HEAT TRANSFER THROUGH AN AIR CURTAIN

Thesis for the Degree of M. S.
MICHIGAN STATE UNIVERSITY

Gad Hetsroni

1960

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This is to certify that the

thesis entitled

"Heat Transfer Through an Air Curtain"

presented by

Gad Hetsroni

has been accepted towards fulfillment
of the requirements for

M.S. degree in Agricultural Engineering

Carl W. Hall
Major professor

Date 9/9/60

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by

GAD HETSRONI

AN ABSTRACT

Submitted to the College of Agriculture and College of Engineering of Michigan State University of Agriculture and Applied Sciences in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

Department of Agricultural Engineering

Approved by Carl W. Hall 9/26/60

September 1960

ABSTRACT

The objective of this investigation was to determine the heat transfer characteristics of a vertical air stream, considering (1) the effect of air velocity and (2) the effect of the depth of the air stream.

An air curtain model was constructed in an opening between two constant temperature chambers. A temperature gradient across the curtain was maintained and the heat quantity transfered through the curtain was measured for various depths of air stream and for various air velocities.

The over-all heat transfer coefficient was calculated and plotted versus the Reynolds number of the air stream.

An equivalent heat transfer coefficient for unit depth of air stream was suggested.

A dimensionless ratio characterizing the heat transfer through a vertical air stream was formed. This ratio, denoted by H, was plotted versus Reynolds number of the flow. A formula describing the curve thus obtained was determined.

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CONTENTS

SUMMARY

An air curtain was installed in an opening between two insulated chambers. The air flow pattern and the heat transfer-mechanism through the curtain were determined for various air velocities and depth with the air moving vertically from top to bottom of the chamber.

The profile of the air flow is approximately parabolic by nature. As the air stream progresses away from the outlet it becomes wider and the profile of velocity distribution becomes flatter. The Reynolds number of the flow, from outlet to inlet grille, does not change considerably.

For air velocities of about 200 fpm to 700 fpm the over-all heat transfer coefficient is proportional to the air velocity and inversely proportional to the depth of the curtain. The values encountered in this investigation ranged from 5.0 to 16.0 Btu per hr. sq. ft. ^{o}F.

The ratio of over-all heat transfer coefficient to an equivalent heat transfer coefficient for a certain depth was denoted by H and plotted on semi-logarithmic paper as a function of Reynolds number.

The formula to fit this curve is

$$H = 0.5\ e^{-1.8/Re \times 10^{-4}}$$

This curve is asymptotic to the value of H=0.5, with a very little variation in H for $Re > 12 \times 10^{4}$.

LIST OF FIGURES

INTRODUCTION

Cold storage and work areas should be as accessible as possible to workers. Entrances to storage and work areas are necessary, however, and a side hung swinging door is customarily used. Frequent openings of the conventional door cause air changes inside the storage area, which introduce refrigeration losses and load the compressors.

Recently there has been much interest in the use of air curtains, or a stream of air as a means of thermally closing entry ways while allowing doors to remain completely unrestricted for workers to enter and leave as appropriate.

The conventional air curtain consists of a vertical downward flow of air across an opening, providing a barrier which can be penetrated by solid bodies only, but not by insects.

Little data are available on the heat transfer characteristics of air curtains. Since such information is essential in design of this kind of closure, the objective of this investigation was to measure the heat transfer characteristics of air curtains, considering (1) the effect of air velocity and (2) the effect of the depth of the air stream.

REVIEW OF LITERATURE

Large refrigeration losses are often encountered from air changes in cold storage rooms, or air conditioned areas. The number of air changes per day in such places depends of the volume considered and to a large extent on the number of door openings per day. Wherever heavy traffic is encountered, particularly entrances that require opening and closing of heavy doors, these losses might be very costly. Large losses may prevent the refrigeration unit from pulling down the extra load and from maintaining a uniform and constant temperature inside the room.

Williams (1958) studied refrigeration losses through the door openings and conveyor passes into cold rooms in a dairy plant. Under normal operation conditions for a 38°F room, losses through a 42 in. x 78 in. door ranged from 722 Btu to 951 Btu per min. Losses through conveyor passes ranged from 145 Btu per min. if the pass was located 30 in. above the floor, to 200 Btu per min. for passes located 10 in. above the floor.

When a nylon curtain was used instead of the conventional door, temperatures in the cold storage room were maintained constant but 70 percent increase in compressor's operation was required. His observations were made using a diverted air flow technique in which air from a blower was directed down across the door opening at 900 ft. per min. The air was recirculated through a false floor and suction fan. Refrigeration losses through the open door were reduced from about 950 Btu per min. to 200 Btu per min. A 20 percent reduction in compressor operating time was observed when the door was closed at night and the diverted air flow technique employed during the day.

The concept of an air curtain is not new by any means. A patent for a device to seal an entrance from outside weather by means of air flow, was issued to T. Van Kennel in 1904 by U.S. Patent No. 774,730 November 8, 1904 (Norton, 1959). The Van Kennel air curtain consisted of air streams introduced from both sides of an entrance from nozzles to counteract wind forces blowing through the entrance. Since then the idea has been successfully adopted in Switzerland and England. Since 1952 air curtains have been used in the United States-mainly in department stores and banks. There is an air curtain 41 ft. wide in Reno, Nevada, and there is an air curtain 18 ft. high in Lynn, Massachusetts. A more recent use is an 89 ft. wide entrance closure of Pan American's air terminal at Idlewild Airport, New York, as described by Norton (1959).

Gygax (1957) presents the principles of the air curtain as follows:

> A series of nozzles above the entrance discharges layers of air downward to an air return grille in the

floor. The air is returned to the top nozzles through ductwork which contains filters, heating and cooling elements, and blowers. This moving curtain of air has sufficient depth to protect and insulate the store interior from outside temperature, dust, dirt, rain, snow, debris, insects and unwanted animals. Persons passing through the air curtain feel a gentle breeze and a pleasant sensation of warmth in the winter and coolness in the summer.

The advantages of open entrances sealed by an air curtain include: saving valuable floor space near entrance, eliminating door maintenance and repairs, eliminating door accidents, and reducing heating or cooling losses where heavy traffic is encountered, particularly in cold storage rooms and walk-in freezers which normally require heavy doors.

There is only very little technical data presented in the literature concerning the air velocity, required thickness of the curtain and temperature of air.

Cadiergues (1957) suggested a limit of 2,000 ft. per min. at head level. Air temperature at this level should be around 88°F. The discharge opening should preferably be a long slot with the duct having the same cross section throughout. The cross sectional area of the duct should not be less than 2.5 times that of the slot. The return grille should be about three to four times the width of the discharge slot. The discharge air volume may vary from 1,500 to 15,000 cfm per linear foot of opening. Some commercial systems use 1,000 cfm per linear foot of opening and a higher air velocity (believed to be one ft. deep).

No data concerning the heat transfer pattern and the insulation value of an air curtain were found in the literature. To the knowledge of the author, there is nothing in the literature to suggest that research work with the purpose of the present investigation has been previously undertaken.

EQUIPMENT

An air curtain model, supplied by the American Air Curtain Co., was used in this investigation. The air is forced through an adjustable distributor forming six nozzles (Figure 1). An air curtain is formed which has a vertical cross-section of 7.75 sq. ft. (32.5 in. wide and 34.3 in. high). A plastic sheet divided into grids covers one edge of the curtain.

The distributor is operatively connected by means of a servo motor to a control mechanism. The flow of air forming the air curtain is produced by four fans driven by four 1/15 hp electric motors. Two fans draw the air from the inlet chamber and deliver it through a duct to the outlet chamber. The two other fans are located above the outlet chamber and accelerate the air into it. Thus, the air flows in a closed circuit.

The air velocity of the curtain is controlled by the speed of the fans and can be varied by means of an autotransformer (Variac) in the electrical circuit.

The curtain was operated with the distributor directing the air in a vertical stream.

The air curtain model was placed between two insulated chambers. The one on the right in Figure 1 is a cold storage room normally maintained at 40°F by a F-12

Figure 1. Experimental set-up used for the investigation.

refrigeration unit driven by $\frac{1}{2}$ hp electric motor. On the other side a 2 ft. x 2 ft. x 2 ft. box, constructed of $\frac{1}{2}$ in. plywood with 6 in. redwood fiber insulation, was maintained at 85°F by a 1310 watt, 230 volt electric heating mat.

Instrumentation - a) The operation time of the refrigeration unit was recorded by an operation recorder and measured with an electric clock connected in parallel with the compressor.

b) The heating time and energy used by heating mat were recorded and measured by means of an operation recorder and watt-hour meter.

c) The temperatures at various locations were sensed with copper-constantan (1938 calibration: 24 B & S gauge) thermocouples, and recorded every hour by a recording potentiometer. An automatic switch was used on this recorder to enable the use of 32 thermocouples and to operate the recorder every hour. The thermocouples and recorder were checked for proper wiring and operation prior to measurements. The location of the thermocouples is shown in Figure 2.

d) The speed of the fans was controlled by a Variac. The voltage was measured with a Simpson meter and the speed of the fan was measured with a Strobotac.

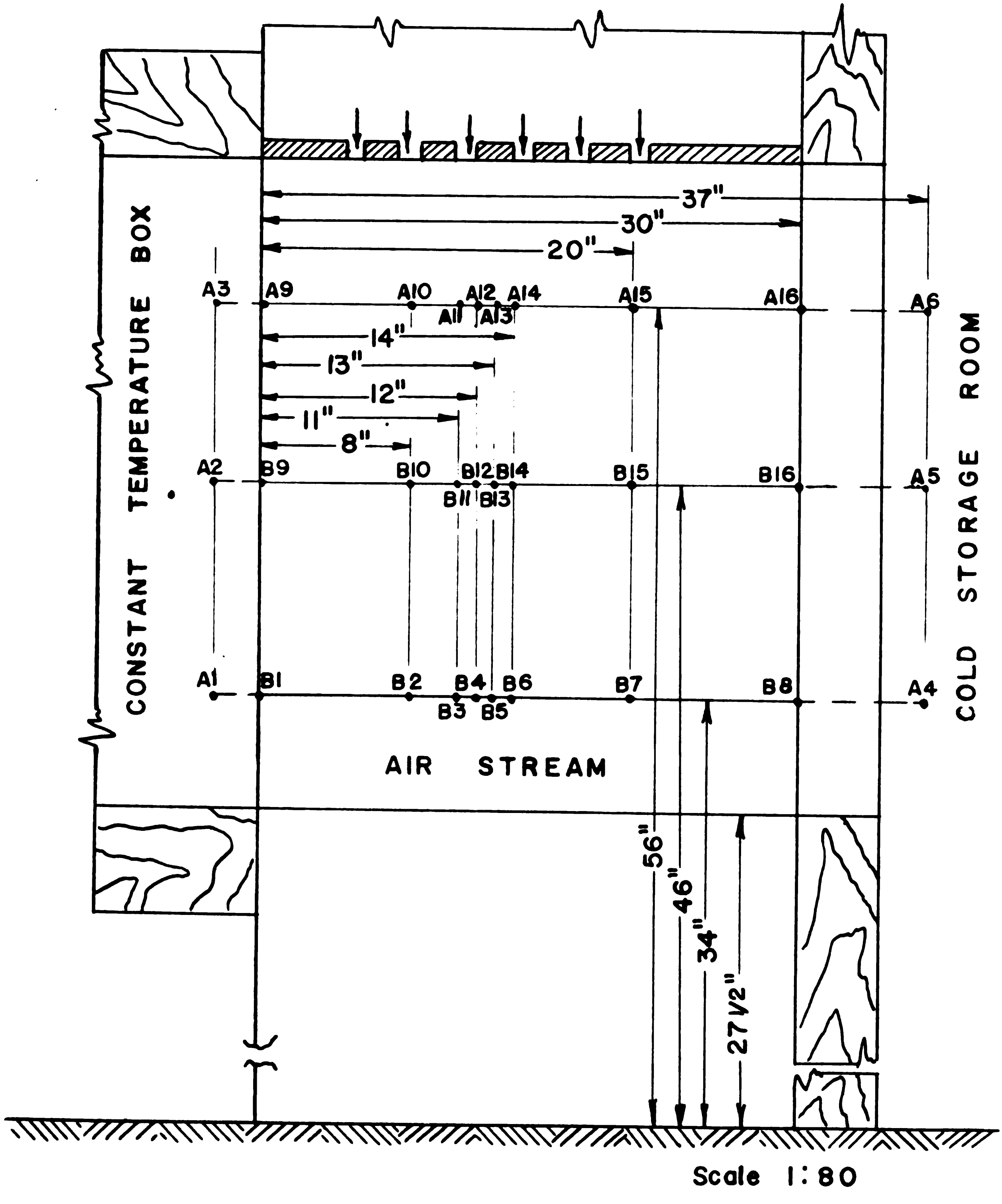

FIGURE 2. ARRANGEMENT OF THERMOCOUPLES

PROCEDURE

A series of tests on air velocity were first run to obtain air velocity patterns for three depths (variable horizontal cross-section) of the curtain. The velocity at each point was determined by means of a Pitot tube and inclined manometer. Measurements were taken at the middle plans of the curtain, every two inches in height and every inch in depth. The values are an average with respect to time, and particular care was taken to obtain reproducible values. The accuracy of the readings was usually $\pm$ 0.001 in water column.

The set-up for velocity measurement is shown in Figure 3.

At the same time a closed door test (Test 2) was run to determine the heat losses from the constant temperature box and the operation time of the refrigeration unit of the cold storage room. The constant temperature box was fastened to the door of the cold storage room (3/4 in. plywood, 4 in. redwood fiber insulation, 1/2 in. plywood). The energy used by the box to maintain the constant temperature of 86 $\pm$ 1^{o}F. the operation time of the compressor of the refrigeration unit and the temperatures inside and outside the chambers were recorded daily with seven replications.

Later an opening of 7.75 sq. ft. (32.5 in. wide; 34.3 in.

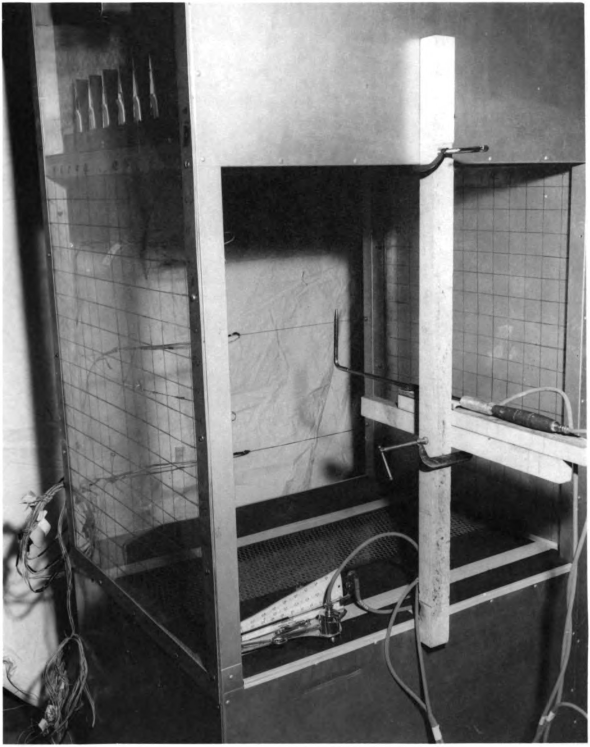

Figure 3. Set-up for velocity measurements.

high) was cut in the door and the air curtain model was fastened to it. The constant temperature box was fastened to the opposite side of the curtain. The thermocouples were placed in the curtain and connected to the recording potentiometer.

Three of six outlet nozzles were blocked and the inlet chamber grille was covered except for an opening 10 in. wide, below the three unblocked nozzles. The curtain was then run six times, tests 3, 5, 6, 7, 8 and 9. The velocity of the fans was changed after each test by means of the variable autotransformer (Variac). The voltage on the Variac outlets, temperatures in the cold storage room, in the constant temperature box, in the air curtain and outside, the operation time of the compressor and energy used by the constant temperature box, were recorded daily, with five replications for each test.

After the series of tests with three nozzles were completed, the air velocity distribution was determined again for the various voltages. Whereas previous air velocity patterns were determined with the same temperature on both sides of the curtain, the air velocity pattern was determined with the curtain installed on the cooler and with the temperature difference maintained. It was necessary to enter the cooler to make the Pitot tube measurements in the air stream. Measurements of the air velocity were taken every inch in width and on three levels of height at the same levels at which the temperature measurements were made, i. e., 56 in., 46 in., and 35 in. from the floor.

Later the covers on three nozzles and on the grille were removed and the series of tests with six nozzles were conducted. The same data as for previous tests were recorded daily with five replications for each test (Tests 10, 11, 12 and 13).

After these tests the air velocities were determined in identical manner as for three nozzles. Again special care was taken to assure that the manometer readings were reproducible.

RESULTS AND DISCUSSION

Air Velocity Distribution

The velocity at various points was calculated from the data obtained by the measurements with the Pitot tube. The calculations were based on the formula

$$V_1 = C\sqrt{2d_m\ v\ g\ h}$$

where V_1 - velocity of flowing gas, fps.

C - velocity coefficient of the Pitot tube. In this case the Pitot tube had a long impact-opening extension and a value 1.0 can be used for C (Eckman, 1950).

d_m - density of manometer fluid, 62.3 lb. per cu. ft. for water.

v - specific volume of flowing gas, 13.5 cu. ft. per lb. for air at 70°F and 50% relative humidity.

g - accelaration of free fall, 32.2 ft. per sec. per sec.

h - manometer differential, ft.

With appropriate values the formula simplifies to

$$V = 4005\sqrt{h} \qquad (1)$$

where V - velocity of flowing gas, fpm.

h - manometer differential, inches of water column.

The air velocity measurements were performed at the middle plane of the curtain. It was observed that air velocity was higher closer to the air return duct and somewhat lower further to the front (Figure 3). It was then assumed that the middle plane would yield an air velocity which would be average of the whole width of the curtain.

The air velocity pattern was determined for two different set-ups. One series of tests was first conducted to obtain the air velocity distribution as occurs in a curtain operating simply in a room with no pressure difference on both sides of the air stream. The results thus obtained were plotted in Figures A-1, A-2 and A-3. These figures are in the form of charts with constant velocity lines.

By studying these figures some general facts can be observed:

1. As the air stream progresses away from the outlet it becomes wider, and the profile of velocity distribution becomes flatter. This is due to the increase in mass of air flowing in the jet and decrease in the maximum velocity, according to the momentum theory: $MV = \text{constant}$.
2. Cross section of the velocity map yields a velocity distribution somewhat similar to a parabola with the highest velocity at the center of the air stream.

3. The air flows in a way similar to a stream of air in a slightly diverging duct.
4. The nozzles form a kind of obstacle in the air stream. Negative pressure zones, due to eddies, can be noticed near the outlets. At lower levels the flow is nearly parabolic again, even at multi-nozzle stream.
5. At about half way from nozzles to the grille the air stream develops a bend. The stream then reforms in its original path.
6. A smoke gun was used for some observations and an air spill was detected in a zone high about six inches from the grille. Above this zone, and mainly near the nozzles, air was entrained into the stream from the outside.

The above facts suggest several conclusions concerning the air flow pattern:

1. The air flow is approximately parabolic by nature and does not consist of several independent streams flowing from several separated nozzles operating in parallel.
2. The fans at the air outlet chamber accelerating the air downwards are more effective than the two fans in the inlet chamber.
3. The Reynolds number of the flow does not change considerably from the nozzles to the grille. The

decrease in air velocity is compensated by the increase in hydraulic diameter.

Another series of tests (Tests 3 and 5 through 13) was conducted with the air curtain operating between the constant temperature box and the cold storage room. A temperature gradient, and probably a pressure gradient also, existed across the air stream.

Manometer readings were taken and velocities calculated for air streams out of three nozzles and out of six nozzles, each at various fan speeds and each at three heights from the floor.

The depth of the air stream at the various levels was taken as the distance between the two points on both sides of the air stream where the manometer readings were approximately zero.

The results are presented in Figures A-4, A-5 and A-6. An average air velocity and depth of curtain for each level was then calculated. These were averaged to get an average air velocity and average depth of curtain for each test.

From this average a Reynolds number was calculated for each test. The formula used was

$$Re = \frac{V\ D_H}{u} \qquad (2)$$

where Re - Reynolds number, dimensionless.

V - average air velocity for the test, fpm.

D_H - hydraulic diameter, ft.

$$D_H = 4 \frac{A}{P} = \frac{2 \text{ x depth of curtain x width}}{(\text{depth} + \text{width})}$$

u - kinematic viscosity sq. ft. per min.

A value for air at 70°F of 9.9×10^{-3} sq. ft. per min. was used.

These numbers are summarized in Table 1 and in Figure A-7.

The maximum percentage error for the hydraulic diameter was estimated to be about 10 percent, that of the velocity about 2.5 percent and of Reynolds number about 13 percent. A sample calculation is presented in the Appendix.

Temperature Distribution

The temperature in the constant temperature box, in the air stream and in the cold storage room, were recorded every hour by means of the thermocouples and a recording potentiometer.

Ten hourly series of recordings were chosen, in the day that had the closest U-value to that test average. The values for the various thermocouples were averaged from the ten readings and the temperature distribution determined for that test.

The temperature distributions thus obtained are presented in Figures A-8 through A-12.

It is worthwhile to mention that the temperature in the constant temperature box was kept fairly constant but the temperature in the cold storage room varied from test to test and was dependent on the U-value of the air curtain. The reason was that the refrigeration unit, though operating continuously throughout all the tests, did not succeed in pulling down the load and in maintaining constant temperature in the cold storage room.

Over-all Heat Transfer Coefficient (U)

In steady-state one dimensional heat conduction the formula used is

$$q = U\,A\,\Delta t \qquad (3)$$

or

$$q = U\,A\,(t_h - t_c) \qquad (4)$$

where
- q - conducted heat quantity. Btu per hr.
- A - cross sectional area (height x width of opening), sq. ft.
- t_h - temperature at the warm side, $^{\circ}F$.
- t_c - temperature of the cold side, $^{\circ}F$.
- U - over-all heat transfer coefficient, Btu per hr. sq. ft. $^{\circ}F$.

Solving equation (4) for U yields

$$U = \frac{q}{A(t_h - t_c)} \qquad (5)$$

The value for A = 7.75 sq. ft. was obtained by measuring the cross sectional area of the air curtain (width x height).

The value for q was obtained by measuring the energy input in the constant temperature box. The value measured was kilowatt-hr. per hr. This value, converted to Btu per hr. by multiplying by conversion factor of 3413 Btu per kilowatt-hour, was called q'.

From this quantity the heat quantity transferred by radiation (q_r) was subtracted. This quantity was calculated by

$$q_r = FC\,A(T_h^4 - T_c^4) \qquad (6)$$

where F - a modulus which modifies the equation for perfect radiators to account for emissivities and shape factors, assumed to be one, because the chambers were approximately in shape of cave with wood and pitch walls.

C - Stefan-Boltzmann constant 0.1714×10^{-8} Btu per hr. sq. ft. R^4.

T_h, T_c - absolute temperature of warm and cold sides, ^oR.

Another quantity subtracted from q' was the heat quantity lost to the room, which was determined in Test 2 the closed door test. This quantity was assumed to be constant for all tests and a value of $q_1 = 80$ Btu per hr. was used.

The heat quantity transferred through the air curtain by combined mechanism of convection and conduction was denoted by q and is

$$q = q' - q_r - q_1 \qquad (7)$$

substituting the value thus obtained in to equation (5) yields the U-value for that particular day.

The accuracy of U-value was then statistically determined. A sample calculation is presented in the Appendix.

The values obtained for the over-all heat transfer coefficient and the confidence limits are summarized in Table 1. The over-all heat transfer coefficients are plotted on semi-logarithmic paper in Figure 4.

The plot of U-value for three nozzles in Figure 4 shows a trend of the curve to become asymptotic to the value of U=15 Btu per hr. sq. ft. ^{o}F. Bearing in mind the fact that the number of observations in this investigation was not large enough to prove this fact, a general conclusion may be suggested: When Reynolds number reaches a certain value or the air velocity reaches the value of about 650 fpm for this depth, the flow is fully turbulent with fully developed eddy currents flowing throughout the air stream.

This hypothesis is further developed in the following paragraphs.

Equivalent Heat Transfer Coefficient (L)

In steady-state one dimensional heat conduction through a medium the formula used is

$$dq = - kA \frac{dt}{dx} \qquad (8)$$

where k - the unit thermal conductivity, Btu ft. per hr. sq. ft. ^{o}F.

A - area, sq. ft.

$\frac{dt}{dx}$ - temperature gradient, ^{o}F.

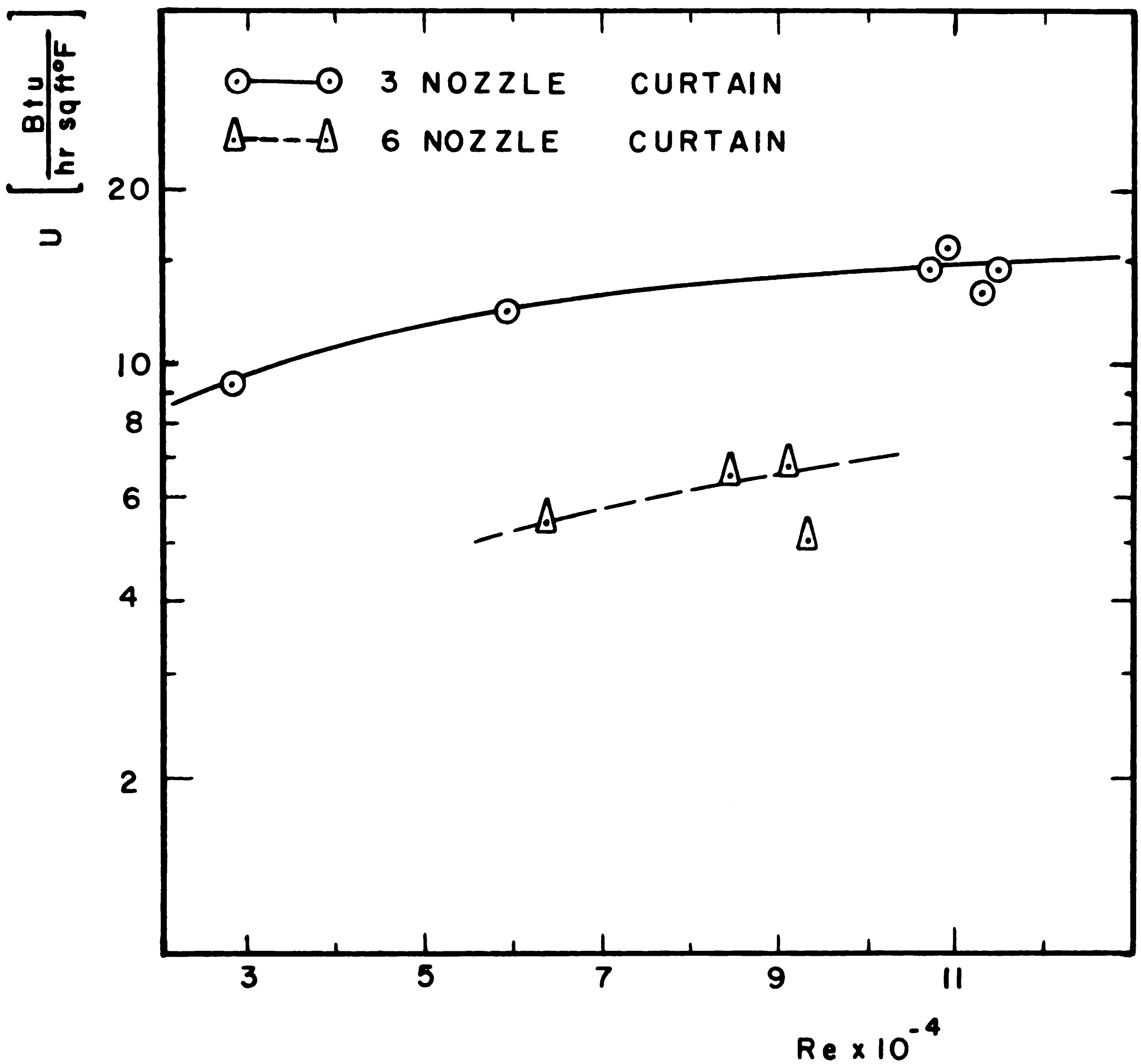

FIGURE 4 SEMI-LOG PLOT OF U- VALUE VERSUS REYNOLDS NUMBER.

The k-value for still air at 70°F is as low as 0.015 Btu per hr. ft. °F, which could not cause the rather high U-values experienced in this investigation. It was assumed therefore, that an equivalent coefficient for heat transfer through flowing air should exist. This coefficient was denoted as L and has the units of Btu ft. per sq. ft. hr. °F.

Equation (8) could then be modified to

$$q = LA\Delta t \qquad (9)$$

or

$$L = \frac{q}{A\Delta t} \qquad (10)$$

The L-value for each day was obtained in a similar manner to the U-value, except for the temperature gradient term. This gradient was established from the temperature distribution in the air stream (Figures A-8 through A-12 in the Appendix) as an average of the three levels. The temperature gradient was obtained for a depth of 6 in. for the three nozzles curtain and 14.4 in. for the six nozzle curtain, allowing some depth for the boundry layers of the air stream.

The appropriate values were then substituted in equation (10) to obtain the L-value for each test.

A sample calculation is presented in the Appendix.

The results are summarized in Table 1.

The method used to obtain the L-value is not precise by any means and the values thus obtained could serve as an approximation only. A more precise analysis has to consider mass transfer, air velocity, rate of turbulence etc. This could have not been done with the limited number of observations in this investigation.

H-Ratio

To generalize the results obtained in this investigation a dimensional analysis was carried out.

The dimensionless number that includes most of the units that characterize the flow is the Reynolds number (Re). The pressure gradient that exists in the air stream was neglected here, though it might be of some importance in practical applications of air curtains.

Another dimensionless number resulting from the analysis was the ratio of the over-all heat transfer coefficient to the equivalent heat transfer coefficient for a certain depth of air stream. This number was denoted as H and is defined as

$$H = \frac{U}{L} D_o \qquad (11)$$

where H - dimension less ratio.

U - over-all heat transfer coefficient, Btu per hr. sq. ft. ^oF.

L - equivalent heat transfer coefficient, Btu per hr. ft. ^oF.

D_o- depth of the air stream when emerging from the nozzles, ft.

The H-number for the various tests was calculated using U-values for each test and an average L-value. The variation of L-value for the various tests was of such magnitude that an average of all the tests (L = 20.9 Btu per hr. ft. $^{\circ}F$) was assumed to be as close as possible to the precise value.

The D-values used were 0.625 ft. for the three nozzle curtain and 1.4 ft. for the six nozzle curtain.

A sample calculation is presented in the Appendix.

The values thus obtained are summarized in Table 1 and plotted on semi-logarithmic paper in Figure 5.

The formula for the over-all heat transfer coefficient, U, is

$$U = \frac{1}{\frac{D}{L} + \frac{1}{h_1} + \frac{1}{h_2}} \qquad (12)$$

where h_1, h_2 = surface unit conductance, Btu per hr. sq. ft. $^{\circ}F$.

assuming $h_1 = h_2 = h$

gives

$$U = \frac{1}{\frac{D}{L} + \frac{2}{h}} \qquad (13)$$

but

$$h = \frac{L}{x} \qquad (14)$$

where x - thickness of boundry layer, ft.

substituting eq. (14) in eq. (13) yields

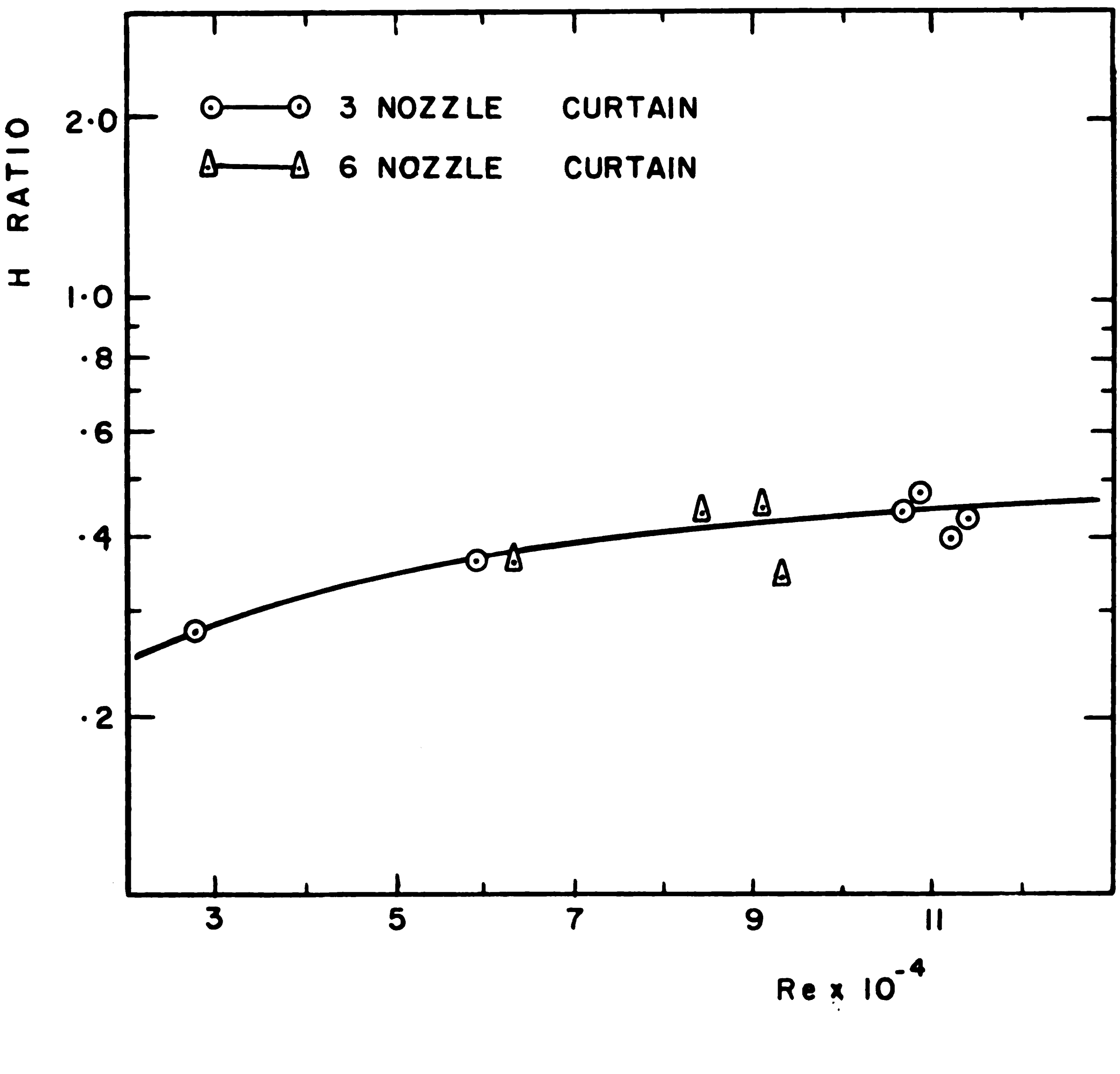

FIGURE 5 SEMI-LOG PLOT OF H-RATIO VERSUS REYNOLD S NUMBER.

$$U = \frac{1}{\frac{D}{L} + \frac{2x}{L}} \qquad (15)$$

substituting equation (15) in equation (11) yields

$$H = \frac{U\ D_o}{L} = \frac{D_o}{(\frac{D}{L} + \frac{2x}{L})L} = \frac{D_o}{D + 2x} \qquad (16)$$

for air streams of high velocity when the nozzles are not very high above the grille, or for large D_o values, it can be assumed that $D = D_o$.

Equation (16) then becomes

$$H = \frac{1}{1 + \frac{2x}{D}} \qquad (17)$$

It is worthwhile to study one extreme case in this equation: When $D = 2x$ equation (17) becomes $H = 0.5$. This merely means that the boundary layers, from two sides of the stream, meet somewhere in the middle of the air stream. The hypothesis that the flow becomes fully mixed, with fully developed eddy currents that run from side to side, at about $Re = 12 \times 10^4$ (for a three nozzles curtain) was previously stated.

The U-values obtained for a Reynolds number of about 11×10^4 was $U = 15$ Btu per hr. sq. ft. °F when $D = 0.625$ ft. and $L = 20.9$ Btu per hr. ft. °F then

$$H = \frac{15 \times 0.625}{20.9} = 0.45$$

which is close to the value obtained using a theoretical approach.

The formula that seems to fit this curve best, for the range of Re = 2.5×10^4 to Re = 11.5×10^4 is

$$H = 0.5\, e^{-1.8/Re \times 10^{-4}}$$

This formula describes a line asymptotic to H = 0.5, as suggested by the theoretical approach. The range of Re $< 2.5 \times 10^4$ was not tested in this investigation and no data are available to verify the suggested formula.

Table 1. Summary of Tests

Test No.	No. of Nozzles	Fan Voltage	Air Stream D-ft.	Air Stream $\bar{V}$, fpm	Air Stream Re x 10^{-4}	U	L	H
9	3	45	0.97	189	2.80	9.2 ± .20	20.8	.275
3	3	60	1.14	355	5.92	12.2 ± .67	17.3	.364
8	3	75	1.31	582	10.68	14.7 ± .48	19.3	.440
6	3	90	1.17	640	10.91	15.7 ± .20	18.7	.469
5	3	115	1.14	673	11.41	14.2 ± .63	18.3	.424
7	3	135	1.20	646	11.23	13.0 ± .24	18.7	.389
12	6	60	1.56	308	6.37	5.4 ± .36	23.2	.360
11	6	90	1.53	411	8.43	6.5 ± .46	29.2	.434
13	6	105	1.81	399	9.12	6.7 ± .57	25.4	.446
10	6	115	1.81	409	9.33	5.0 ± .11	17.8	.334

CONCLUSIONS

1. The profile of the air velocity in the curtain is approximately parabolic and does not consist of several independent streams emerging from separated nozzles operating in parallel.
2. As the air stream progresses away from the nozzles it becomes wider, and the profile of velocity distribution becomes flatter. The Reynolds number of the flowing air remains approximately constant.
3. For the velocities tested the over-all heat transfer coefficient is proportional to air velocity and inversely proportional to the depth of the curtain. The values of the over-all heat transfer coefficient obtained were from 5.0 to 16.0 Btu per hr. sq. ft. ^{o}F for a depth of 0.97 to 1.81 ft. and for an average velocity from 189 to 673 fpm.
4. The ratio of over-all heat transfer coefficient to the equivalent heat transfer coefficient (a value of 21 Btu per hr. ft. ^{o}F was used) for a certain depth of curtain was found to characterize the heat transfer through an air curtain. This ratio was denoted by H.
5. The H ratio is related to Reynolds number of the flow by the formula

$$H = 0.5\ e^{-1.8/Re \times 10^{-4}}$$

RECOMMENDATIONS FOR FUTURE STUDY

1. Determine U-values for higher air velocities and larger Reynolds numbers.
2. Verify the equivalent heat transfer coefficient for air streams at various velocities.
3. Determine the effect of air curtains with air streams at low velocity introduced in the doorway opening from nozzles at the top and at the two sides.
4. Estimate the costs of air curtains with various combinations of depth and air velocity. Plot the expenses and savings in cooling versus various curtains and find the break-even point.

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APPENDIX

SAMPLE CALCULATIONS

A sample calculation is given for Test 7.

1. U-value

Energy consumed by heating element in constant temperature box — 19.1 kilowatt-hr. per 25.02 hr. or $q' = .763$ kilowatt-hr. per hr. = 2600 Btu per hr.

The temperatures in cold storage room, and in constant temperature box, as averaged from 25 recordings

$$t_h = 87.4^{\circ}F; \quad t_c = 64.3^{\circ}F; \quad \Delta t = 23.1^{\circ}F$$

Heat transferred by radiation

$$q_r = F \times C \times A \ (T_h^4 - T_c^4) =$$

$$= 1 \times 0.1714 \times 10^{-8} \times 7.75 \ (547^4 - 524^4) =$$

$$= 200 \text{ Btu per hr.}$$

Heat losses to room, $q_1 = 80$ Btu per hr.

Heat transferred through curtain by conduction and convection -

$$q = q' - q_r - q_1 = 2600 - 200 - 80 =$$

$$= 2320 \text{ Btu per hr.}$$

The over-all heat transfer coefficient for that day, using equation (5):

$$U = \frac{q}{A\Delta t} = \frac{2320}{7.75 \times 23.1} = 13.0 \text{ Btu per hr. sq. ft. } ^{\circ}F.$$

2. L-value

The temperature gradient

$$\frac{\Delta T}{\Delta x} = \frac{7.1\ ^{\circ}F}{6 \text{ in.}} = 14.2\ ^{\circ}F \text{ per ft.}$$

q = 2320 Btu per hr. A = 7.75 sq. ft.

substituting in equation (10)

$$L = \frac{q}{A\,\Delta T} = \frac{2320}{7.75 \times 14.2} = 21.1 \text{ Btu per hr. ft. } ^{\circ}F.$$

3. H-ratio

Using the average of U-value for Test 7 and the average L-value for all tests.

U = 13.0 Btu per hr. sq. ft. $^{\circ}$F.

L = 20.9 Btu per hr. ft. $^{\circ}$F.

D_o = 0.625 ft.

substituting in equation (11)

$$H = \frac{U}{L} D_o = \frac{13.0 \times 0.625}{20.9} = 0.389.$$

ACCURACY OF RESULTS

Accuracy of the results can be estimated after the precision of measurements is estimated and the maximum error calculated.

1. Velocity

$$V = 4005\sqrt{h}$$

$$|\Delta V| = \frac{4005}{2}\frac{|\Delta h|}{\sqrt{h}}$$

assuming $\Delta h = \pm 0.001$ in. for $h = 0.020$ in.

$$V = 4005\sqrt{h} = 567 \text{ fpm.}$$

$$|\Delta V| = \frac{4005}{2} \times \frac{0.001}{\sqrt{0.020}} = 14.2 \text{ fpm.}$$

maximum percentage error

$$\frac{|\Delta V|}{V} \times 100 = \frac{14.2}{567} \times 100 = 2.5\%$$

2. Hydraulic Diameter

$$D_h = 4\,\frac{A}{P}$$

where A - cross section area of air stream = 3 ft. x depth of curtain.

P - wetted perimeter of air stream = 2(3 ft. + depth of curtain).

D - depth of curtain = 1 ft. $|\Delta D| = 1$ in.

$A = 3.0$ sq. ft; $\quad |\Delta A| = 3 \times D = \frac{3 \times 1}{12} = 1/4$ sq. ft.

$P = 8.0$ ft; $\quad |\Delta P| = 2 \times D = \frac{2 \times 1}{12} = 1/6$ ft.

$$D_h = 4\frac{A}{P} = \frac{4 \times 3}{8} = 1.5 \text{ ft}; \quad |\Delta D_h| = \frac{4}{P}|\Delta A| + \frac{4A}{P^2}|\Delta P| =$$

$$= \frac{4}{8} \times \frac{1}{4} + \frac{4 \times 3}{64} \times \frac{1}{6} =$$

$$= 0.156 \text{ ft.}$$

$$\frac{|\Delta D_h|}{D_h} \times 100 = \frac{0.156}{1.5} \times 100 = \underline{10.4\%}$$

3. Reynolds Number

$$Re = \frac{VD_h}{u}$$

$u = 9.9 \times 10^{-3}$ sq. ft. per min. $\quad |\Delta u| = .15 \times 10^{-3}$ sq. ft. per min.

$V = 567$ fpm. $\quad |\Delta V| = 14.2$ fpm.

$D_h = 1.5$ ft. $\quad |\Delta D_h| = 0.156$ ft.

$$Re = \frac{VD_h}{u} = \frac{567 \times 1.5}{9.9 \times 10^{-3}} = 8.6 \times 10^4$$

$$Re = \frac{D_h}{u}|\Delta V| + \frac{V}{u}|\Delta D_h| + \frac{VD_h}{u^2}|\Delta u| = \frac{1.5}{9.9 \times 10^{-3}} \times 14.2 +$$

$$+ \frac{576}{9.9 \times 10^{-3}} \times 0.156 + \frac{567 \times 1.5}{98 \times 10^{-6}} \times 0.15 \times 10^{-3} =$$

$$= 2.1 \times 10^3 + 8.9 \times 10^3 + 1.3 \times 10^3 = 1.13 \times 10^4$$

$$\frac{|\Delta Re|}{Re} \times 100 = \frac{1.13 \times 10^4}{8.6 \times 10^4} = \underline{13.2\%}$$

4. Over-all Heat Transfer Coefficient (U)

The precision for the U-value was determined statistically. A calculation example is given in Test 7.

The figures are presented in the following table.

	$\underline{U}$	$\underline{(U_1 - U)^2}$
	13.1	.0064
	13.3	.0785
	13.0	.0004
	12.8	.0485
	12.9	.0145
Total	65.1	.1483

$$\bar{U} = \frac{65.1}{5} = 13.02$$

$$s^2 = \frac{\text{Sum of squares}}{\text{degrees of freedom}} = \frac{.1483}{4} = .037.$$

standard deviation $s = .192$

Taking level of significance $a = .05$ the Student's

$t = 2.776$

The interval estimation of the results

$$U = \bar{U} \pm \frac{s}{\sqrt{n}} = 13.02 \pm \frac{1.92}{\sqrt{5}} \times 2.776 =$$

$$= \underline{13.02 \pm .24.}\ \text{Btu per hr. sq. ft. } {}^{\circ}\text{F.}$$

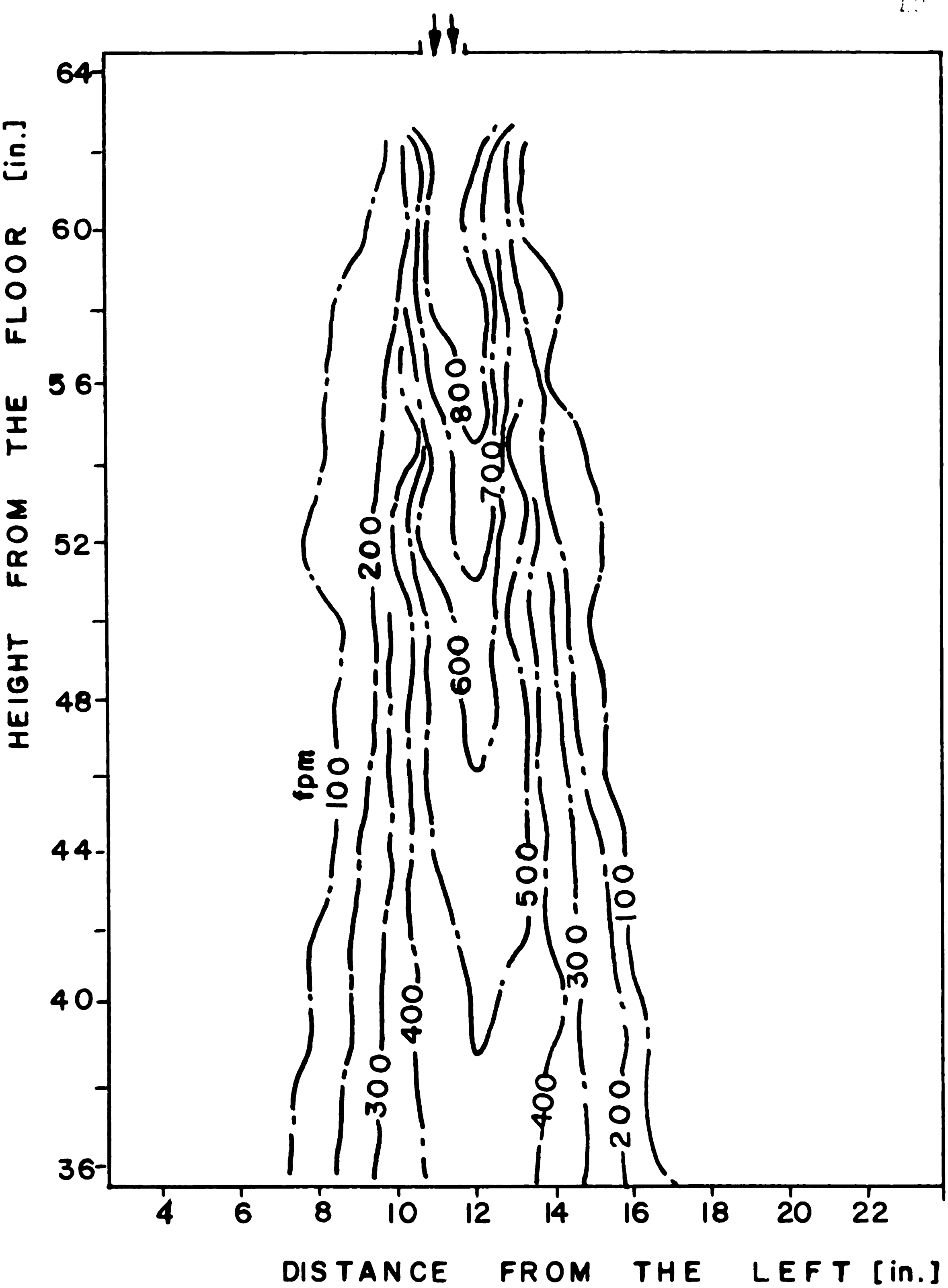

FIGURE A.1 AIR VELOCITY DISTRIBUTION

1 NOZZLE , 45 VOLTS

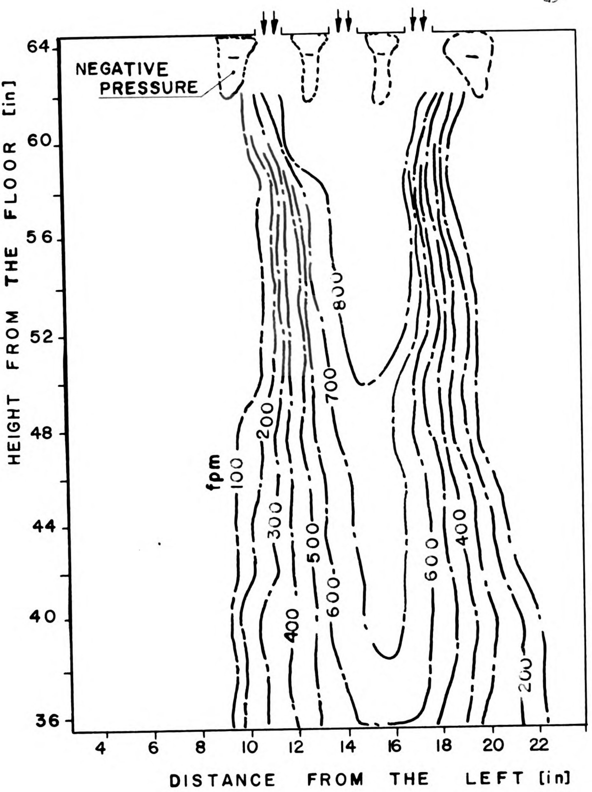

FIGURE A.2 AIR VELOCITY DISTRIBUTION

3 NOZZLES , 60 VOLTS

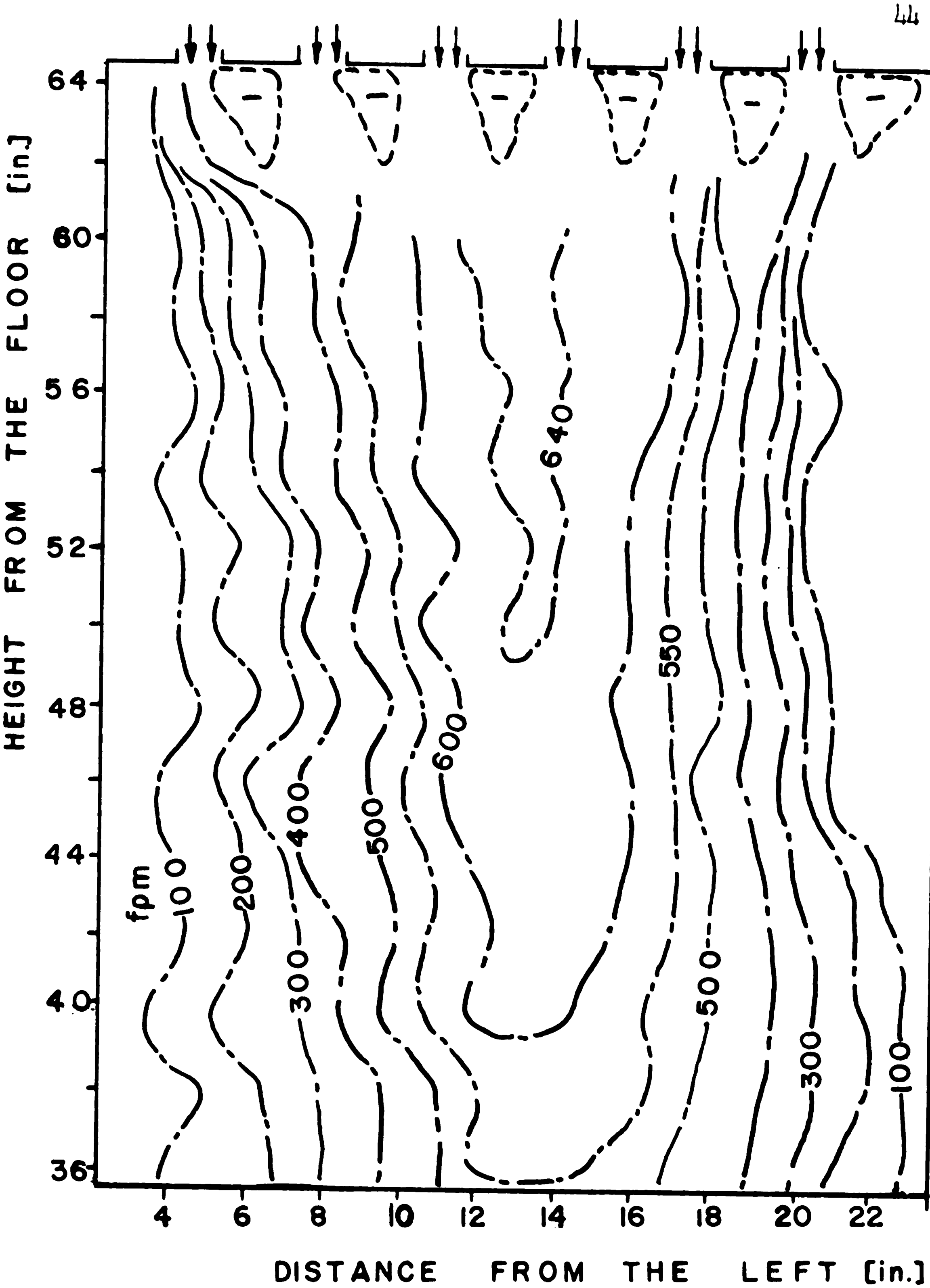

FIGURE A.3 AIR VELOCITY DISTRIBUTION
6 NOZZLES , 115 VOLTS

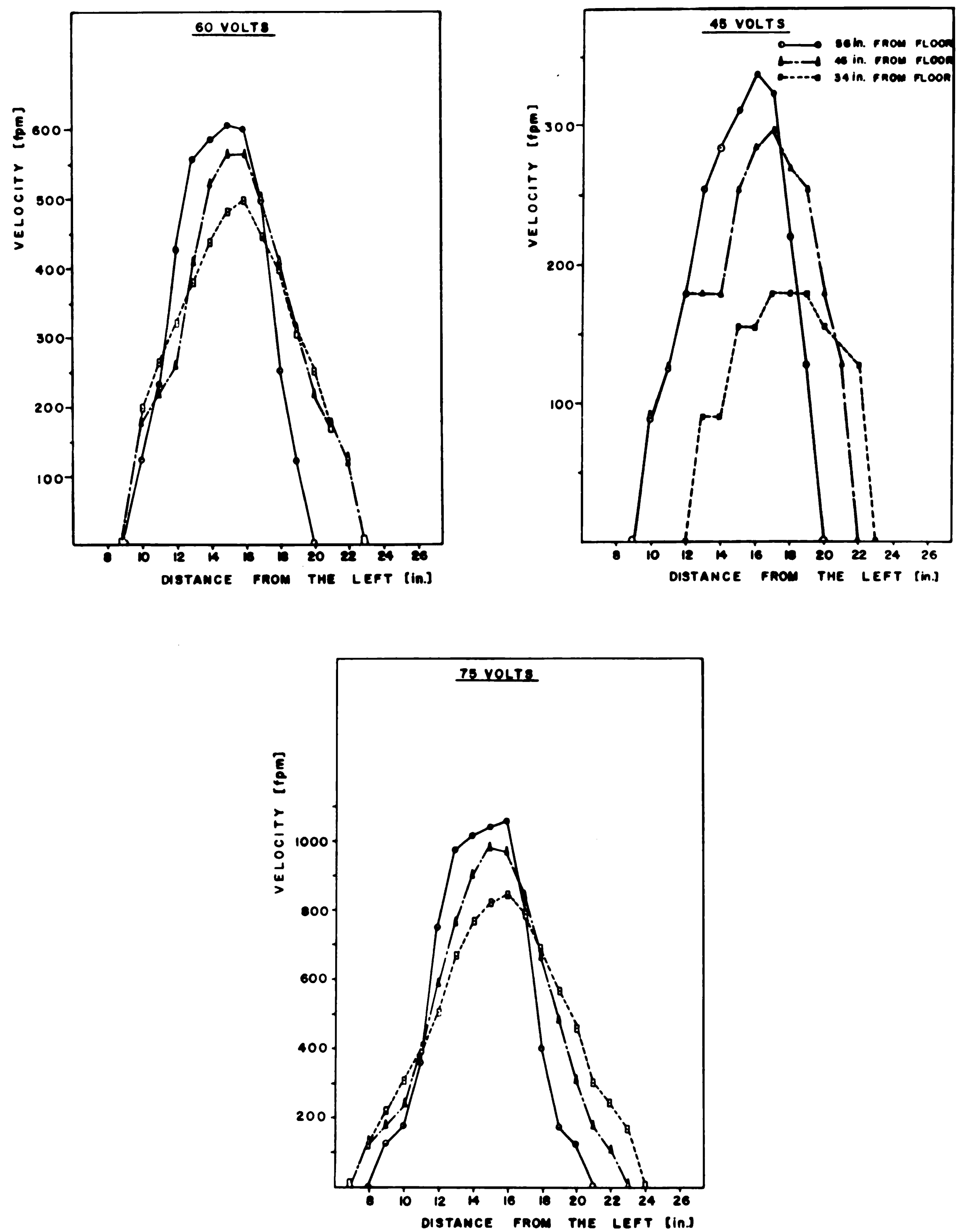

FIGURE A.4 AIR VELOCITY DISTRIBUTION
THREE NOZZLE CURTAIN OPERATING
BETWEEN CHAMBERS.

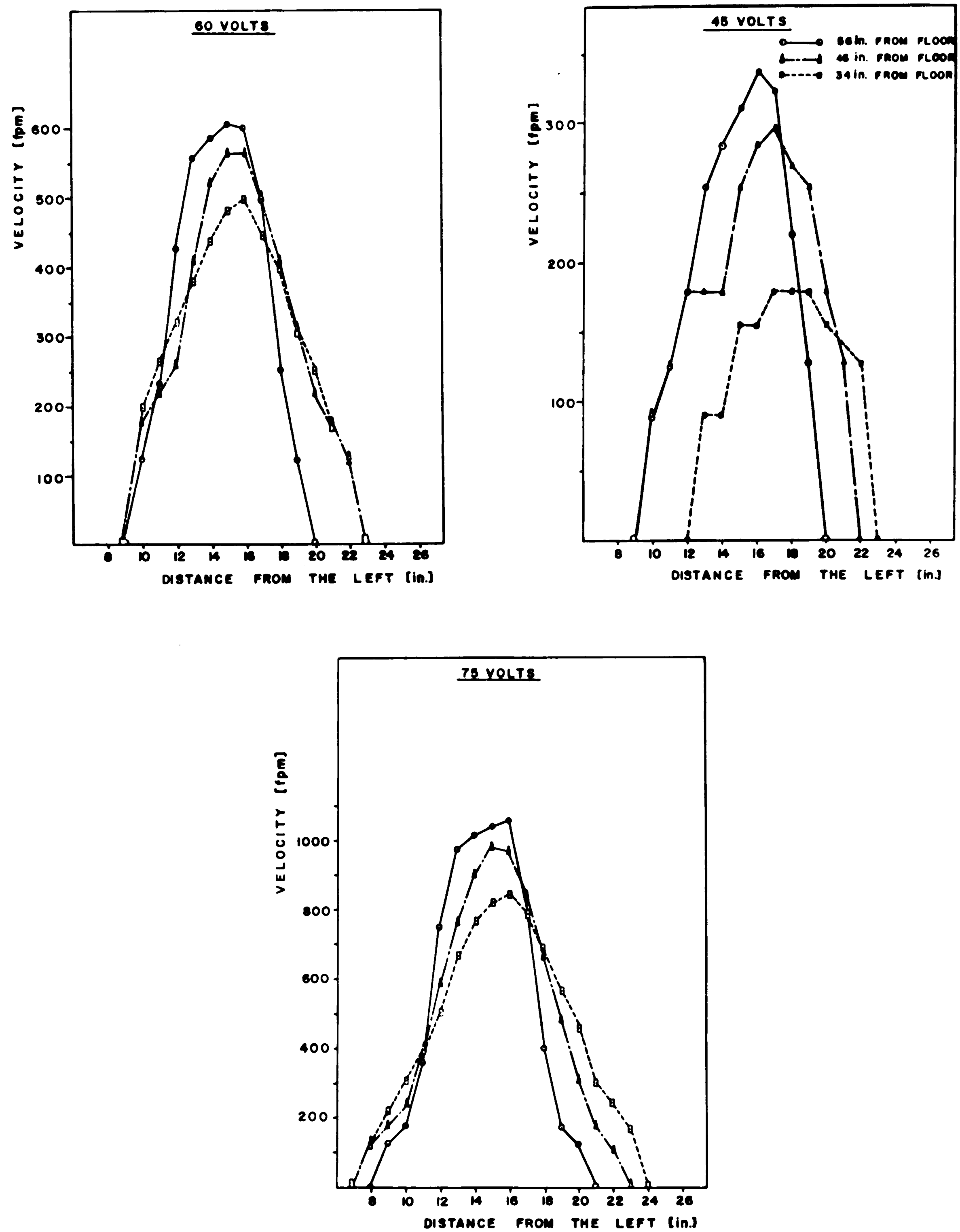

FIGURE A.4 AIR VELOCITY DISTRIBUTION
THREE NOZZLE CURTAIN OPERATING
BETWEEN CHAMBERS.

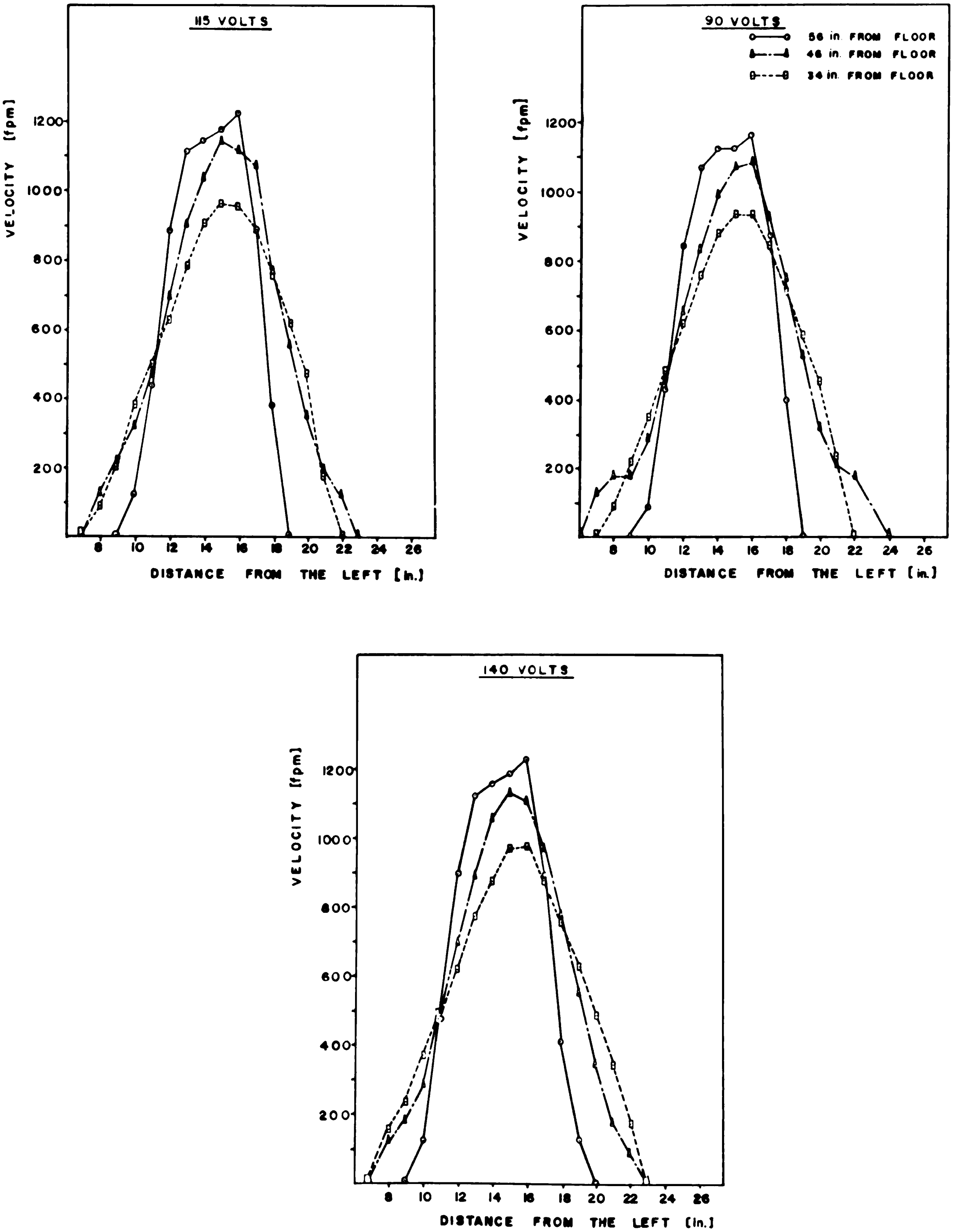

FIGURE A.5 AIR VELOCITY DISTRIBUTION

THREE NOZZLE CURTAIN OPERATING BETWEEN CHAMBERS.

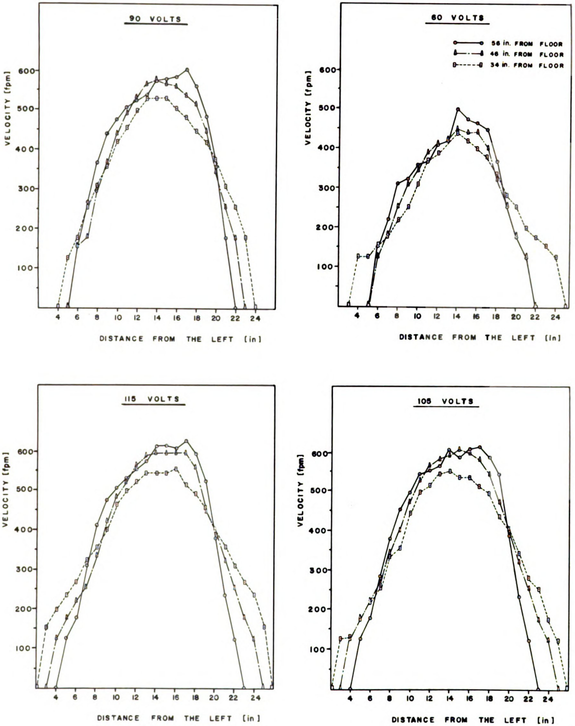

FIGURE A.6 AIR VELOCITY DISTRIBUTION
SIX NOZZLE CURTAIN OPERATING
BETWEEN CHAMBERS.

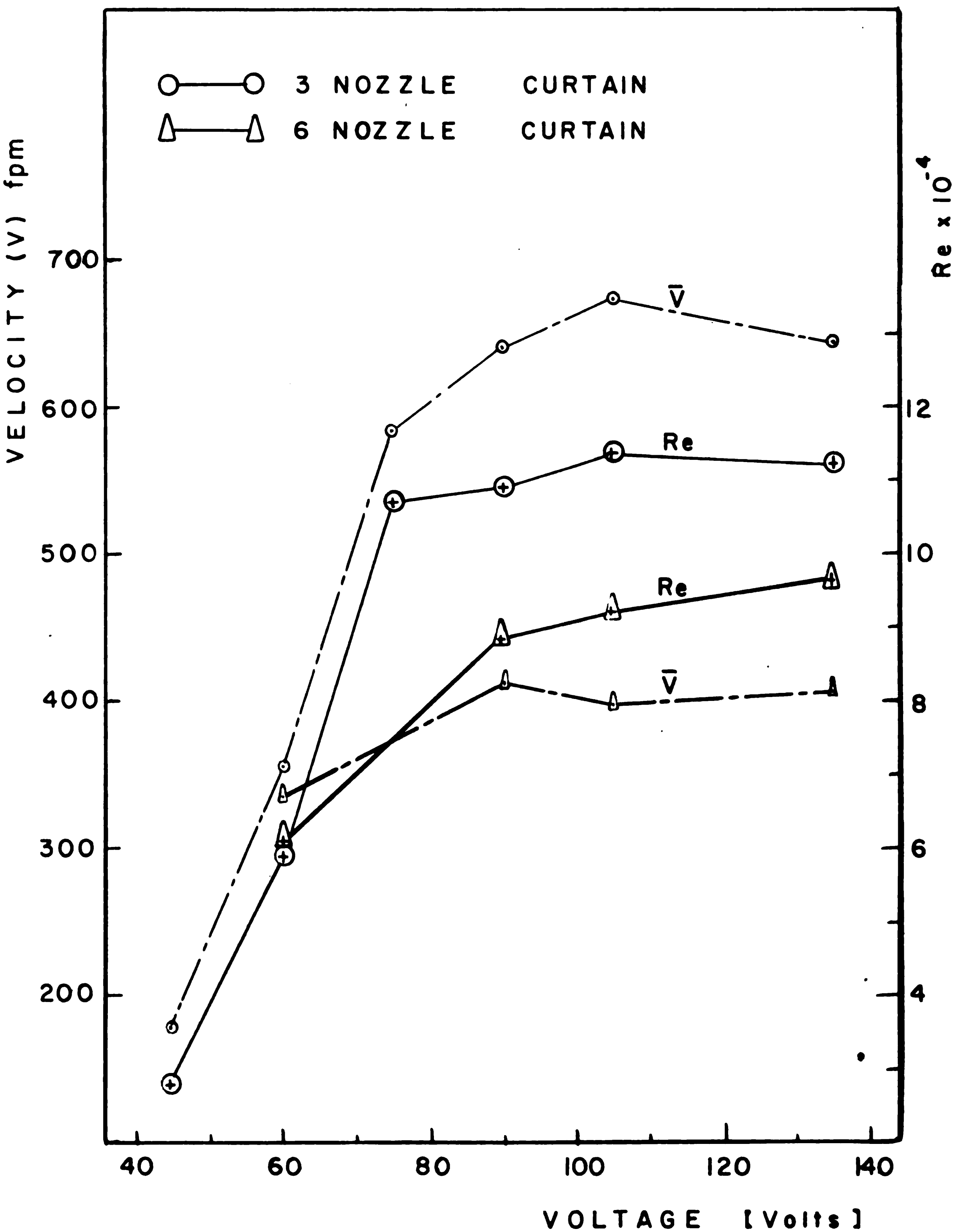

FIGURE A.7 AVERAGE AIR VELOCITY AND REYNOLDS NUMBER VERSUS THE VOLTAGE.

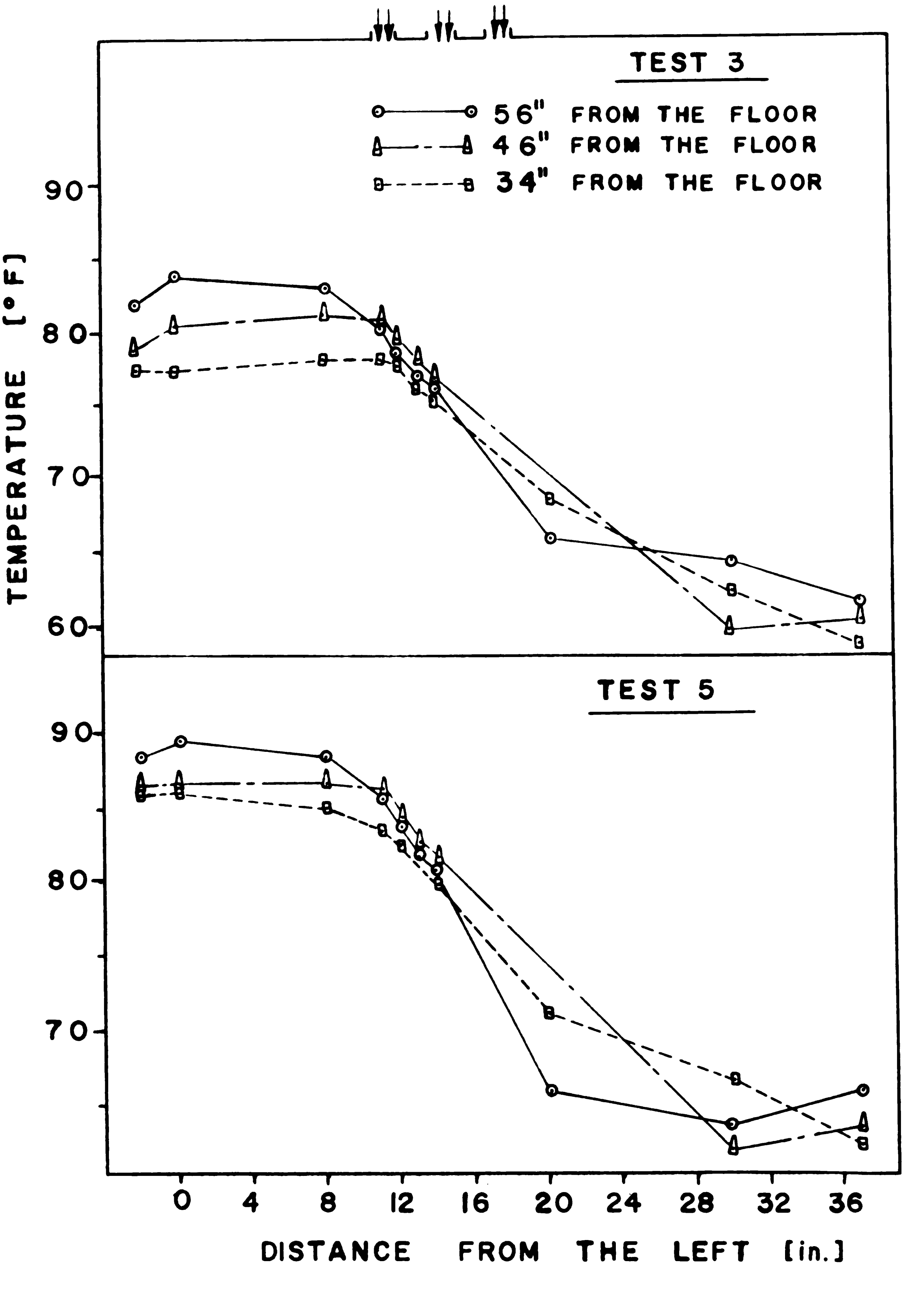

FIGURE A.8 AIR TEMPERATURE DISTRIBUTION

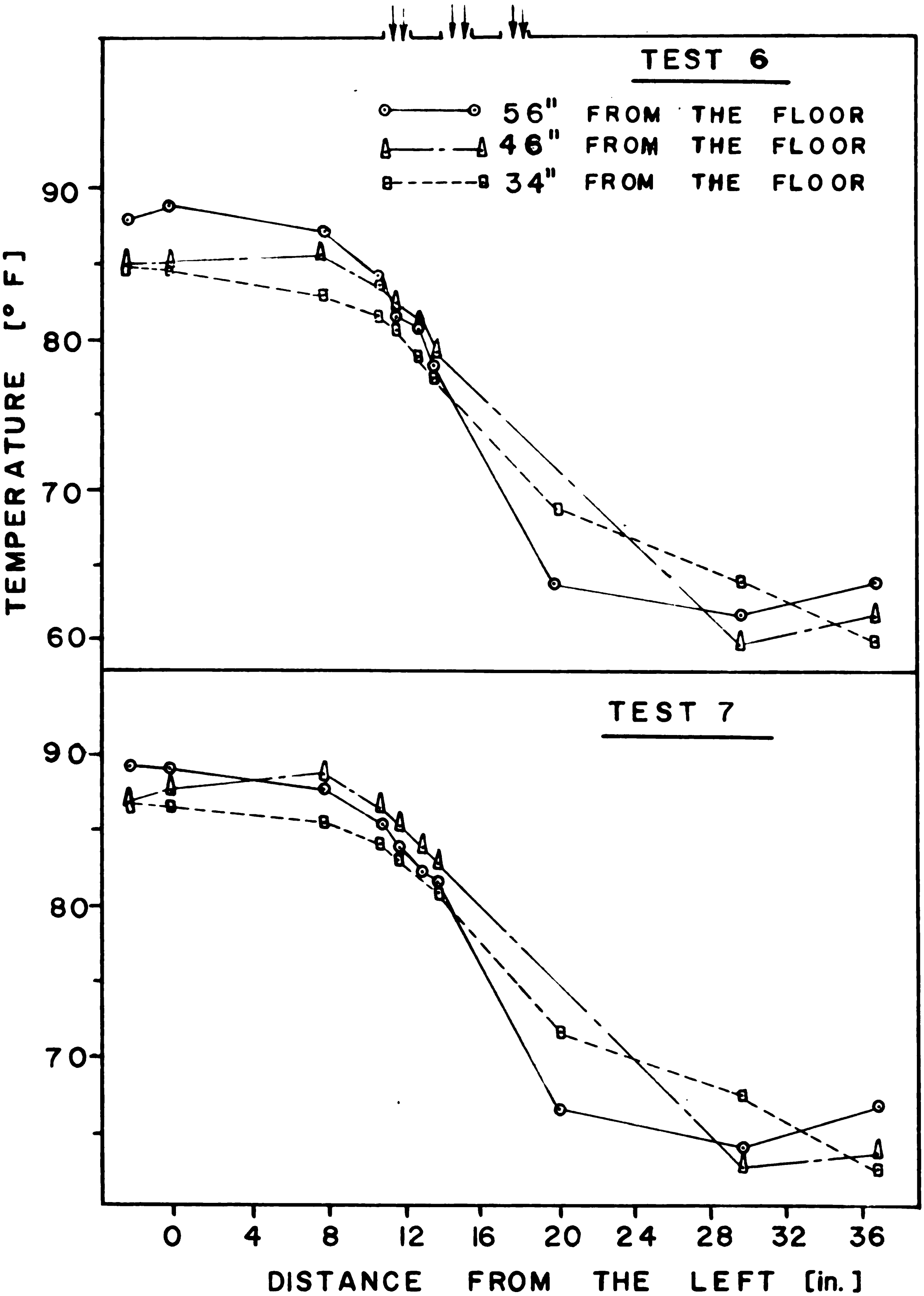

FIGURE A.9 AIR TEMPERATURE DISTRIBUTION

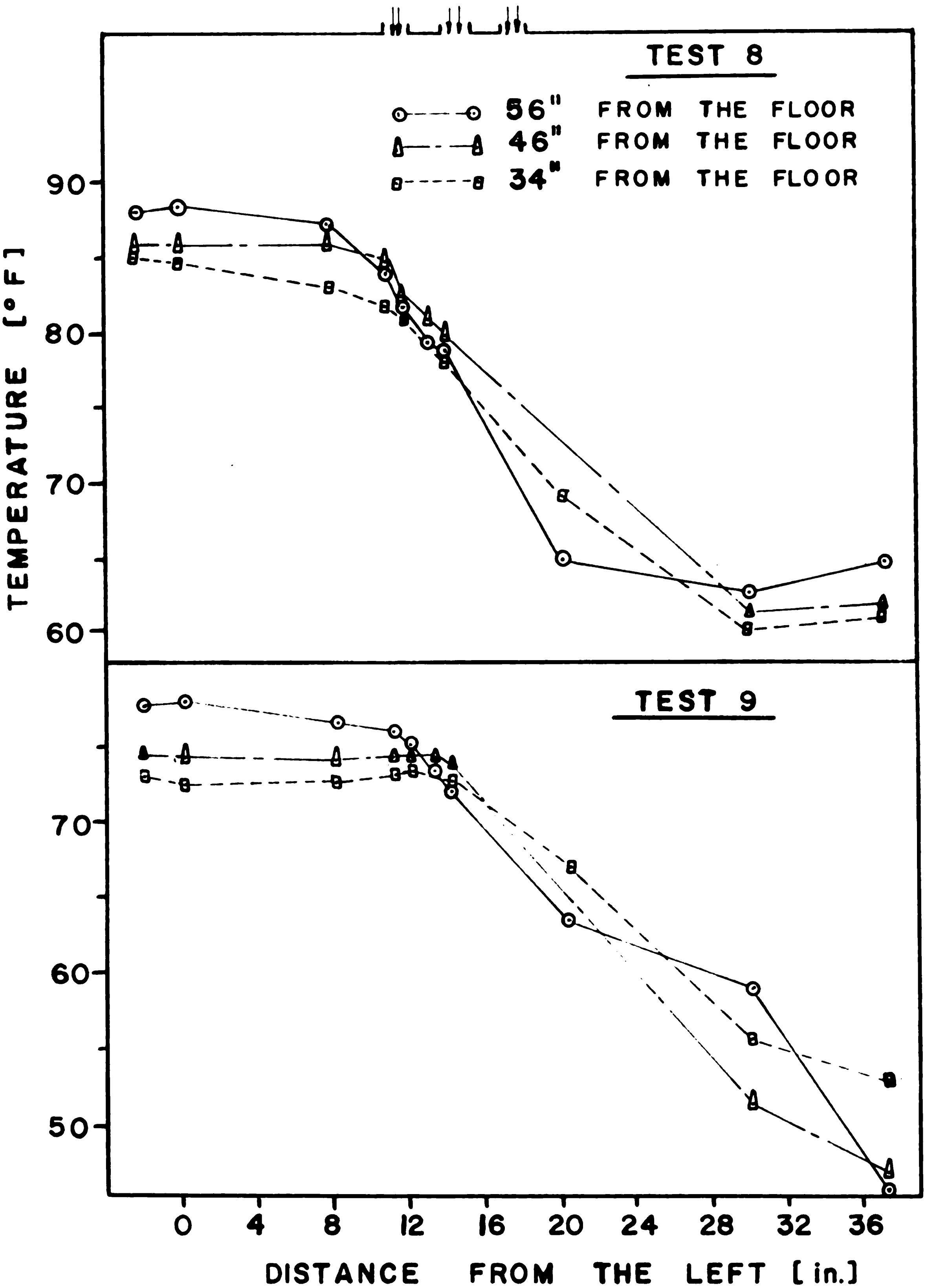

FIGURE A.10 AIR TEMPERATURE DISTRIBUTION

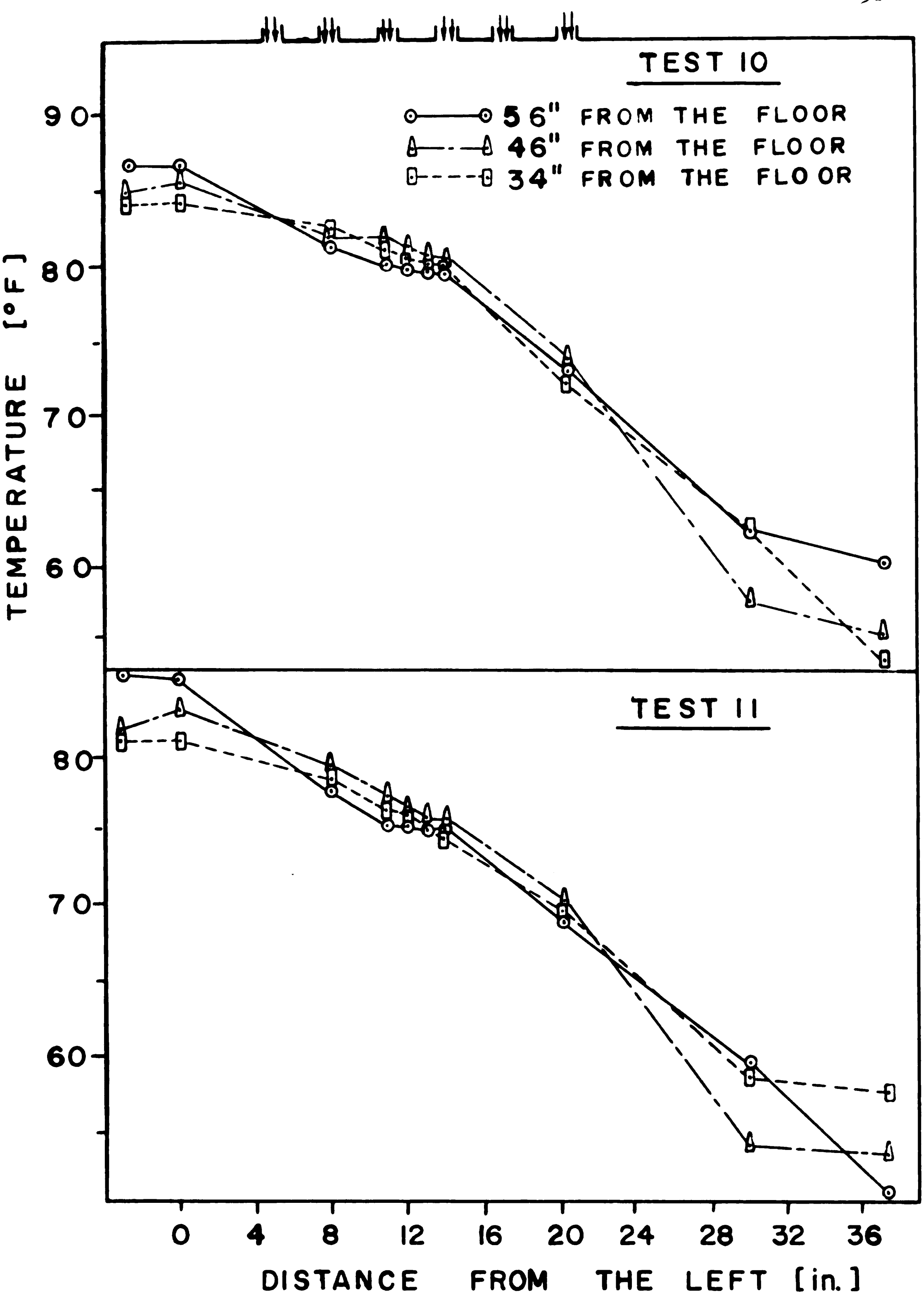

FIGURE A.11 AIR TEMPERATURE DISTRIBUTION

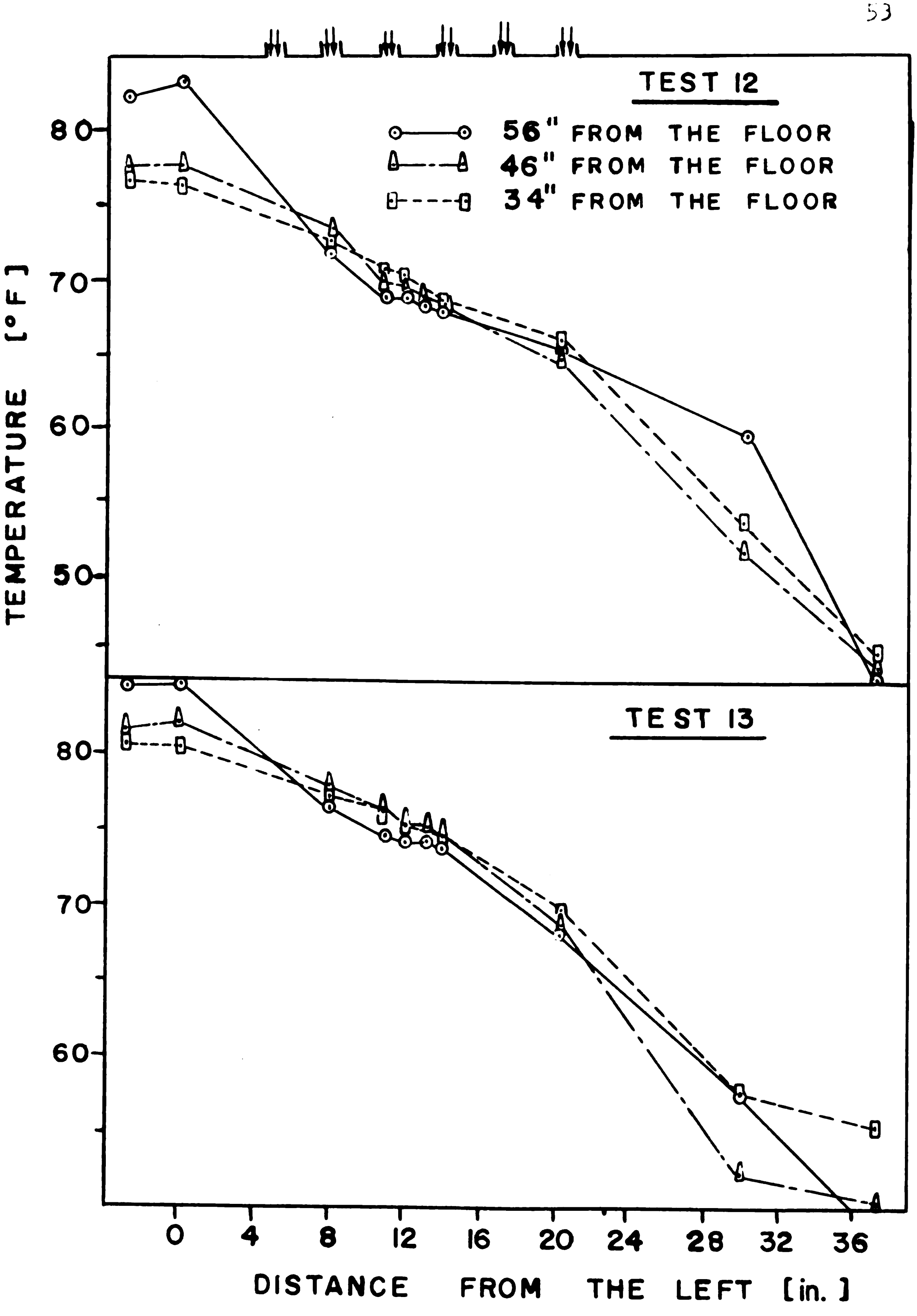

FIGURE A.12 AIR TEMPERATURE DISTRIBUTION