seem to be upheld. For example, in the case of Series C-1.5, if the total infiltration loss of 6877 Btu per hr for the first story and the basement was assumed to be nonexistent when outdoor air amounting to 180 cfm was introduced into the Residence, the total calculated heat loss would be 22,430 Btu per hr. This value would be less than that for Series A-111 and would have indicated no increase in fuel consumption, which was contrary to the actual findings.

16. Selection of Furnace Size

A greatly oversized furnace would require a higher initial cost than one of barely adequate capacity, but would have reserve capacity for infrequent demands brought about by extreme weather conditions. The selection of a furnace that is barely adequate for average weather demands may provide better means of temperature control in mild weather but may lack reserve capacity. Hence, the selection of a furnace involves a compromise between these two limits.

A detailed discussion of the problems involved in selecting the maximum fuel input rate and the furnace size has been reported.\(^2\) A single equation was suggested for determining the fuel input rate for any residential warm-air heating system, as follows:

\[
H_i = \frac{H_n + H_b + H_f}{(e_n + e_b) - e_f}
\]

in which

- \(H_i\) = fuel input rate required for design weather conditions, Btu per hr
- \(H_n\) = design heat loss for spaces above first-story floor level, including normal infiltration losses, Btu per hr
- \(H_b\) = design heat loss for spaces and construction below the first-story floor level, Btu per hr

In the case of slab-floor construction the heat loss through the floor would be included. If a basement space is heated
to 70 °F, the normal infiltration losses for the basement should be included. If the basement is not heated to 70 °F, an approximation should be made of the expected basement-air temperature. In the Residence studies, a value of 60 °F was assumed.

\[ H_a = \text{outdoor-air heat loss, Btu per hr. No corresponding reduction should be made in the normal infiltration losses calculated in items } H_a \text{ and } H_b. \]

For simplicity, assume that the outdoor air enters the furnace at outdoor-air temperature.

\[ e_b = \text{bonnet efficiency ratio. For gas-fired furnaces in which the heat losses from casing, bonnet, flue pipe, and duct work are regained in the spaces included in the calculations for } H_a \text{ and } H_b, \text{ a value of } 0.9 \text{ is suggested. For furnace-closet installation vented to the attic, and for furnaces located in the attic, crawl space, or outside the structure, a value of } 0.8 \text{ is suggested for gas-fired furnaces since the heat regain will be small.} \]

\[ e_t = \text{duct transmission efficiency ratio. If the trunk duct and branch ducts are located in the spaces included in the calculations for } H_a \text{ and } H_b, \text{ a value of } 1.0 \text{ is suggested, since the heat loss from duct work is regained. For duct systems located in attic spaces or vented crawl spaces, a value of } 0.9 \text{ is suggested for insulated ducts.} \]

The application of the suggested equation to the actual conditions maintained in the Residence during Series C-1.5 when the outdoor air rate was equivalent to 1.33 air changes per hour shows that:

\[ H_t = \frac{31047 + 21300 + 15550}{(0.9) (1.0)} = 67897 \]

\[ = 75400 \text{ Btu per hr} \]

An inspection of the fuel consumption curve in Fig. 11 shows that the average fuel input rate for an 80 °F indoor-outdoor temperature difference was about 1310 cu ft per day. However, a curve drawn through the points representing the maximum deviations from the average curve gives a maximum gas consumption of about 1400 cu ft per day. This corresponds to an actual fuel input rate of about 58,000 Btu per hr, or 77 percent of the calculated input of 75,400 Btu per hr. The reserve of about 23 percent should prove to be adequate for all contingencies.

17. Indoor-Air Conditions

For each of the four main series, average room-air temperatures measured at four different elevations in the rooms were obtained over a wide range of weather conditions. The differences in temperature between those at the sitting level and those at the other three levels were separately plotted for each series. The room-air temperature observations were made at 7 a.m. and represented the maximum temperature differentials obtained during the day and hence the least favorable aspect of plant performance.(1) However, in comparing the performances obtained with different heating systems, the observations at 7 a.m. following a long period of night-time operation provided the best index of plant performance unaffected by extraneous heat gains. The typical curves of temperature differentials shown in Fig. 14 are similar in trend to those obtained in previous studies.(2) The differentials obtained for an indoor-outdoor temperature difference of 35 °F have been transferred to Table 6, item (1), as have similar values for the other

\[ \text{Fig. 14. Room-Air Temperature Differentials for Outdoor Air-Flow Rate of 1.33 Air Changes per Hour} \]
three series. A comparison of these values for the four series indicates that the introduction of outdoor air had little influence on the room-air temperatures.

However, increases in outdoor-air rates were accompanied by marked reductions in indoor relative humidity, as shown by item (2) in Table 6. The variations in indoor relative humidity over a wide range of indoor-outdoor temperature differences are clearly indicated in Fig. 15. Although the relative humidity of the outdoor air increases in magnitude as the temperature decreases, the humidity ratio, or the pounds of moisture per pound of dry air, decreases sharply. For example, for indoor-outdoor temperature differences of 30 F and 60 F, the corresponding humidity ratios of the outdoor air were observed to be 0.005 and 0.001 lb of moisture per lb of dry air. Normally in residences the indoor humidity ratio is higher than the outdoor ratio. Hence, any introduction of outdoor air into a structure, whether by normal infiltration or through an outdoor-air duct, will result in a displacement of comparatively moist indoor air by drier outdoor air. It can be expected that the indoor relative humidity will decrease as the indoor-outdoor temperature difference becomes greater and with an increase in the outdoor-air rate. These two trends are confirmed by the results shown in Fig. 15.

The fact that large decreases in indoor relative humidity were obtained when the outdoor-air rate was increased from 0 to 0.44 air changes and also from 0.44 to 0.89 air changes, whereas no further reduction was obtained when the air rate was increased from 0.89 to 1.33 air changes, is difficult to rationalize. As in the case of all previous measurements of relative humidity, the range of observed values for any given outdoor temperature is exceedingly large due to wide variations in outdoor humidity ratio and to effects of time lag; thus difficulty is experienced in sharply defining an average value. In any case, it is probable that the greatest change in relative humidity occurs with the first small increment of air change and that diminishing effects result from further increases in the outdoor-air change rate. It may be observed that when the larger rates of air change were used the indoor relative humidity during cold weather was lower than a minimum desired value of about 30 percent. The trends indicate an undesirably low relative humidity and the necessity for humidification in extremely cold weather.

In the case of the Residence, in which the occupancy consisted of two adults, the moisture input from such sources as cooking, washing, and bathing was not excessive. However, in some installations for which excessive relative humidities have been reported in extremely cold weather, Fig. 15 shows the ease with which reductions can be made in the relative humidity by the introduction of outdoor air. In a case of the latter sort, the use of an outdoor-air duct would appear to be a desirable addition to the forced warm-air heating system particularly when an adjustable air damper is provided. Attention should be called to the increased load caused by the introduction of outdoor air.

The effect on the “freshness” or quality of the indoor air produced by introducing large quantities of outdoor air was pertinent to this study. It is true that any filtered outdoor air would tend to dilute the smoke, odor, or dust concentration. In this connection, the equivalent air change due to normal infiltration with Series A-111 was about 0.95 air change per hr, whereas for Series C-1.5 the total equivalent air change due to infiltration and outdoor-air addition was about 2.28 per hr or about 2.4 times as much as for Series A-111. The necessity for adequate additions of outdoor air can arise periodically as a result of laundry drying, cooking of odorous foods, and tobacco smoking. Unfortunately, reliable measurements of odor and fume concentrations were not possible, so that only qualitative impressions by the occupants were noted. From the standpoint of the occupants the quality of the indoor air as judged by odor con-
centration was satisfactory when large ventilation rates were in use. However, no difference in the air quality was observed with reduced rates of outdoor air under the normal living conditions which prevailed in this study.

18. Summary of Studies with Outdoor-Air Supply

The demand for clean, fresh indoor air has been particularly great in the case of well-built, tight structures in which odors from cooking and smoking are not readily dispersed. With a forced-air system, outdoor air can be introduced readily into the return-air inlet of the furnace, cleaned by means of the air filter, tempered by the furnace, and positively discharged into the conditioned spaces. Available design data concerning the operation with outdoor-air supply have been based largely upon empirical assumptions, some of which have not been verified in this study.

One of the most interesting aspects of this investigation was that, with some wind directions, the indoor-air pressure was increased over the normal atmospheric pressure by wind action alone, even when no outdoor air was positively introduced. The magnitude of this pressure increase was found to be dependent upon both wind velocity and wind direction, as well as chimney draft effect. However, the introduction of as much as 1.53 air changes per hr of outdoor air gave practically no additional increase in indoor pressure. Since the wind velocity and wind direction did not remain constant, some variation was found in the outdoor-air supply rate. Although these variations were not unduly large for the Residence installation in which the intake was located in a window well, it might be advisable to locate the intake opening so that intake would be least influenced by the wind. In this connection, although no studies were conducted with other locations of the outdoor-air intake, the use of an attic intake would seem to have merit, not only because the wind effects might be minimized, but also because the entering air would be preheated to some extent.

No evidence was obtained that any appreciable reduction in normal infiltration resulted from the introduction of large rates of outdoor air. Approximations of the total area of openings in the Residence that could be assigned to window and door crackages as well as chimney openings revealed that even a well-built, tight structure is of relatively porous construction that permits ready dissipation of any air introduced from the outdoors into the structure. The use of outdoor air resulted in appreciable increases in fuel consumption. These increases were found to correlate with the calculated increases in heat loss when the heat loss for the entire structure was taken into account, but not when the heat losses for the first-story rooms alone were considered.

The introduction of outdoor air resulted in a marked reduction in the indoor relative humidity, in fact, under some conditions to an undesirably low value with no artificial humidification. It was not possible to devise any convincing procedure for determining odor and fume reductions or decreases in dust content. As far as indoor-air temperatures and automatic control of the burner and blower were concerned, no significant differences were experienced over the range of outdoor-air rates investigated.
V. PLANT PERFORMANCE WITH FURNACE-CLOSET INSTALLATION

19. Specifications for Furnace-Closet Gas-Fired Installations

In many homes the trend towards the allotment of minimum space for mechanical equipment has resulted in a furnace-closet installation in which the furnace is isolated from the remainder of the house. It is located in a closet that is barely large enough to accommodate the furnace. Some means must be provided to permit air to enter the closet so that the combustion process occurring within the furnace will not be hampered. Furthermore, since the heat dissipated from the casing, bonnet, and flue pipe will be confined to the small closet space, ventilation must be provided to prevent excessive temperatures within the space. Under extremely adverse circumstances, high closet temperatures may not only create a fire hazard but may also cause faulty operation of the electrical equipment and failure of the insulation on electrical wires.

The recommended installation method which will provide the necessary air for combustion and ventilation is shown in Fig. 16; this was based upon research(13) conducted by the American Gas Association Testing Laboratory. In most cases the closet is located so that normal infiltration air through window and door crackages in the house can enter the ventilating openings into the closet and thereby satisfy the requirements for combustion air.

As Davis(19) has indicated, “a different problem arises if the confining closet is so located that its upper and lower ventilating openings cannot communicate with freely inter-connected room areas which combined have an adequate infiltration rate. This situation might occur if the two openings from the closet were cut through into a small central hallway, which is possible of isolation from the rest of the house by the closing of doors. In such instances, the ‘American Standard for Installation of Gas Piping and Gas Appliances in Buildings,’(20) recommended the two openings be extended to a source of adequate outdoor air, both ducts always terminating in a region where they will be equally affected by wind pack pressure.”(19)

The detailed specifications from the American Standard Z21.30 mentioned by Davis are as follows:

3.15 Air for Combustion:

(c) Where appliances are installed in a confined space within a building of unusually tight construction, air for combustion and ventilation must be obtained from outdoors or from spaces freely communicating with the outdoors. (Crawl space or attic.) Under these conditions, the openings called for in Fig. 16 shall be replaced by two openings having a combined area of not less than one square inch per 1000 Btu per hr of input rating. One opening shall be near the top of the enclosure and one near the bottom. These openings shall be of approximately equal area and shall communicate with the selected source or sources of adequate air supply, by ducts. Where ducts are required, they shall be continuous and of the same cross-sectional area as the openings to which they connect. The minimum dimension of rectangular air ducts shall be not less than 3 in. Any duct from the top opening must be horizontal or pitched upward.(23)

Clearance requirements listed in American Standard Z21.30 are as follows:

Minimum clearances for listed Central Heating Boilers and Furnaces, Distance from Combustible Construction, in.

Above — 18 in. (A vertical clearance of 6 in. may be used with warm-air furnaces equipped with mechanical means to circulate the air and with an approved temperature limit control that cannot be set higher than 250 F.)

Front — 18 in.

Projecting Flue Box or Draft Hood — 6 in.

Listed central heating-furnaces shall be installed with clearances not less than specified (above), except that appliances listed for installations at lesser clearances may be installed in accordance with their listings.(29)

Pertinent requirements of the American Standards Association(21) concerning temperatures adjacent to the furnace casing, as well as flue gas temperatures, are as follows:

2.16.1 The maximum temperature rise at points 6 in. from the back and sides of a central furnace shall not exceed 90 F above the inlet air temperature after the furnace has been operated for one hour at 1.5 normal test pressure applied at the inlet of the gas pressure regulator.

2.16.2 The maximum temperature at points on the floor under forced air type furnaces shall be not more than 90 F in excess of the inlet air temperature. This requirement shall not apply to furnaces having downward warm air discharge provided they are marked “For Installation on Non-Combustible Floors Only.”(23)
The actual furnace-closet installation in the Residence was based largely on the recommendations given in the American Standard Z21.30. The details of the closet were previously described in Section 7 and Figs. 5 and 6.

20. Objectives and Procedure

For this investigation in which natural wind conditions would have a direct bearing on the performance of the blower-furnace unit, the inadequacies of the usual laboratory experimentation became apparent. It was considered necessary to conduct the investigation in an actual field installation in which all the factors that might affect the performance were naturally created and not artificially simulated. These experiments in the Residence, however, involved some limitations that should be appreciated. In the first place, since no control could be exercised over the outdoor temperatures, wind velocities and wind directions, it was necessary to accumulate large amounts of data over an extended period of time in order to determine the principal effects and possible causes. In the second place, since the laborious analysis could be made only for a single residence, the results were confined to the Residence and the peculiar geographical and atmospheric conditions that prevailed.

The specific objectives of the study designated as Series D and E were:
(a) To establish the validity of the requirements given in American Standard Z21.30 for ventilation opening areas and combustion opening areas
(b) To determine the air temperatures obtained in the closet, particularly at times of large heating demands
(c) To determine whether reductions in the recommended opening areas would result in adverse conditions of burner operation
(d) To compare the over-all performance of the furnace-closet installation with those of the same furnace installed in the open basement.

Two features of the Residence installation were not in conformity with usual installation practice. In the first place, the furnace closet was located in the basement instead of on the first floor. This location necessitated the use of greater heights for ducts V and CV, which increased the frictional resistance of air flow through the ducts but also increased the motive head for gravity circulation of heated air. However, the difference in elevations of the two ducts was about the same as that in a first-story closet installation. The effective chimney height was also greater so that the draft effect produced by the chimney on the closet space would be greater than that for a first-story closet installation and would tend to counteract the motive head of duct V. In the second place, all joints in the closet were sealed so that little possibility existed of air transfer between the closet space and the basement. It is probable that the pressures obtained in the closet space would be of greater magnitude than those for a closet which is not sealed against air leaks.

Two different rates of fuel input rates were maintained, namely 46,000 Btu per hr and 90,000 Btu per hr. The sizes of the ducts V, V-H, CV, and CV-H were each 7 in. square. The diameter of the flue connection to the furnace was 5 in. As shown in Tables 8, 9, and 10, a large number of combinations of opening areas for ducts V or V-H, ducts CV or CV-H, and flue pipe were investigated.

21. Air-Flow Rates in Vertical Ducts with Large Opening Areas

For the initial studies the opening areas shown in Fig. 5 were used. At the time that the studies were initiated, the specifications stipulated that each of the two opening areas were to provide 1 sq
### Table 8

Studies with Furnace-Closet Installation and Vertical Ducts V and CV

<table>
<thead>
<tr>
<th>Series</th>
<th>Duct V</th>
<th>Duct CV</th>
<th>Flue Pipe</th>
<th>Duct V Ventilation</th>
<th>Duct CV Ventilation</th>
<th>Closet Air Temp. from Ceiling, 1 in. from Ceiling (Based on In-out Diff. of 50°F)</th>
<th>Period of Observations</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1)</td>
<td>(2)</td>
<td>(3)</td>
<td>(4)</td>
<td>(5)</td>
<td>(6)</td>
<td>(7)</td>
<td>(8)</td>
</tr>
<tr>
<td>D-1</td>
<td>49.0</td>
<td>49.0</td>
<td>19.6</td>
<td>Upward Downward</td>
<td>Downward Downward</td>
<td>85</td>
<td>Jan. 21 to Feb. 18 and Apr. 3-9, 1950</td>
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<tr>
<td>D-2</td>
<td>49.0</td>
<td>49.0</td>
<td>12.6</td>
<td>Upward Unstable</td>
<td>Downward Downward</td>
<td>90</td>
<td>Feb. 18 to 23, 1950</td>
</tr>
<tr>
<td>D-3</td>
<td>24.5</td>
<td>24.5</td>
<td>12.6</td>
<td>Upward Unstable</td>
<td>Downward Downward</td>
<td>90</td>
<td>Feb. 23-29, 1950</td>
</tr>
<tr>
<td>D-13</td>
<td>24.5</td>
<td>5.3</td>
<td>12.6</td>
<td>Downward Downward</td>
<td>Downward Downward</td>
<td>50</td>
<td>Nov. 21 to Dec. 1, 1950</td>
</tr>
<tr>
<td>D-16</td>
<td>24.5</td>
<td>12.2</td>
<td>12.6</td>
<td>Upward Downward</td>
<td>Downward Downward</td>
<td>85</td>
<td>Dec. 2-18, 1950</td>
</tr>
<tr>
<td>D-17</td>
<td>24.5</td>
<td>12.2</td>
<td>12.6</td>
<td>Upward Downward</td>
<td>Downward Downward</td>
<td>90</td>
<td>Feb. 26-28, 1950</td>
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<td>D-5</td>
<td>12.2</td>
<td>0.0</td>
<td>12.6</td>
<td>Downward Downward</td>
<td>Downward Downward</td>
<td>85</td>
<td>Dec. 14-17, 1950</td>
</tr>
<tr>
<td>D-20</td>
<td>5.3</td>
<td>0.0</td>
<td>12.6</td>
<td>Downward Downward</td>
<td>Downward Downward</td>
<td>60</td>
<td>Feb. 28-Mar. 2, 1950</td>
</tr>
<tr>
<td>D-18</td>
<td>12.2</td>
<td>12.2</td>
<td>12.6</td>
<td>Downward Downward</td>
<td>Downward Downward</td>
<td>60</td>
<td>Jan. 6-23, 1951</td>
</tr>
<tr>
<td>D-19</td>
<td>0.0</td>
<td>5.3</td>
<td>12.6</td>
<td>Downward Downward</td>
<td>Downward Downward</td>
<td>60</td>
<td>Dec. 29-31, 1950</td>
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### Table 9

Studies with Furnace-Closet Installation and Horizontal Ducts V-H and CV-H

<table>
<thead>
<tr>
<th>Series</th>
<th>Duct V-H</th>
<th>Duct CV-H</th>
<th>Flue Pipe</th>
<th>Duct V-H Ventilation</th>
<th>Duct CV-H Ventilation</th>
<th>Closet Air Temp. 1 in. from Ceiling (Based on In-out Diff. of 50°F)</th>
<th>Period of Observations</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1)</td>
<td>(2)</td>
<td>(3)</td>
<td>(4)</td>
<td>(5)</td>
<td>(6)</td>
<td>(7)</td>
<td>(8)</td>
</tr>
<tr>
<td>E-7</td>
<td>49.0</td>
<td>49.0</td>
<td>12.6</td>
<td>Outward Inward</td>
<td>Inward</td>
<td>85 F</td>
<td>Mar. 30-Apr. 9</td>
</tr>
</tbody>
</table>

### Table 10

Results of Special Studies with Blocked Flue in Furnace-Closet Installation

<table>
<thead>
<tr>
<th>Series</th>
<th>Fuel Rate, Btu per hr</th>
<th>Operating Time, Continuous Min.</th>
<th>Ave. CO₂ in Air, %</th>
<th>Max. Flue Gh, f</th>
<th>Flue Temp., F</th>
<th>Ceiling Surface Temp., F</th>
<th>Air in Duct Temp., F</th>
<th>Air in Duct Temp., F</th>
<th>Air in Duct Temp., F</th>
<th>Air in Duct Temp., F</th>
<th>Air in Duct Temp., F</th>
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<tbody>
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<td>46000</td>
<td>21.5</td>
<td>8.9</td>
<td>376</td>
<td>277</td>
<td>48</td>
<td>151</td>
<td>142</td>
<td>61</td>
<td>56</td>
<td>56</td>
</tr>
<tr>
<td>2</td>
<td>46000</td>
<td>24.5</td>
<td>8.9</td>
<td>376</td>
<td>285</td>
<td>30</td>
<td>167</td>
<td>160</td>
<td>65</td>
<td>56</td>
<td>56</td>
</tr>
<tr>
<td>3</td>
<td>46000</td>
<td>5.3</td>
<td>8.9</td>
<td>376</td>
<td>285</td>
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<td>167</td>
<td>160</td>
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<tr>
<td>4</td>
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<td>8.9</td>
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<td>13</td>
<td>147</td>
<td>147</td>
<td>147</td>
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<tr>
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<td>8.9</td>
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<td>152</td>
<td>152</td>
<td>152</td>
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</tbody>
</table>

* Note: *CO₂* indicates that the CO₂ content of the flue gas at the flue outlet.

### Notes
- Upward flow from closest to attic; downward flow from attic to closest.
- Air flow rates were 340 cfm and 670 cfm, respectively, for fuel input rates of 46,000 Btu per hr and 90,000 Btu per hr. In all cases the flue pipe carried products of combustion out of the residence.
- Air flow rates were 340 cfm and 670 cfm, respectively, for fuel input rates of 46,000 Btu per hr and 90,000 Btu per hr. In all cases the flue pipe carried products of combustion out of residence.
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in. of free area per 1000 Btu per hr of input. Hence, for a 46,000 Btu per hr input, a 7-in. square duct was applicable. The ventilation duct V was intended to convey the heated air from the closet and discharge it into the attic. On the other hand, duct CV was intended to convey cool attic air into the lower part of the closet, at which point part of the air was intended for the combustion process and the remainder for ventilation through duct V and the draft hood. The air required for combustion alone could be calculated for the known input rate and CO₂ content of the flue gas at the flue outlet. Combustion data[26] indicated that a CO₂ content of 5.5 percent in the flue gas corresponded to an air requirement of 33 lb of air per lb of natural gas. For a caloric value of 22,710 Btu per lb of gas,
this amounted to 1.452 lb of air per 1000 Btu input. For an air density of 0.075 lb per cu ft, the air-flow rate was about 0.323 cfm per 1000 Btu per hr input. Since the fuel input rate was 46,000 Btu per hr, the total air-flow rate for combustion purposes only was estimated to be approximately 15 cfm. Dilution of the flue gas as a result of air entering the draft hood would increase this minimum air requirement.

It was not possible to predetermine the flow rate of the ventilation air that might leave the closet space through either duct V or the draft hood. In any case, it was known that the air-flow rates would be small and that the frictional resistances in the 7-in. square ducts would also be small. It was anticipated, therefore, that the small magnitudes of the air velocities, the differential pressures between closet space and attic, and the differential pressures between closet space and first-story rooms would require unusually sensitive instruments.

What was not anticipated, however, was the direction of the air flow in ducts V and CV. Duct CV was intended to convey air downwards from the attic to the closet and duct V to convey air upwards. In actual operation both ducts served to supply air downwards from the attic to the closet. For example, as shown in Fig. 17 for an indoor-outdoor temperature difference of 35 F, duct V supplied about 27 cfm and duct CV about 33 cfm, or a total of about 60 cfm. For an indoor-outdoor temperature difference of 55 F, the corresponding air supply rates were about 28 cfm and 47 cfm or a total of about 75 cfm. These magnitudes were greatly in excess of the minimum value of 15 cfm required for combustion air alone, and gave indications that ducts V and CV were more than ample in size.

A wide deviation from average flow rates was obtained during instantaneous observations as indicated by the scatter of points plotted in Fig. 17. The deviations were caused primarily by variations in wind velocity, as typified by the record for April 3, 1950, in Fig. 18. At 6:30 p.m. when the average wind velocity was only 6 mph, the flow rates for ducts V and CV were 18 and 24 cfm, respectively. Two hours later when the average wind velocity had suddenly increased to about 20 mph, with momentary gusts as high as 30 mph, the flow rates for each of the ducts V and CV increased to over 50 cfm.

Although duct V was operating in a manner contrary to the intended purpose and was supplying air to the closet instead of exhausting air from it, the over-all operation was satisfactory since the combustion process was not impaired. At this stage of the investigation, it became apparent that the draft effect of the chimney was quite large, as evidenced by the fact that the flow in ducts V and CV was downward from the attic and that large flow rates to the closet were obtained. An analysis indicated that reductions in areas of openings to the closet could be made.

22. Air-Flow Rates in Vertical Ducts with Reduced Opening Areas

The next phase of this investigation was devoted to successive reductions in the areas of flue pipe, duct V, and duct CV, as shown in Table 8. By inserting a strip of metal in the flue pipe the effective opening area was reduced to the equivalent of a 4-in. diam flue pipe. This reduced area was used in all the studies designated by Series D-2 through D-20, for which the fuel input rate was 46,000 Btu per hr.

In Series D-2, for which no changes were made in the sizes of ducts V and CV, an interesting phenomenon was observed which has been designated (Column 6) as an "unstable" direction of flow. Under normal operation both ducts V and CV served to conduct air from the attic downwards to the furnace closet. Occasionally, however, and particularly when long burner operations occurred, the flow in duct V was suddenly reversed in direction, so that air entered the closet through duct CV but escaped from the closet through both duct V and the flue pipe. This action is similar to that some-
times experienced in a gravity, warm-air system in which the flow in a leader pipe may be reversed in direction, particularly if the return-air duct system is inadequate.

A similar, but even more striking example of this unstable direction of flow occurred several times with Series D-3, for which ducts V and CV were each reduced in area by 50 percent. A com-
A complete record of the pertinent data is given in Fig. 19. At noon on Feb. 25, 1950, the wind velocity was about 5 mph; duct V was conducting air at a rate of 10 cfm upwards out of the closet although the pressure in the closet was 0.004 in. less than that in the attic. The stack effect in duct V was apparently sufficient to cause an upflow. The phenomenon shown is similar to that in a gravity warm-air heating system, for which an upflow through the warm-air stacks accompanies a slight sub-atmospheric pressure in the furnace casing. At 1:20 p.m. the wind velocity had increased slightly; this increased the chimney draft effect, and duct V was then conducting air into the closet at a rate of 17 cfm even though the pressure in the closet was 0.004 in. greater than that in the attic. The pressure increase in the closet can be partly attributed to static pressure regain from the velocity pressure of the air flowing in the two ducts. During the next 16 hr the pressure differential between closet and attic increased to 0.010 in.; the air-flow rates into the closet also increased; and the closet-air temperatures at the ceiling near the opening of duct V slowly increased from about 78 F to 98 F. At 5:45 a.m. an initial break occurred in which the downward flow through duct V decreased in magnitude; at the same time the stack effect in duct V was increasing and the wind velocity and resulting chimney draft effects were diminishing. A final decisive break occurred at 6:15 a.m. when the flow through duct V was again upward from the closet to the attic.
The marked effect of chimney draft effect on closet pressure is indicated in Fig. 20, which shows a typical record of the pressure differential between closet and attic for Series D-5. In this series the only connection to the closet consisted of duct V and the flue pipe. Both connections were restricted in area and had approximately the same opening areas. The chimney draft effect was sufficiently large so that the pressure in the closet was less than that in the attic, and each gust of wind was reflected in a greater pressure differential. The resistance to air flow of duct V was great, and the closet was tightly constructed so that the pressure in the closet was also less than that indoors. The conclusion was reached that the pressure in the closet was dependent largely upon the magnitude of the chimney draft, which in turn was dependent upon wind direction, wind intensity, and length of burner operation.

When the opening area of duct V was increased to approximately twice that of the flue pipe opening, Series D-4, the resistance to air flow in duct V was markedly reduced. The resulting closet pressure was only slightly lower than attic pressure, and was practically the same as that indoors.

For all variations of ducts V and CV and flue pipe openings, no difficulties were experienced as far as burner operation was concerned, even for the extreme cases of Series D-19 and D-20 listed in Table 8 for which the total opening area was only 0.1 sq in. per 1000 Btu per hr of input. However, it may not be desirable to have widely fluctuating pressures in the closet, as were observed in Series D-5 and D-20 when the single duct V was provided with an opening area equal to or smaller than that of the flue pipe.

From the results of these studies, it is not possible to give quantitative values of pressures and flow rates that can be expected to recur in this or any other installation. Nevertheless, a qualitative analysis of the various influences affecting the pressures and flow rates will be attempted. The tightly constructed furnace closet can be considered as a plenum chamber to which are attached three openings, namely duct CV, duct V, and the flue pipe. At any given time, the pressures, both positive and negative, of all three connections are acting upon the closet space. Also, the sizes of the opening areas and the duct lengths affect the resistance to passage of air or flue gas products. Specifically, the following pressures can be considered as acting simultaneously on the closet and the connecting ducts:

(a) A potential-stack effect exists in the duct V, because of the temperature difference between the warm air near the ceiling of the closet and the cool outdoor air. The temperature of the air near the closet ceiling increases with prolonged operations of the burner, and thereby increases the potential stack effect.

(b) A relatively large negative pressure, or draft effect, is produced at the opening of the draft hood. This draft effect increases with higher flue gas temperatures and wind velocities.

(c) A pressure of relatively small magnitude is created in the attic by wind action.

(d) The relative effects of these pressures are dependent also upon the opening areas for the three connections.

Apparently, the direction of air flow in duct V is dependent upon the magnitude of the draft effect on the closet and the stack effect in duct V. When the draft effect is large as compared with the stack effect in duct V, the air will flow from the attic to the closet; conversely, when the draft effect is small, the air may flow from the closet to the attic.

23. Closet-Air Temperatures with Vertical Ducts

Temperature measurements were made of the closet air at several elevations over a range of weather conditions. The air temperatures in the lower part of the closet decreased as the outdoor temperature decreased, since the incoming air through duct CV was taken from the unheated attic. On the other hand, the closet ceiling-air temperature increased with colder weather as a result of longer burner operations. The maximum temperature observed was about 100.5 F for an outdoor temperature of -6.0 F, as shown in Fig. 21. Little difference in maximum temperatures was noted between the various series for an indoor-outdoor temperature difference of 50 F (Table 8, Column 9). With the exception of Series D-2 and D-3, ducts V and CV were delivering cool attic air to the closet and the flue pipe was venting the closet. Even for Series D-2 and D-3, whenever the closet ceiling-air temperature approached a value near 100 F, duct V would become "primed" and the warm air in the closet would escape through both duct V and the flue pipe. The maximum air temperature of about 100 F was not considered hazardous.
24. Performance of Furnace and Blower

Performance curves for the furnace and blower are shown in Fig. 22. No distinction could be made between the various series. As may be noted from the curve representing the "total hours of operation per day" for the burner, the fuel input rate selected for this series was barely sufficient to take care of the maximum load represented by design weather conditions. In this connection, the fuel input rate of 46,000 Btu per hr was originally selected for Series A-111 and was found to provide an adequate reserve capacity. However, in the furnace-closet investigation, the fuel consumption was about 7 percent greater than when the furnace was not enclosed in a closet, and thereby the reserve capacity was reduced. The increase in fuel consumption can be accounted for by the fact that the heat loss from the casing, bonnet, and that portion of the flue pipe within the closet was not regained in the house; instead it heated the closet air which was vented by the flue pipe or duct V.

25. Performance with Horizontal Ducts

A similar study of air-flow rates and closet-air temperatures was made with two 7-in. square horizontal ducts, V-H and CV-H, located as shown in Figs. 5 and 6. A summary of the observed performance is given in Table 9. In each of the seven different arrangements tested, the actual flow of air in both ducts was from the outdoors to the closet. In other words, the chimney draft effect was predominant over any slight stack effect of duct V-H. The closet ceiling-air temperature was not greater than 85°F for an indoor-outdoor temperature difference of 50°F. Hence, even under design weather conditions the ceiling-air temperature should not present any hazard.

A record of the air-flow rates revealed that each of the two ducts carried from 20 cfm to 85 cfm, depending to a large extent upon the direction and intensity of the wind, and to some extent on the weather. In general, the air-flow rates increased with severity of weather, since the chimney draft which acted upon the closet also increased. Satisfactory operation from the standpoint of venting action and low closet-air temperatures was obtained for all of the tests made under normal operation, even with combined free-areas of the ducts as small as ¼ sq in. per 1000 Btu per hr of input.

26. Blocked Flue Studies

The most drastic operating condition which can be imposed upon a furnace located in a tight closet is one in which the flue is blocked and all of the combustion products are spilled into the closet. In this extreme condition, ducts CV, or CV-H, would be required to supply outdoor air for combustion, and ducts V, or V-H, would be required to serve as vents for the combustion products. An analysis indicated that the spillage of the combustion gases into the closet would contaminate the air supply and greatly alter the normal combustion process,
perhaps to the extent of reducing the oxygen supply and extinguishing the flame. For the purpose of exploring the operation under blocked flue conditions, a large number of arrangements were investigated, as shown in Table 10.

Three different opening areas for the vertical ducts V and CV were studied with a fuel input rate of 46,000 Btu per hr, as indicated by special Series 1, 2, and 3. No flame failure occurred after more than 60 min at continuous burner operation, even with 5.3 sq in. duct opening areas corresponding to about $\frac{1}{8}$ sq in. per 1000 Btu per hr input for each duct. The low temperatures existing in duct CV indicated that the duct supplied attic air to the closet, while the high temperatures in duct V showed that the duct was serving as a flue. A similar series of tests with 90,000 Btu per hr input also gave no evidence of flame failure for small opening areas corresponding to about $\frac{1}{8}$ sq in. per 1000 Btu per hr in input for each duct.

The use of the smallest opening areas of ducts V and CV gave evidence that some parts of the ceiling surface would reach unacceptably high temperatures. In American Standard Z21.30, a maximum temperature rise for combustible materials of only 90 F above normal room temperature is stated as the limit of acceptability, although experimental evidence to support this low value seems to be lacking. At any rate, for a room-air temperature of 70 F, this rise corresponds to a surface temperature of about 160 F. Ceiling-surface temperatures in excess of 160 F were obtained with Series 6, at the same time that the ceiling-air temperatures had attained a value of 231 F. With the use of opening areas of ducts V and CV, each corresponding to about $\frac{1}{2}$ sq in. per 1000 Btu per hr input (Series 1 and 4), surface temperatures lower than 160 F were obtained. Hence, the conclusion was reached that with vertical ducts, the required opening areas of ducts V and CV should each be at least $\frac{1}{2}$ sq in. per 1000 Btu per hr input.

Three interesting tests were made with horizontal ducts V-H and CV-H, listed as Series 9, 8, and 7. No difficulty was experienced with Series 9 in which the opening areas were each about 1 sq in. per 1000 Btu per hr input. A graphical log of observed temperatures at a number of stations within the closet is shown in Fig. 23. The closet ceiling surface (S-9) immediately above the draft hood reached 160 F at the end of almost 2 hr of continuous burner operation. The average CO$_2$ content of the flue gas of 5.6 percent was the same as that experienced under normal venting conditions, indicating no essential change in combustion.

However, for both Series 8 and 7 in which the opening areas of ducts V-H and CV-H were considerably less than 1 sq in. per 1000 Btu per hr input, the CO$_2$ content of the flue gas increased and flame failure occurred. Apparently, the products of combustion which spilled into the closet could not be vented rapidly enough so that the normal oxygen content was no longer available at the level of the combustion air intake to the burner. Although the flame failure should cause an automatic shutdown of the main gas valve, from the standpoint of good
Fig. 23. Temperatures Obtained in Furnace-Closet Installation with Blocked Flue

Notes:
- Fuel Input Rate: 40,000 Btu/hr
- Average CO Content of Flue Gas: 5.6%
- Duct CV-H: 100% Open (7" x 7") 49 sq. in.
- Duct V-H: 100% Open (7" x 7") 49 sq. in.
- Test Conducted: 3-29-51
ventilation practice it seems desirable to avoid any condition leading to flame failure. Hence, the conclusion was reached that with the use of horizontal ducts, the required opening areas of ducts V-H and CV-H should each be at least 1 sq in. per 1000 Btu per hr input.

27. Summary of Observations with Furnace-Closet Installation

In those cases in which the furnace is enclosed in a small tight closet, care must be taken to provide for those contingencies that might adversely affect the performance of the burner. Three basic requirements must be fulfilled, as follows:

(a) Adequate combustion air must be provided
(b) Any potential fire hazard must not exist
(c) Provision must be made for rare emergency conditions resulting from a blocked flue.

For the purpose of satisfying these basic requirements, American Standard Z21.30 specifications provide for two ducts, in addition to the flue pipe, for connecting the closet to the outdoors or attic. One duct is intended to deliver outdoor air to the closet and the other to vent the closet air to the outdoors. The specifications require that the two openings shall have a combined area of not less than 1 sq in. per 1000 Btu per hr of input rating. One opening is to be near the top of the closet and the other near the bottom.

When two vertical ducts, V and CV, were used, no difficulties were experienced even with combined areas as small as 0.1 sq in. per 1000 Btu per hr of input as long as the flue was not blocked. It is true that duct V did not always function in the intended manner and ordinarily the air flow was from the attic to the closet. That is, the chimney draft effect was predominant over the stack effect of duct V, so that the flow of flue gas products and dilution air was always up the chimney while the flow in duct V was downward. Under some conditions, when the closet ceiling-air temperature reached a sufficiently high value, the stack effect of duct V was apparently strong enough to cause an upward flow of heated closet air to the attic. In any case, no evidence existed of any change in the CO₂ content of the flue gas which might result from a spillage of the combustion gases into the closet. Furthermore, the maximum air temperature in the closet did not exceed about 101 F, which is well below the limits for fire safety.

When the flue was blocked and the combustion gases spilled into the closet to be ventilated upwards into the attic through duct V, no flame failure was observed even with combined areas of ducts V and CV as low as ½ sq in. per 1000 Btu per hr input. From the standpoint of maximum air temperatures in the closet, however, slightly larger opening areas were found to be desirable. The conclusion was reached that ½ sq in. of opening area per 1000 Btu per hr of input rating should be provided for each of the two vertical ducts.

Similar studies with two horizontal ducts, V-H and CV-H, indicated that the most critical conditions were with a blocked flue. When the combined opening area of the two ducts was 1 sq in. per 1000 Btu per hr of input, flame failure was experienced at the end of about 1 hr of continuous operation. No difficulties were encountered with a combined area of 2 sq in. per 1000 Btu per hr of input. The conclusion was drawn that 1 sq in. of opening area per 1000 Btu per hr of input rating should be provided for each of the two horizontal ducts.

The fact that heat losses from the furnace casing, bonnet, and flue pipe were vented from the structure and were not available as a heat regain to the structure resulted in slightly greater fuel consumptions when the furnace was enclosed in a closet than when the furnace was not enclosed.
VI. EFFECT OF EXHAUST FAN ON FLUE ACTION

When large quantities of air are discharged from a tight house by means of a kitchen exhaust fan, the normal venting action of the furnace flue pipe may be disturbed to the extent that spillage of combustion products would occur at the draft hood. A study was conducted with from 100 cfm to 350 cfm of air discharged from the Residence by means of a special exhaust fan located in the basement. In these studies all windows and outside doors remained closed, but the inside door leading to the basement was open, as well as the door to the furnace closet.

The procedure in these studies was to operate the burner and blower continuously until stable temperature conditions were obtained at a time when practically no wind existed. The fuel input rate was 46,000 Btu per hr. The exhaust fan was started only when chimney venting action had been well established, after which the flow rate through the exhaust duct and the draft at the base of the chimney were measured. A thermocouple grid was located in the skirt of the draft hood for determining when spillage of flue gas products took place; a temperature substantially higher than basement air temperature resulted, indicating a reversal of chimney venting action and the spillage of combustion products into the basement. The exhaust fan was then shut off and stabilized temperature conditions were again obtained before repeating the test with a different rate of air flow.

The results obtained from two series of studies are shown in Fig. 24. When no outdoor air was admitted into the return-air duct system, spillage occurred when the exhaust air rate was greater than 182 cfm and the draft at the base of the chimney was less than 0.037 in. of water. The exhaust rate of 182 cfm corresponded to 0.64 air changes per hour, based on the total volume of first-story rooms and basement, or was 12 times greater than the theoretical requirements for combustion air.

In the second series, 120 cfm of outdoor air, or 0.42 air change per hr for the total house volume, was introduced into the return air duct system at the same time that basement air was being exhausted. In this case, spillage occurred with an exhaust rate of 300 cfm, corresponding to an exhaust rate which was exactly 180 cfm greater than the rate of introduction of outdoor air. With the exhaust fan operating, as in the case of the first series, the spillage was found to occur when the chimney draft was less than 0.037 in. of water.

It is not possible from these studies to make generalizations that would be applicable to other installations. The fact remains, however, that the exhaust action of a fan can lead to a reversal of chimney venting action and to spillage of combustion products into the house. Apparently, in each house a critical value exists for the chimney draft, below which spillage can occur, and above which
normal chimney venting action takes place. Assuming that a critical value of draft exists for each separate installation, theoretical considerations lead to the conclusion that spillage would be obtained more readily under any or all of the following conditions:

(a) Under mild weather conditions rather than cold
(b) Under calm outdoor conditions rather than windy
(c) With a low chimney rather than with a high chimney
(d) In a tightly built house rather than in a leaky structure
(e) With an exhaust fan having a large delivery rather than a low delivery
(f) With a cold chimney rather than with a heated chimney

In any given installation if spillage is observed, either the delivery of the exhaust fan should be reduced by dampering, or outdoor air should be introduced directly into the rooms or through the return-air duct of the forced-air heating system.
APPENDIX: MEASUREMENT OF INDOOR-OUTDOOR PRESSURE DIFFERENTIAL

Preliminary observations of the outdoor static pressure made by means of an open-end vertical tube located close to the ground and at a considerable distance from the Residence indicated that velocity components of the wind could not be readily isolated from the static pressure. Following a number of modifications in the static pressure tube, the instrument shown in Fig. 25 was developed and used. The tube containing the eight static pressure holes was free to rotate with the wind. In all important details, the construction was based upon the recommended specifications for Pitot tubes as given by the National Association of Fan Manufacturers.\(^2\) In the studies made during the 1949-50 season, the instrument was mounted on the peak of the Residence roof, about 3 ft 7\(\frac{1}{2}\) in. from the west edge of the peak, and with the tube located 3 ft 5 in. above the peak. Since some question existed whether the instrument was located in a “pressure” or “vacuum” region, depending upon the wind action adjacent to the roof, the instrument height was raised to 6 ft 2 in. above the peak. This outdoor pressure tube was connected to one side of the two draft gauges located in the basement. The indoor static pressure was measured by means of an open-end 1/8-in. pipe inserted through the floor at the northwest corner of the living room. This pipe was connected to the other side of the two draft gauges.

The instantaneous values of the indoor-outdoor pressure differential were measured by means of an inclined draft gauge. For the purpose of obtaining a continuous record of the pressure differential, a draft recorder was also used. A typical record for a 24-hr period was shown in Fig. 8.

In normal operation a time lag of a few seconds duration was observed before a change in wind velocity was reflected in a change of the indoor-outdoor pressure differential.

![Fig. 25. Outdoor Static Pressure Tube](image-url)
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