USING AIR CURTAINS TO SIMPLIFY THE THERMAL DEFOLIATION MACHINE

By

GLEN FELIX MOORE

Bachelor of Science

Oklahoma State University

Stillwater, Oklahoma

1968

Submitted to the Faculty of the Graduate College
of the Oklahoma State University
in partial fulfillment of the requirements
for the Degree of
MASTER OF SCIENCE
May, 1969

USING AIR CURTAINS TO SIMPLIFY THE

THERMAL DEFOLIATION MACHINE

Thesis Approved:

Jay G Porterfield
Thesis Adviser

S. T. McCoy

D. D. Durham
Dean of the Graduate College

PREFACE

This work was conducted as part of Regional Research Project 578, "Mechanized Cotton Harvesting in Oklahoma," of the Oklahoma Agricultural Experiment Station. The primary objective of this study was to determine if the air curtain would adequately replace the doors used on previous thermal defoliation machines. The air curtain provides a closed-open door effect, in that, when properly developed, it allows an unobstructed opening, but provides a heat barrier. In operation, the doors used on the thermal defoliation machine were open most of the time, so it was necessary to find an improved heat barrier to use.

At this point, I would like to express my appreciation to all of those aiding me in this project. In particular, my thesis adviser, Professor Jay G. Porterfield, offered invaluable encouragement throughout the project, and Assistant Professor David G. Batchelder provided valuable technical assistance. I would like to thank the staff of both the Agricultural Engineering Research Laboratory at Stillwater and the Oklahoma Cotton Research Station at Chickasha for their help. I am expecially grateful to Jess Hoisington for help in construction of the model, to Galen McLaughlin for help in modifying and testing the prototype, and to Mrs. Audrey Byrd for help with the statistical analysis.

Also, the financial assistance given by the Cities Service Company and the Natural Gas Processors Association is gratefully acknowledged.

Finally, I would like to thank the draftsmen of the Agricultural Engineering Department for help with the drawings and graphs used, and Mrs. Ruth Terry for typing of the thesis.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure | Page

CHAPTER I

INTRODUCTION

Thermal defoliation of cotton has been shown to be economically feasible and practical, but the equipment used in applying the thermal energy needs refinement.

Defoliation is characterized by the shedding of the leaves of the plant. Thermal defoliation of cotton can result from subjecting the plant to a high temperature environment for a period of a few seconds. If the temperature is too high or the exposure time too long, the leaf will desiccate, but not drop from the plant.

A carefully directed, relatively high velocity stream of air, referred to as an air curtain, has received much attention in recent years. An air curtain allows unrestricted movement of objects and also serves as a convenient heat barrier. These two properties make an air curtain ideally suited for use in machines requiring a high temperature environment with unobstructed entrance and exit of objects. The thermal defoliation machine requires a high temperature heating chamber for the plants, but unrestricted movement of the plants into and out of the heating chamber for minimum damage to the plants.

This study involved the development and testing of the air curtain principle in an attempt to adapt it to the thermal defoliation machine. The study was divided into two parts; the first consisting of the model study, and the second was the testing of the prototype machine.

The model was a scaled version of the prototype heating chamber assembly. It was tested in the wind tunnel facilities of the Agricultural Engineering Research Laboratory of Oklahoma State University. Air curtains were used at the front and rear of the model. Compressed air was used to form the air curtains.

On the prototype, a different arrangement was used. A transverse flow fan was used to develop the air curtains. This fan is unique in that it draws air radially from one side of a centrifugal blower wheel and exhausts it radially at the other side so that along the full length of the fan a uniform air flow was achieved. This type of fan gives a continuous, uniform flow of air along the full width of the machine, achieving the air curtain effect. The prototype was tested at the Oklahoma Cotton Research Station near Chickasha, Oklahoma. One group of field tests was performed with plant response, in the form of percent defoliation and percent desiccation, as the test criteria. Another group of tests involved studying the elevated temperature period and maximum temperature for different operating speeds, temperatures, and heights. The final group of tests was evaluation of the air velocity and temperature environment surrounding the machine.

CHAPTER II

OBJECTIVES

Objectives:

1. To determine by model testing in the wind tunnel if the air curtain is applicable to a thermal defoliation machine.
2. To determine important operating parameters for the prototype by wind tunnel evaluation of the model. Parameters studied were: wind velocity, wind direction, air curtain velocity, and air curtain direction.
3. To determine if the air curtain will provide an adequate heat barrier on the prototype.

Hypotheses:

1. The air curtain principle has been understood since the early 1900's, but only in recent years has it been used successfully. Many stationary applications are currently in use. The column of air forming the curtain diffuses rather slowly as it progresses. Also, since the air curtain allows unrestricted movement of objects and presents a partial heat barrier, it was hypothesized that this would be an ideal arrangement for the thermal defoliation machine.
2. Wind velocity was considered important because the impact forces of the wind must be overcome by a resisting force of

the curtain. It was hypothesized that there existed a maximum wind velocity above which the curtain would be ineffective.

A wind aimed directly into the machine was considered to be the most severe condition since the air curtain would be the only barrier. As the machine was turned until the wind was directed at the sides of the machine, it was hypothesized that the air curtain would be decreasingly important to operation.

The air curtain velocity was considered important as with increasing air velocity, a greater volume of air and more energy will be available to direct against the wind.

The air curtain direction was considered important as this determines the component of the air curtain energy which is directed against the wind. Also important is the amount of diffusion which occurs as the air curtain strikes the ground. As the curtain is directed increasingly outward, less and less air from the curtain turns back into the machine.

3. It was hypothesized that the air curtain will present a definite heat barrier. It was further hypothesized that the most significant heat loss to this arrangement would be in the form of conducted heat carried away by wind. If a significant wind barrier could be found to replace the doors previously used, an increase in efficiency and economy could be expected.

CHAPTER III

REVIEW OF LITERATURE

Thermal Energy in Agriculture

Thermal energy is being used more in agriculture each year. Agricultural operations using heat include weed and insect control, crop drying, thermal defoliation, and as a source for other forms of energy.

Grain drying consists primarily of batch drying of grain after it has been harvested, though some attempts have been made to dry grain in the field, with somewhat uncertain results (1, 2).

Weed control consists of using a direct flame on the weed. The usual result is immediate death of the plant. In some crops, where the cultivated plant is too small for flaming, a preplant herbicide, cultivation or both are used prior to the flaming (3, 4, 5, 6, 7).

The use of heat as an insecticide has received attention recently. Cultural practices seem to have a significant effect in this area (8).

Thermal defoliation consists of subjecting the plant to a high temperature environment for a few seconds. Kent (9) reported that approximately 75 percent defoliation and 90 percent kill was accomplished with an exposure time of five seconds and a temperature of 350 degrees Fahrenheit.

For two second exposure and 500 degree temperature, defoliation was 70 percent and kill was 90 percent. Decreasing exposure time or temperature resulted in less defoliation and desiccation. Also, if the exposure time is too long or the temperature too high, the leaf desiccates, but does not shed from the plant. Some boll and lint damage can also occur from excessive temperatures or exposure times.

Research has been performed previously at Oklahoma State University in developing a thermal defoliation machine (1, 9, 10). Work has been carried on elsewhere, also (11). The current machine at Oklahoma State University uses a series of burners mounted in a duct which is located above the heating chamber. The air is blown forward over the burners by the burner fan and turned downward at the front of the machine to enter the heating chamber. It is directed onto the plants and returned to be reheated. Figure 1 shows a schematic representation of this air flow pattern. The heated air is enclosed in an envelope of cooler air in such a manner that the structural parts of the machine remain at a safe temperature. This is a desirable arrangement, with one exception. On the front and rear of the machine are spring operated doors which must open and close each time a plant enters or leaves the machine, disrupting the heated air flow patterns. For this reason, it would be desirable to have the entrance and exit completely open. On the other hand, with a completely open entrance and exit, there is little chance of maintaining the flow arrangement because of wind effects. With the addition of the air curtain, it was believed that a closed-open door effect could be achieved.

The Air Curtain

The first development of the air curtain came around 1904 when Theophileus Van Kemmel applied for a patent to replace a door with a curtain of moving air. There is no indication that any such installation

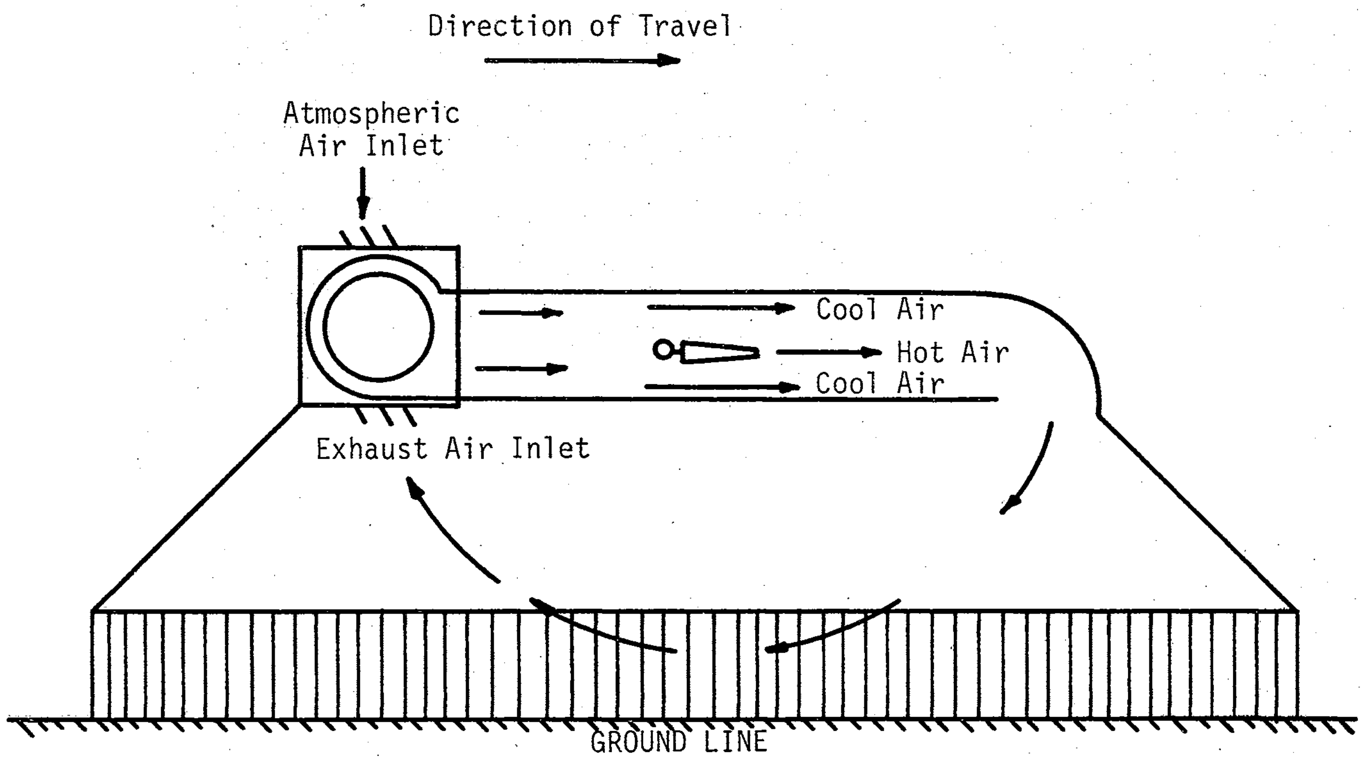

Figure 1. Air Flow Pattern for Heating Chamber

was made. The first recorded installation of the air curtain was in 1916 when an American named Caldwell installed a unit in a building doorway (12). The operation was as follows: Air from the inside was blown downward in front of the door. As it moved downward, it mixed with outside air. At the floor was a grating allowing the air to be returned, with some of the air being returned inside, and the remainder going to the outside. This installation and those following seemed to have limited success. The first truly successful air curtain was installed in the Oscar Weber Department Store in Switzerland in 1952.

An air curtain is a high velocity stream of air which is carefully directed across a doorway or other opening. The air curtain restricts the movement of humidity, dust, insects, and other light material, as well as presenting an insulation barrier to heat. It is not an absolute heat barrier, but is from 75 to 90 percent effective, as compared to a closed door. This seems to defeat the purpose of the door, except in the case where the door would be open a great deal of the time. This occurs with the thermal defoliation machine as it is operating in the field.

There are two basic types of air curtains, the vertical type and the horizontal type. The vertical type is further subdivided into ducted and nonducted return. Air curtains as wide as 87 feet and as high as 18 feet are currently in successful operation (12, 13).

Wind velocity is a major factor to be considered in the design of an air curtain. The angle of discharge and the outlet velocity must be adjusted accordingly. For the curtain to be effective, the outward component of the air curtain must equal the inward component of the wind. Without a screen to break the direct blast of the wind, 15 miles per hour is usually the maximum velocity that a curtain can deflect (13). In some instances an automatic controller has been used to change the direction of the air curtain to suit an approaching wind (14).

The air curtain is of value anywhere heavy traffic is encountered and it is desirable to maintain a temperature difference across an opening (12). This includes almost all public buildings, commercial and industrial application, garages and service stations, and other places requiring frequent opening and closing of large doors.

Developing the Air Curtain

To develop the air curtain, it is necessary to have a uniform air flow for the complete width of the opening which the air curtain is to cover. For permanent installations in buildings, it is possible to use a recirculation system of ducts and fans, continually reusing the same air. On the thermal defoliation unit, a recirculation arrangement is not possible because of space limitations and the machine configuration. Other arrangements were studied.

Diffusion of a jet is a complex phenomena. Figure 2 is a two-dimensional representation of the flow pattern assumed by Albertson, et. al. (15). It seems there are two zones of flow for this diffusion. As the flow first emerges from the boundary opening, a zone of flow establishment is formed. Since the fluid discharged from the boundary is of relatively constant velocity, compared to the surroundings at the exit section, there will be a definite velocity discontinuity. This will be a region of high shearing forces. Consequently, a great deal of turbulence will develop. As movement progresses away from the exit section, the turbulence will progress laterally toward and away from the

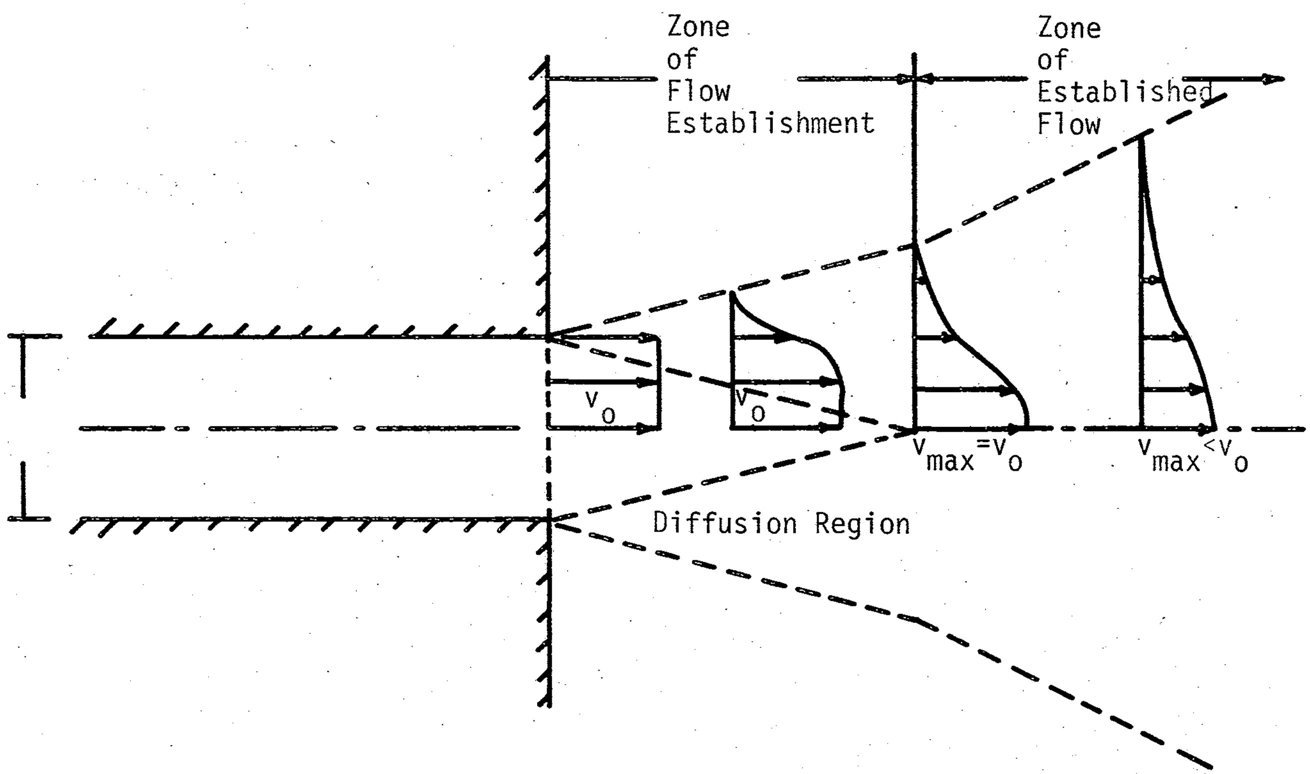

Figure 2. Diffusion Patterns of Jet

center of the flow. This mixing has a two-fold effect: the fluid within the turbulent part of the jet is decelerated, and the surrounding fluid is drawn into the flow and accelerated. Finally, the turbulent diffusion region reaches a point where all of the flow is composed of the turbulent flow. This is the zone of established flow.

Albertson, et. al. (15) developed a series of equations to describe flow of submerged jets. A primary equation which they derived was:

$$X_o = B_o/C_1 \sqrt{\pi},$$

where X_o is the length of the zone of established flow, B_o is the width of the emerging jet, and C_1 is a numerical constant determined by the exit boundary conditions.

For the zone of established flow, they developed prediction equations for the maximum velocity and the velocity destribution at given distance from the exit.

Using the principles as developed by Albertson, et. al., it is possible to develop a flow arrangement which will approximate the flow of an air curtain by incorporating only the outlet portion of the air curtain. This can be done if the centerline velocity of the curtain is not significantly different than the exit velocity before the ground is reached.

Transverse Flow Fans

The transverse flow fan is a unique, relatively new fan design. Its appearance is similar to a centrifugal fan. When it was introduced to North America only a short time ago, its applications seemed very limited. Also, it had several serious faults, among which were high noise level, instability when operated away from a point of maximum efficiency, and sensitivity to variation in shroud dimensions. Through careful design, the faults have been overcome or reduced significantly.

The transverse flow fan is unique in that it both draws air into the fan and discharges it in a radial direction, while other fans must draw the air into the fan in a direction parallel to the axis of rotation and discharge it in a radial direction.

Several advantages are inherent in the performance of the transverse flow fan. Slower rotor speeds and higher static pressures can be achieved with a given rotor size. The blades have a self cleaning action because the air flows both directions across the blades. The rotor length is not limited by fan wheel dimensions, that is, the rotor can be any length (16).

As would be indicated by the sensitivity of the configuration to changes in shroud dimensions, the flow is relatively complex. The air is drawn in from one side of the rotor, passed through in a direction transverse to the rotor, then discharged. A vortex is produced which has its center inside the rotor, and which rotates in the same direction as the fan wheel. If the center of the vortex approaches the center of the wheel, a significant decrease in static pressure occurs. This was a serious problem with early designs, but in recent designs, guiding vanes and other changes have stabilized the flow arrangement.

Although several configurations have been developed, there are two basic configurations currently in use. The main difference in the two is the direction of the flow path. In the most popular type, the Datwyler or D-type, there is almost a complete reversal of flow direction. In the Coester or C-type, the flow is in line, or little direction change. The fan wheel size is determined using fan equations similar to fan equations used for more conventional configurations (17, 18).

Heat Transfer Properties of the Air Curtain

Heat transfer properties of an air curtain were investigated by Hetsroni (19). He used two well insulated chambers which were separated by an air curtain. One chamber was heated and the other cooled. By carefully measuring the amount of heat added to one chamber and removed from the other, and knowing the physical properties of the air curtain, he was able to obtain a prediction equation of the following form:

$$Nu/Pr = K(0.3058 - 0.2718a') \, Re \sqrt{H/b_o}.$$

Nu, Pr, and Re are the Nusselt number, Prandtl number, and Reynolds number, respectively. K, a', H, and b_o were numbers determined by his system. K and a' were experimentally determined quantities, H was the air curtain height, and b_o was the half-thickness of the air curtain at the outlet.

Another equation which Hetsroni used was:

$$h = q/HTm.$$

This is a definition of the heat transfer coefficient. H is the height of the air curtain, Tm is the temperature difference from one side of the curtain to the other, q is the amount of heat transferred across the curtain per unit width per unit time, and h was the heat transfer coefficient.

CHAPTER IV

MODEL DESIGN

Preliminary Considerations

In designing the model, only the heat chamber was modeled. This eliminated the need to reproduce the structural part of the prototype on a smaller scale. Also, several simplifying assumptions were made concerning the heating chamber. The internal air flow circulation was considered to be unimportant to the operation of the air curtains. This greatly simplified construction of the model. The effects of the cotton plant were neglected. That is, no attempt was made to simulate vegetation surrounding the model. Further, it was assumed that adequate results could be obtained with the model using much lower temperatures than were present in the prototype. With these considerations in mind, the following design was used.

In an attempt to provide a simple but adequate model, two basic factors were important to the model design. The first is the source of air used to generate the air curtains. Air could be taken from the immediate surroundings of the model using a fan, or an external air source since a compressed air system was already present in the Agricultural Engineering Research Laboratory. Next to be considered was the type of air curtain generating system. Due to the limited supply of air available, preliminary investigation included tests to determine the effectiveness of simulating an air curtain with high velocity air exiting from a series of small holes in a pipe. The final arrangement used was a one-inch outer diameter thin-walled pipe with an effective length of 10 1/2 inches and 45 holes of .0225 inch diameter. Average air velocity versus pressure measurements were made using an Alnor type 3002 velometer. The results of these tests are presented in Chapter V and the original data is given in Appendix A, Tables A-I through A-III.

The heat source consisted of a one kilowatt electrical resistance heater connected to a volt meter and an ammeter to determine the heat input.

Dimensional Analysis

Dimensional analysis principles were used to develop the experimental design. Pertinent quantities are presented in Table I.

The matrix rank of the pertinent quantities of Table I is five. With twelve pertinent quantities and a matrix rank of five, seven Pi terms are necessary to completely describe the system. The arrangement of Pi terms chosen for this study are presented in Table II.

Pi 7 is the dependent pi-term since it involves the heat transfer coefficient, h, which is an indication of the effectiveness of the air curtain.

Pi 1 is the Prandtl number, and is essentially constant for air for both the model and the prototype.

Pi 2 and 3 are Reynolds numbers associated with the air curtain and the model, respectively. The Reynolds number is an index of the ratio of inertial forces to viscous forces.

Pi 4 is a geometric ratio of air curtain thickness to height which will remain essentially constant for both the model and the prototype.

TABLE I

PERTINENT QUANTITIES OF DIMENSIONAL ANALYSIS

NO.	SYMBOL	DESCRIPTION	UNITS	DIMENSIONS
1	C_p	Specific Heat at Constant Pressure	Btu/lbm-°F	$HM^{-1}\theta^{-1}$
2	h	Heat Transfer Coefficient	Btu/ft^2-°F-sec.	$HL^{-2}\theta^{-1}T^{-1}$
3	d	Thickness of Air Curtain	ft.	L
4	V	Air Velocity Relative to Model	ft/sec.	LT^{-1}
5	v	Air Curtain Velocity	ft/sec.	LT^{-1}
6	a_1	Front Air Curtain Angle	Radians	-
7	a_2	Rear Air Curtain Angle	Radians	-
8	H	Height of Air Curtain	ft	L
9	u	Absolute Viscosity of Air	lbf-sec/ft^2	FTL^{-2}
10	υ	Kinematic Viscosity of Air	lbf-sec-ft/lbm	$FTLM^{-1}$
11	k	Thermal Conductivity of Air	Btu/sec-°F-ft	$HT^{-1}\theta^{-1}L^{-1}$
12	N_e	Newton's Second Law Coefficient	lbf-sec^2/ft-lbm	$FT^2L^{-1}M^{-1}$

H - Heat, θ - Temperature, F -Force, M - Mass, L - Length, T - Time

TABLE II

Pi TERMS FOR DIMENSIONAL ANALYSIS

Pi-TERM	DEFINITION	DESCRIPTION
Pi 1	$C_p u/kN_e$	Prandtl Number
Pi 2	dvN_e/υ	Air Curtain Reynolds Number
Pi 3	HVN_e/υ	Model Reynolds Number
Pi 4	d/H	Air Curtain Thickness to Height Ratio
Pi 5	a_1	Front Air Curtain Angle
Pi 6	a_2	Rear Air Curtain Angle
Pi 7	hH/k	Nusselt Number

Pi 5 and Pi 6 are measures of front and rear air curtain angles, respectively. As the angle of each increases, a greater component of the curtain velocities will be directed outward to oppose an oncoming wind. Also, diffusion of the air curtain into the inner part of the model will be less. The front angle was measured from the vertical in a forward direction and the rear angle from the vertical in a rearward direction. In the model study, the front air curtain was defined as the air curtain on the end of the model facing into the wind. The rear air curtain was the air curtain on the end away from the wind.

Model Design

Using dimensional analysis principles, for the model to adequately describe the prototype, it is necessary that each Pi-term operate at the same value for both the model and the prototype. Also, to develop an adequate prediction equation, it is necessary to vary only one Pi-term for each series of the test. This is sometimes difficult to do. However, using the preliminary test data as a measurement of air curtain velocity the model procedure presented in Table III was arranged. Initially, the front air curtain was operated at 30 degrees and the rear one at 60 degrees. As the tests proceeded, it became apparent that 45 degrees was the optimum angle for both. The test procedure was modified to incorporate this change.

In the test procedure, tests 3-1 to 3-6 are the variation of Pi 5, the front air curtain angle; 3-14 to 3-21 are the variation of Pi 6, the rear air curtain angle; 3-22 to 3-25 are the variation of Pi 2, the air curtain Reynolds number; and 3-26 to 3-30 are the variation of Pi 3, the model Reynolds number. Pi 1, the Prandtl number, and Pi 4, the air curtain thickness to height ratio, were held constant. Tests 3-7 to 3-13 were conducted to determine an equivalent heat transfer coefficient with no air curtains in operation. Tests 3-31 to 3-38 were conducted to determine the effects of wind direction. In tests 3-31 to 3-34, the model was turned to 22 1/2 degrees, then to 45 degrees, to 67 1/2 degrees, and finally to 90 degrees, where the angle is the angle between the direction of the wind to the centerline of the model. As the model is turned at an angle to the wind, an unprotected side region between the end of the side and the air curtain is exposed. A triangular plate was used to cover this region on all four corners, and tests 3-31 to 3-34 were repeated for 3-35 through 3-38 respectively.

The model was constructed of 28 gauge galvanized sheet metal, using a double walled construction to provide an air insulation of one-half inch to minimize heat loss through the walls. The size of the model

TABLE III

TEST PROCEDURE

TEST NUMBER	Pi 1	Pi 2	Pi 3	Pi 4	Pi 5	Pi 6
3-1	.71	7720	30966	.05	0.0000	0.7854
3-2	.71	7720	30966	.05	0.2618	0.7854
3-3	.71	7720	30966	.05	0.5236	0.7854
3-4	.71	7720	30966	.05	0.7854	0.7854
3-5	.71	7720	30966	.05	1.0472	0.7854
3-6	.71	7720	30966	.05	1.3090	0.7854
3-7	.71	0	22551	.05	----	----
3-8	.71	0	26682	.05	----	----
3-9	.71	0	30966	.05	----	----
3-10	.71	0	36361	.05	----	----
3-11	.71	0	43669	.05	----	----
3-12	.71	0	50424	.05	----	----
3-13	.71	0	50662	.05	----	----
3-14	.71	7720	30966	.05	0.7854	0.0000
3-15	.71	7720	30966	.05	0.7854	0.2618
3-16	.71	7720	30966	.05	0.7854	0.5236
3-17	.71	7720	30966	.05	0.7854	0.7854
3-18	.71	7720	30966	.05	0.7854	1.0472
3-19	.71	7720	30966	.05	0.7854	1.3090
3-20	.71	7720	30966	.05	0.7854	1.5708
3-21	.71	7720	30966	.05	0.7854	1.8326
3-22	.71	3088	30966	.05	0.7854	0.7854
3-23	.71	4632	30966	.05	0.7854	0.7854
3-24	.71	6176	30966	.05	0.7854	0.7854
3-25	.71	9264	30966	.05	0.7854	0.7854
3-26	.71	7720	22551	.05	0.7854	0.7854
3-27	.71	7720	25582	.05	0.7854	0.7854
3-28	.71	7720	36707	.05	0.7854	0.7854
3-29	.71	7720	43945	.05	0.7854	0.7854
3-30	.71	7720	50424	.05	0.7854	0.7854
3-31	.71	7720	30966	.05	0.7854	0.7854
3-32	.71	7720	30966	.05	0.7854	0.7854
3-33	.71	7720	30966	.05	0.7854	0.7854
3-34	.71	7720	30966	.05	0.7854	0.7854
3-35	.71	7720	30966	.05	0.7854	0.7854
3-36	.71	7720	30966	.05	0.7854	0.7854
3-37	.71	7720	30966	.05	0.7854	0.7854
3-38	.71	7720	30966	.05	0.7854	0.7854

was determined from the heating chamber dimensions of the prototype. A scale of one eighth was used.

Temperatures were recorded at four points within the model and an external ambient temperature point. Recording was done on a Honeywell Electronik 16 multipoint recorder.

Each test was allowed to reach equilibrium conditions and six temperature readings were taken at each recording point for each test.

The location of the four temperature recording points inside the model is indicated in Figure 3. An average internal temperature was determined in the following manner. The three rearward points were averaged to determine a mean rear temperature. Then the front temperature and the mean rear temperature were averaged to determine an internal average temperature. Using the external ambient temperature, average temperature, and the heat input of the resistance heater, a value for the heat transfer coefficient was determined according to the following formula:

$$h = q/Ht_m.$$

In this equation, q is the heat input per unit length of air curtain, H is the air curtain height, t_m is the temperature difference of the external ambient temperature and the internal average temperature, and h is the heat transfer coefficient.

The wind velocity was measured using a Dwyer manometer and pitot tube arrangement. The velocity was changed using a variable speed drive on the wind tunnel drive.

The front and rear air curtain angles were changed manually for each setting. Figures 4 and 5 show views of the model as it was placed in the wind tunnel.

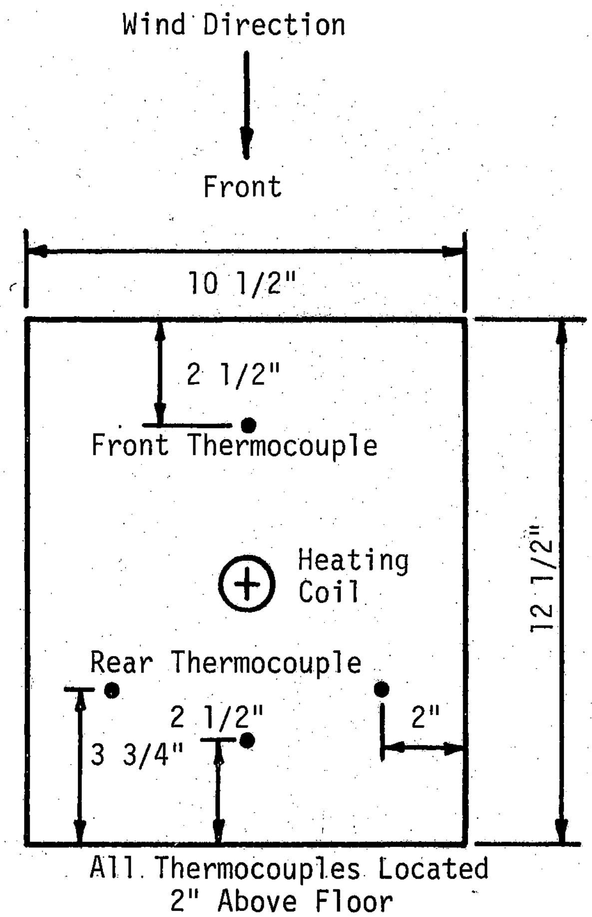

Figure 3. Temperature Measuring Points of Model (Top View)

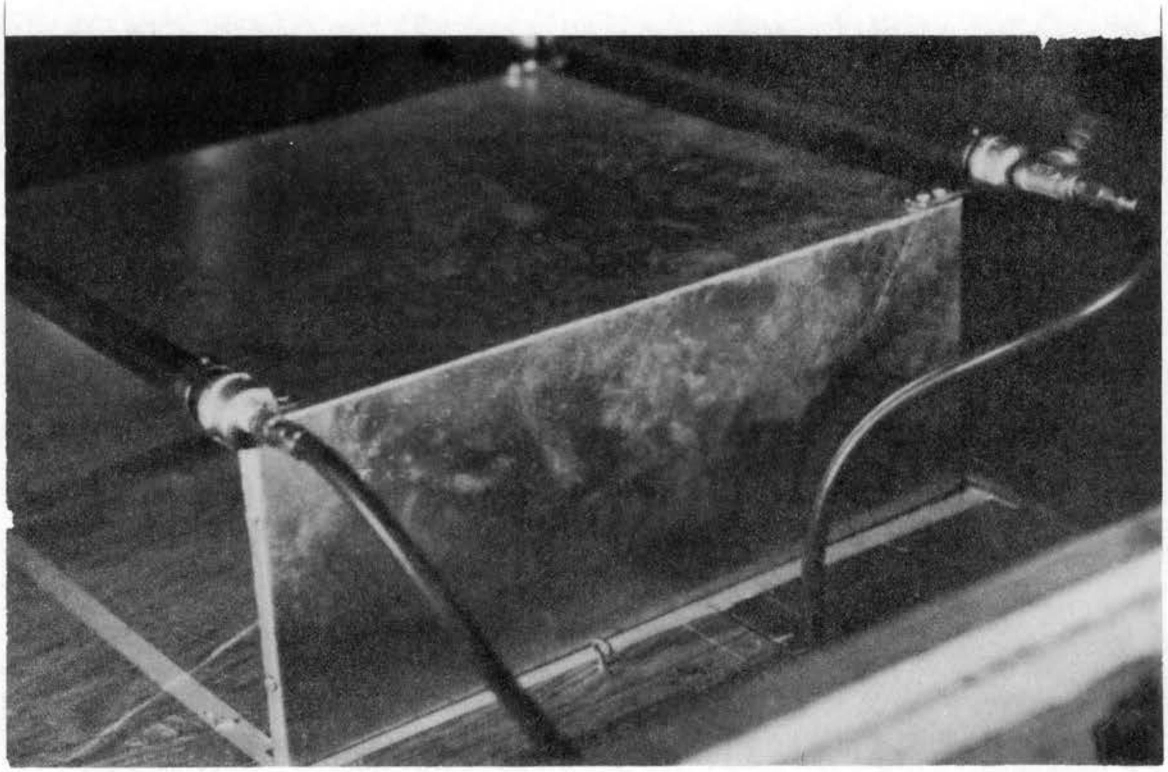

Figure 4. Left Side of Model, as Tested in Wind Tunnel, Without Corner Shields and With 0° Lead Angle.

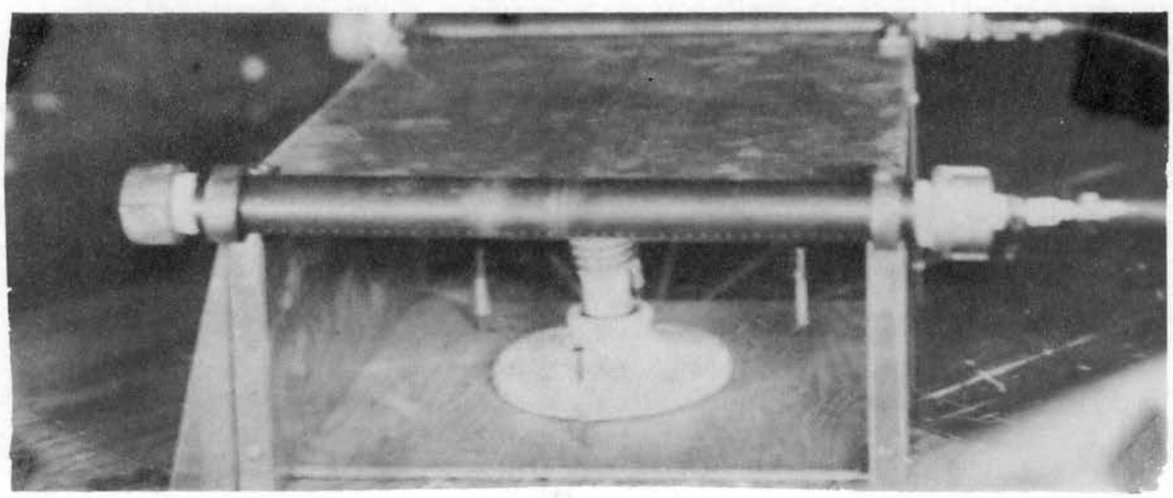

Figure 5. Left Side of Model, With Corner Shields, And With 45° Lead Angle.

CHAPTER V

MODEL RESULTS

Development of the Air Curtain

Preliminary tests were carried out using one inch diameter pipe with small holes as an air curtain generator. Different combinations of number and diameter of holes were tested. Figures 6, 7, and 8 are graphs showing results of the tests. Figure 6 shows the variation of velocity with length for a pipe with 23 holes of .0225 inch diameter. Hole number one is closest to the entrance of the compressed air. Figure 7 shows the variation of velocity with length for 46 holes of .0225 inch diameter. Again, hole number one is closest to the entrance of the compressed air. The length of these pipes is 18 inches. Comparison of Figures 6 and 7 will show that there is not a significant change in velocity with an increase in the number of holes, but with a larger number of holes, there is less variation of velocity from one end of the pipe to the other. On the model, the pipes were 12 inches long, but similar results would be expected. Figure 8 is a graphical representation of the pressures and velocities used in further model studies with the 12 inch pipes.

Appendix A, Tables A-I and A-II present the results of the two test series conducted with the two pipes described above, and Appendix A, Table A-III is the results for the 12 inch pipe. Figures 6, 7, and 8 are based on a distance of two inches from the pipe, but Appendix A,

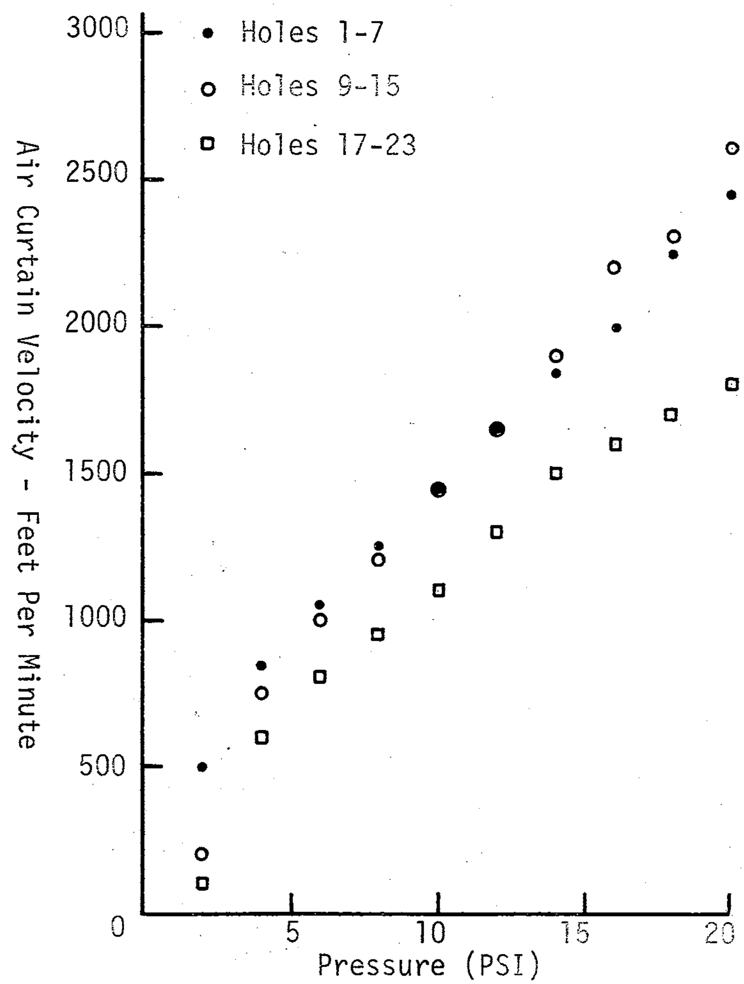

Figure 6. Air Curtain Velocity vs. Pressure, 23 Hole Pipe

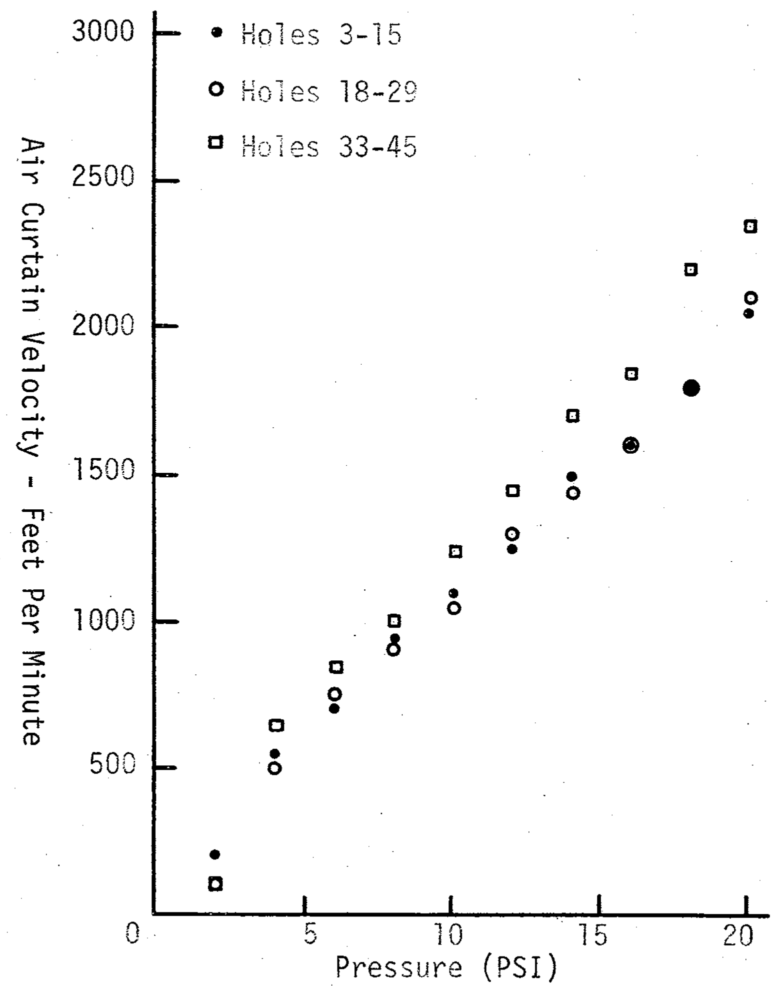

Figure 7. Air Curtain Velocity vs. Pressure, 45 Hole Pipe

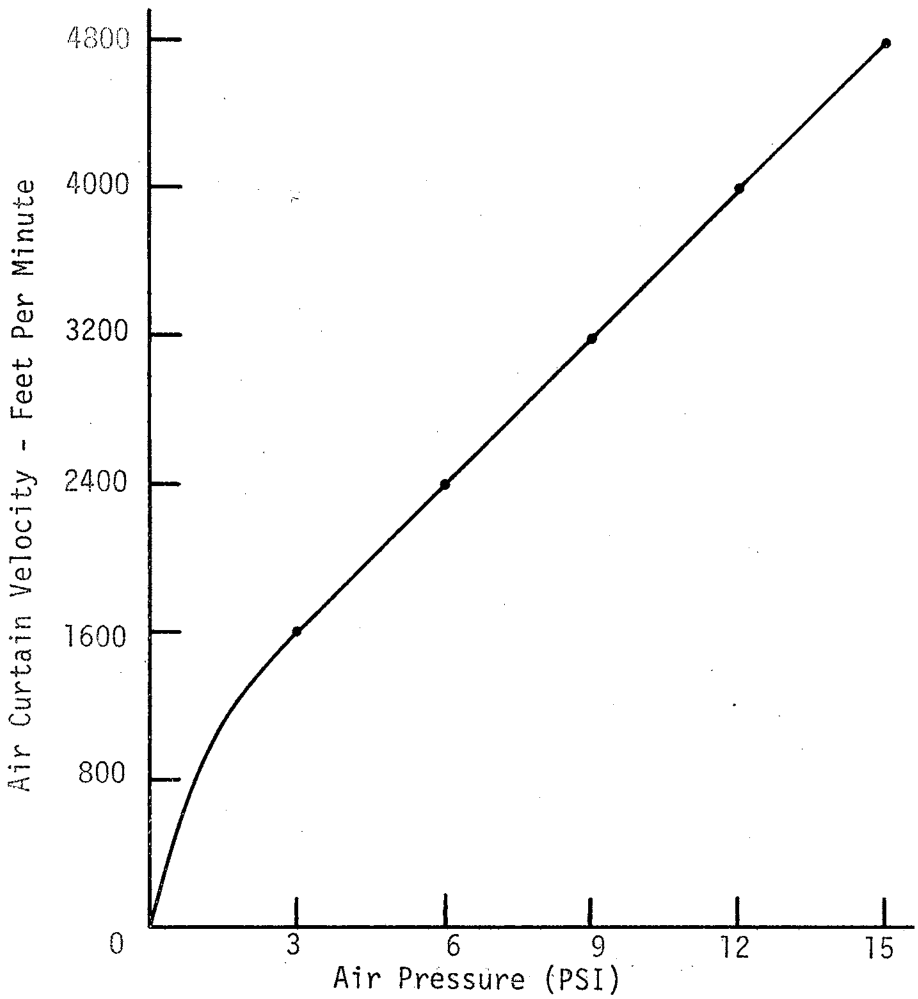

Figure 8. Air Curtain Velocity vs. Pressure
48 Hole, 12 Inch Length Pipe

Tables A-I and A-II present data for distances of one inch, two inches, three inches, and five inches from the pipe, as well as variation along the length of the pipe. Appendix A, Table A-III is for two inches only, with average velocities presented. Measurements for variation along the length of the pipe were made using three sections of the pipe, and not a measurement of each individual hole.

Model Studies

After development of the air curtains using the pipes and determination of an appropriate means of measuring air curtain velocity using an air pressure gauge, the model studies were performed as outlined in Chapter IV. The results of the model studies are tabulated in Table IV, and Appendix A, Table A-IV shows the original test conditions, along with the ambient, front, and mean rear temperatures as recorded.

Using the data from the model studies and a computer program with logarithmic transformation, a prediction equation of the form:

$$Pi_7 = C_0 \times Pi_2^{A_2} \times Pi_3^{A_3} \times Pi_5^{A_5} \times Pi_6^{A_6}$$

was found. Values for the constants were:

$$C_0 = .2185 \times 10^{-2},$$
$$A_2 = -0.5084,$$
$$A_3 = +1.5376,$$
$$A_5 = -0.1397,$$
$$A_6 = +0.0436.$$

A correlation coefficient of .9420 was achieved.

Figure 9 is a graphical representation of Pi_7, the Nusselt number, versus Pi_2, the air curtain Reynolds number; Figure 10 is a graphical representation of Pi_7 versus Pi_3, the model Reynolds number; Figure 11

TABLE IV

MODEL STUDY RESULTS

TEST NUMBER	NUSSELT NUMBER	TEST NUMBER	NUSSELT NUMBER
3-1	769.3	3-20	205.4
3-2	298.2	3-21	237.3
3-3	221.5	3-22	528.1
3-4	160.1	3-23	403.2
3-5	145.5	3-24	202.4
3-6	112.8	3-25	209.0
3-7	301.0	3-26	172.9
3-8	408.4	3-27	180.0
3-9	564.5	3-28	339.8
3-10	746.0	3-29	952.9
3-11	1266.0	3-30	1486.9
3-12	1684.8	3-31	273.9
3-13	3165.0	3-32	226.9
3-14	126.2	3-33	90.0
3-15	185.3	3-34	99.1
3-16	196.4	3-35	249.5
3-17	143.4	3-36	237.8
3-18	189.1	3-37	179.9
3-19	201.2	3-38	161.4

is a graphical representation of Pi_7 versus Pi_5, the front air curtain angle; and Figure 12 is a graphical representation of Pi_7 versus Pi_6, the rear air curtain angle. Also shown of Figure 10 is a representation of model experiments 3-7 through 3-13. These experiments were performed with no air curtain and variable amounts of wind. This gives an indication of the improvement of performance attributable to the air curtain. Figure 13 is a representation of model experiments 3-31 to 3-38. These experiments were performed with the model turned at various angles to the direction of the wind. Also, shown on this figure is the effect of operating with the triangular corner shields in place and removed.

Figure 9 shows the variation of the Nusselt number with the variation of the air curtain Reynolds number. It can also be assumed to be a representation of the heat transfer coefficient versus the air curtain velocity. For the air curtain to be effective, the outward component of the air curtain must be equal to the opposing inward wind velocity. For this reason, a higher air curtain velocity should result in improved effectiveness.

Figure 10 is a representation of the Nusselt number as a function of model Reynolds number, or it is the same as the heat transfer coefficient as a function of wind speed. With increasing wind velocity, a greater component of the air curtain is necessary to resist the inward component, or with a fixed curtain velocity and angle, a greater part of the wind will penetrate the curtain, resulting in an increasing Nusselt number.

Comparison of the two curves shown in Figure 10 shows that for higher wind velocities, the two curves are not widely separated. For a fixed outward curtain component and increasing wind velocity, the amount

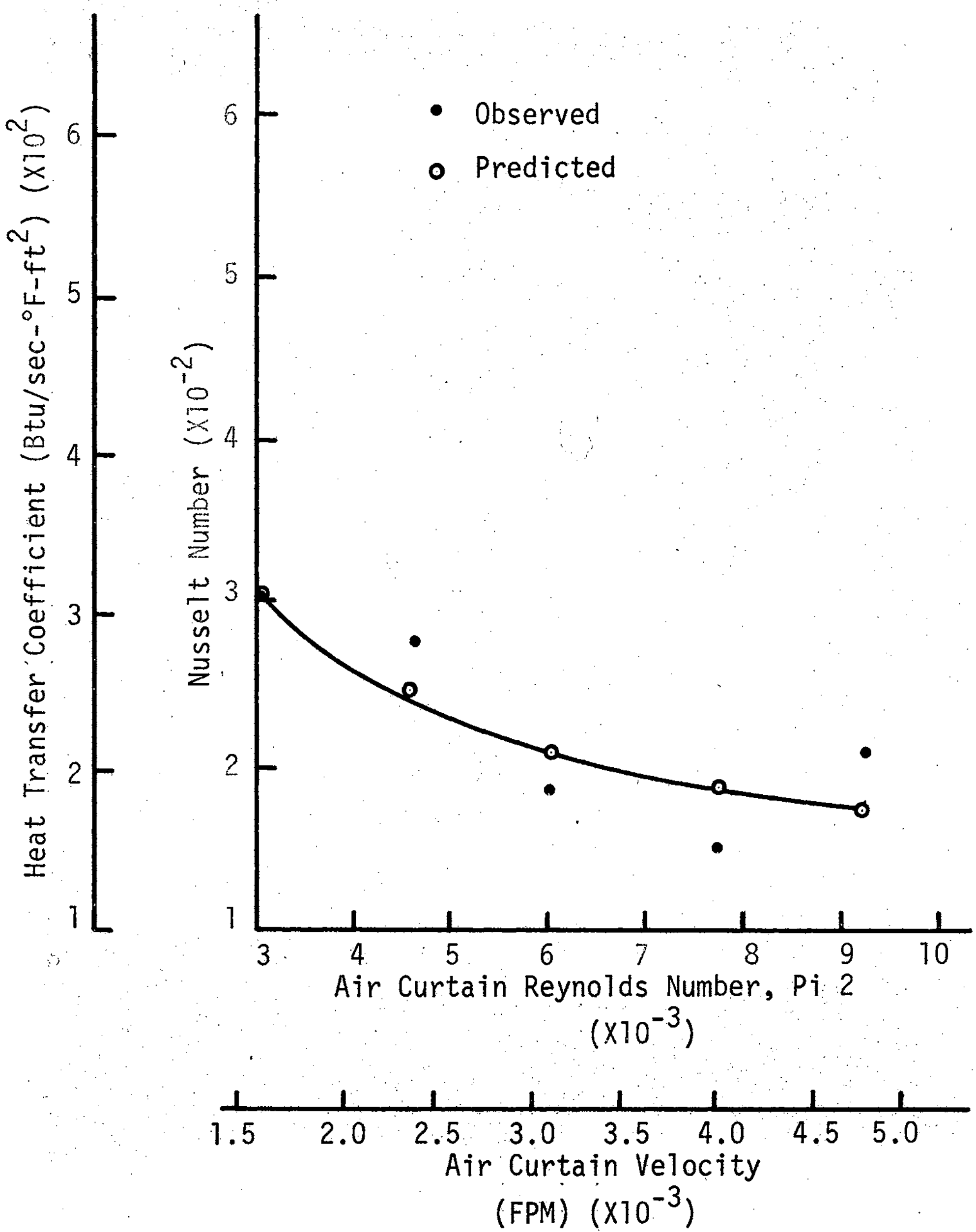

Figure 9. Pi 7 vs. Pi 2,
Nusselt Number vs. Air Curtain Reynolds Number

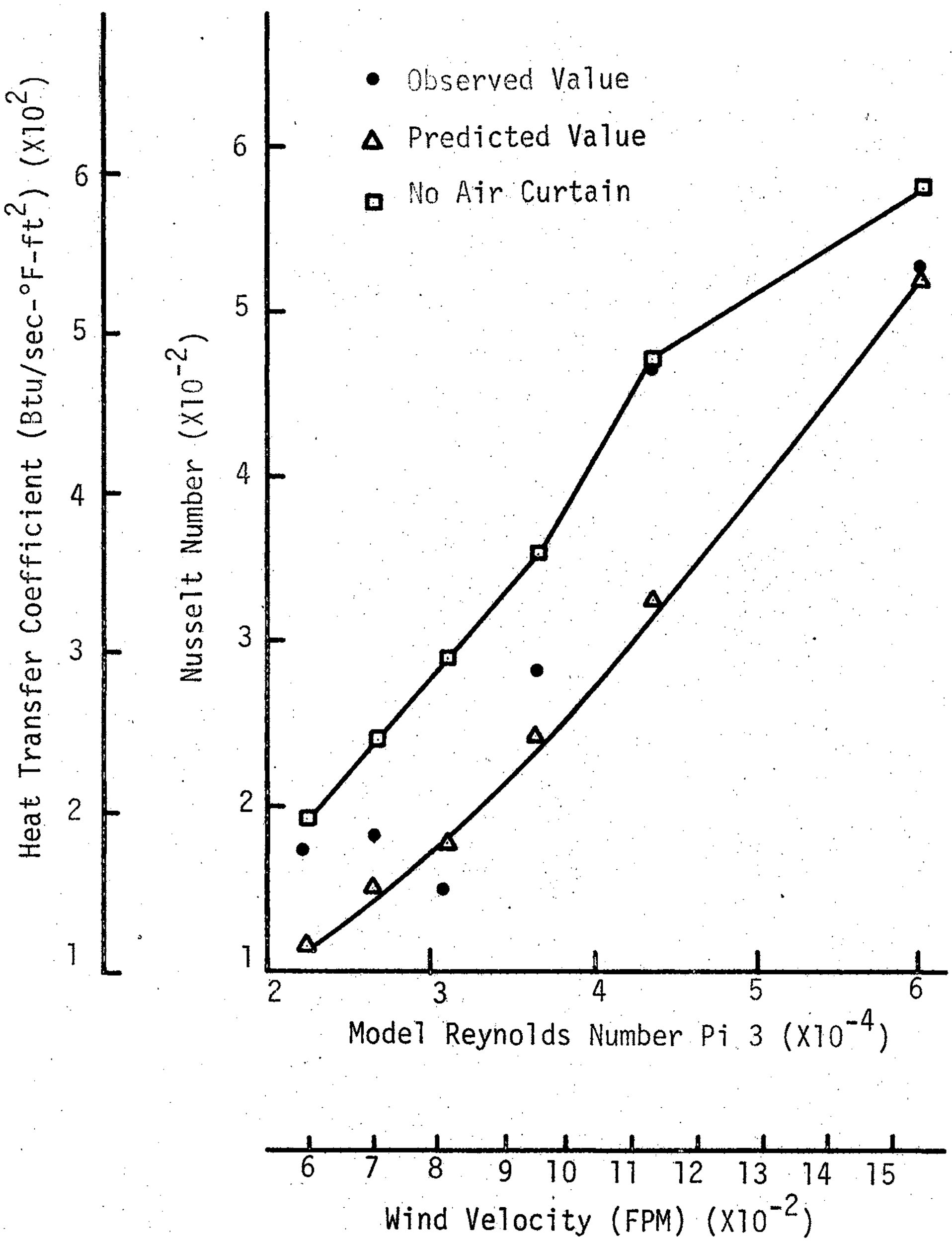

Figure 10. Pi 7 vs. Pi 3,
Nusselt Number vs. Model Reynolds Number

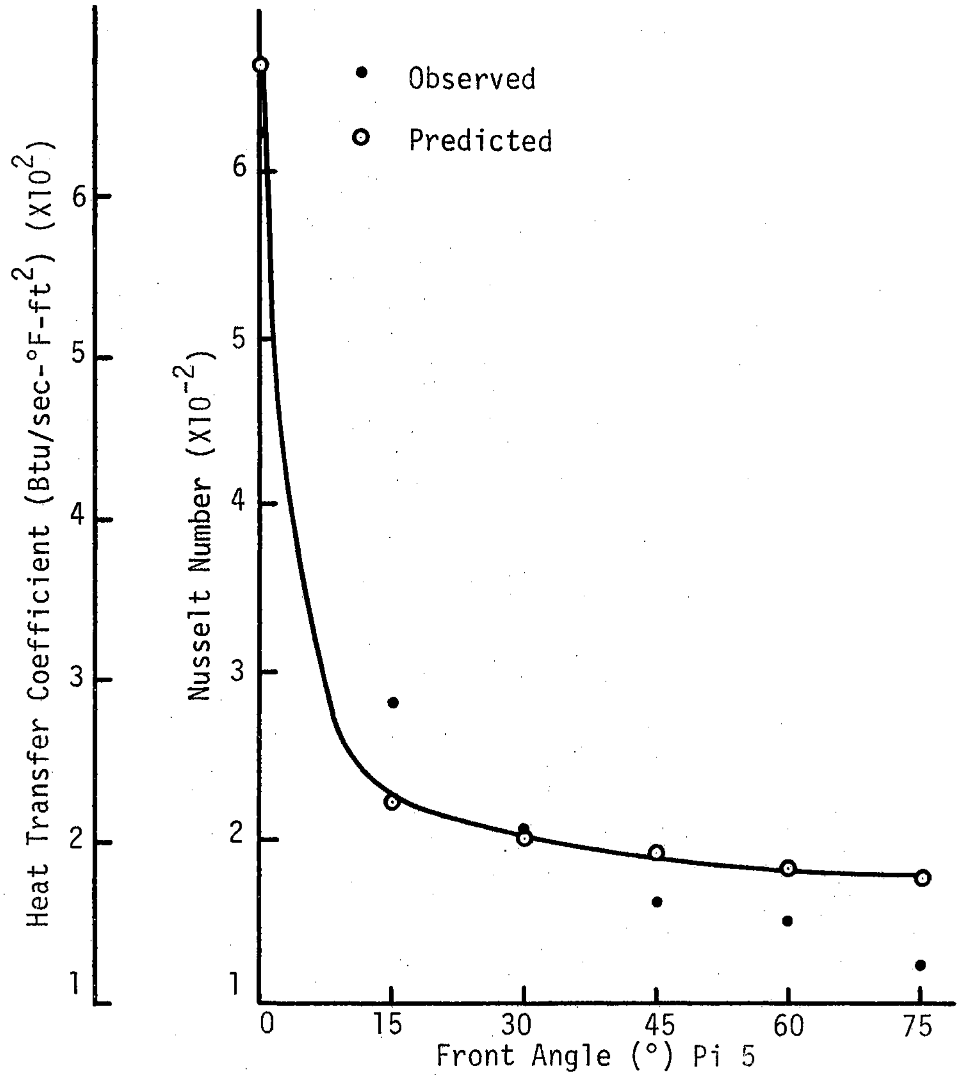

Figure 11. Pi 7 vs. Pi 5,
Nusselt Number vs. Front Air Curtain Angle

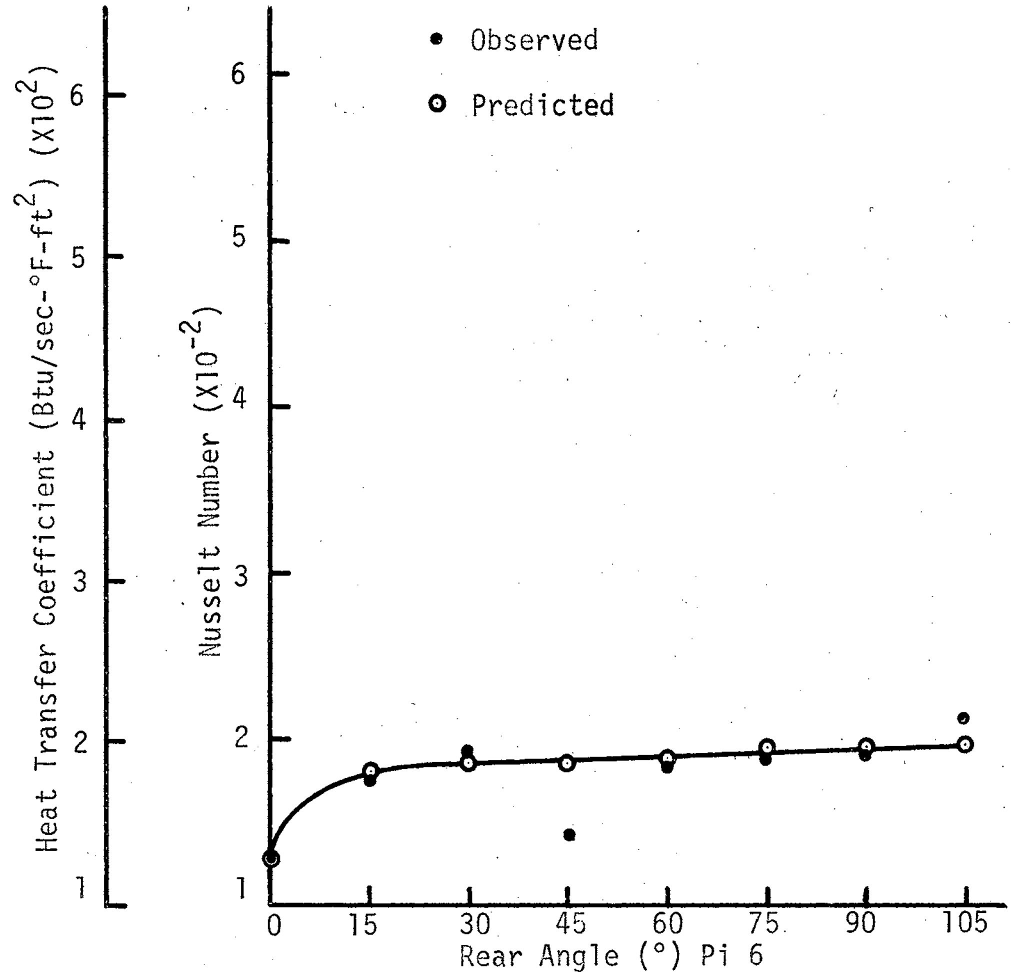

Figure 12. Pi 7 vs. Pi 6,
Nusselt Number vs. Rear Air Curtain Angle

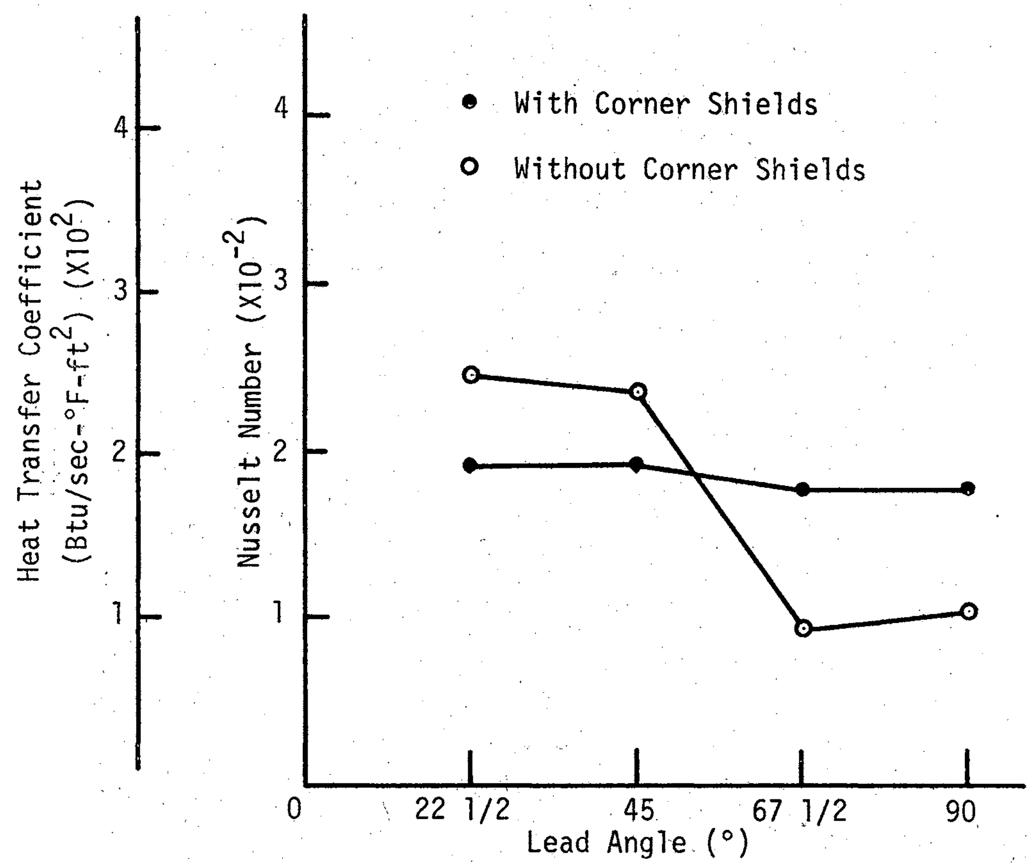

Figure 13. Nusselt Number vs. Lead Angle

of wind penetrating the curtain will increase, also. The two curves will exhibit a similar shape. The difference between operation with no air curtain and with the air curtains operating against a high velocity wind is attributable to the decrease in inward velocity of the wind caused by the outward component of the air curtain.

Figure 11 shows the variation of the Nusselt number with the front air curtain angle. For very small angles, there is almost no outward component to resist the wind. As the angle is increased, the amount of outward component will change rapidly at first, and then, at about 45 degrees, begin to become more constant. This is indicated by the rapid decrease of the Nusselt number for low angles, and the relative constancy for larger angles.

Figure 12 shows the variation of the Nusselt number with the rear air curtain angle. Since there is not a direct wind blowing against this air curtain, its primary purpose is to keep the heated air from escaping the heating chamber. As the graph indicates, this is best accomplished at lower angles. As the angle increases, the air curtain will leave an open space behind the model in which turbulence and low pressure areas are developed, resulting in an increase of the Nusselt number.

Figure 13 is a graph of the Nusselt number as a function of the lead angle, or the angle of the centerline of the model with the direction of the wind. With the triangular shields in place, a more stable arrangement is present because there is less disrupting air entering or leaving at each corner. For operation without the shields, the heat transfer coefficient is large for the smaller angles because the wind is able to enter or leave at each corner. For the larger angles, there is a tendency for the wind to blow on past the open ends, causing the heat transfer coefficient to decrease. Also, for operation at an angle other than zero, it is possible that the temperature points used will no longer give an accurate temperature profile.

CHAPTER VI

PROTOTYPE DESIGN

The prototype unit used in this study consisted of a two row defoliation unit mounted on a high clearance tractor. Figures 14, 15, 16, and 17 show the various views of the prototype. Design and operation of the original machine was described by Perry (1) on pages 13 to 28. The air envelope effect which was achieved was considered a desirable arrangement, and attempts were made to maintain this effect in the modified version. The changes made can be classified into four categories: (1) removal of 10 inches from the bottom of the sides of the heating chamber and the use of sash chain to form a flexible curtain at the bottom of the unit; (2) movement of the fresh air inlet on the burner fan from the rear of the fan to the top; (3) removal of front and rear doors on the machine and replacement with air curtains; and (4) removal of the hydraulic drive on the burner fan and replacement with a mechanical drive.

Operation of the original machine required that the heating chamber be maintained as near to the ground as possible. In rough terrain and crossing irregular surfaces, the bottom of the heating chamber would often contact the ground. Considering the fact that the hot air tends to rise quickly into the upper portion of the chamber, it seemed reasonable that some sort of flexible curtain would be adequate to maintain the internal configuration of the chamber and also to restrict entrance of

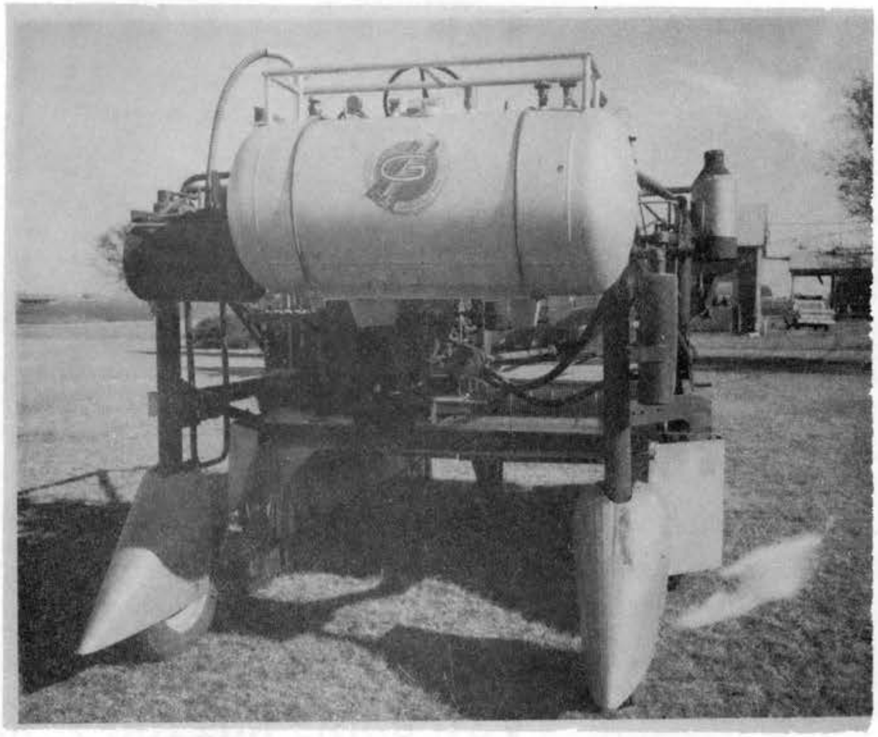

Figure 14. Front View of Thermal Defoliation Machine.

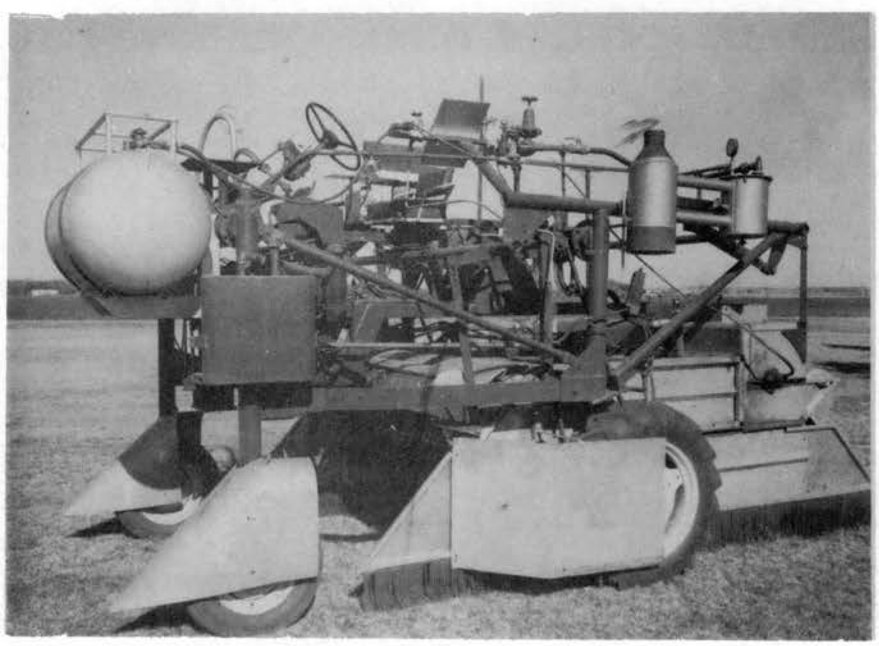

Figure 15. Left Side of Thermal Defoliation Machine

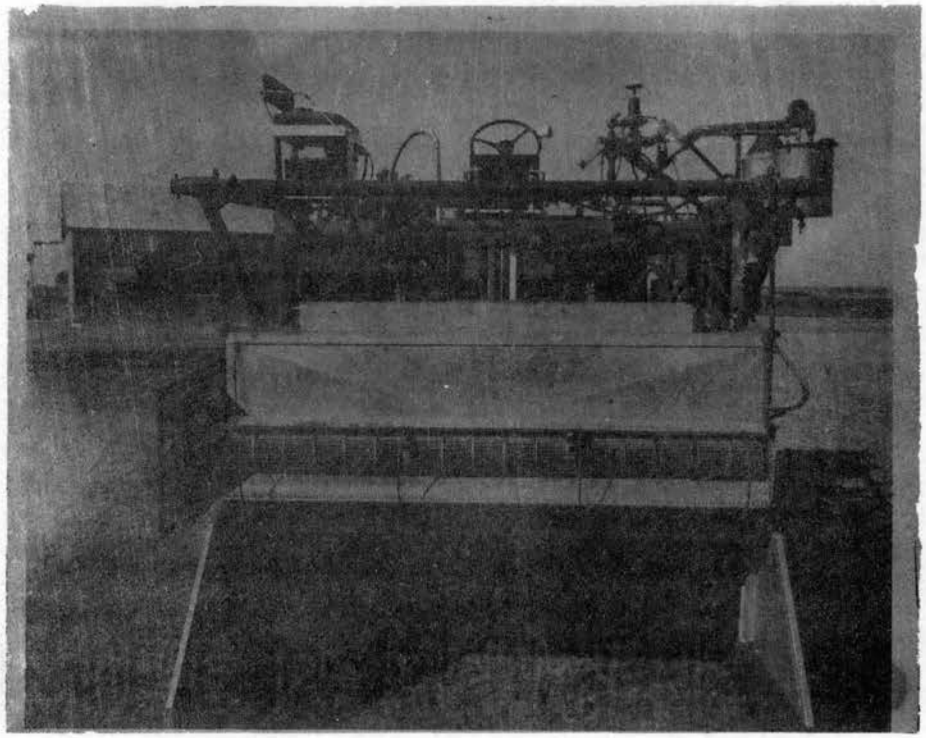

Figure 16. Rear View of Thermal Defoliation Machine.

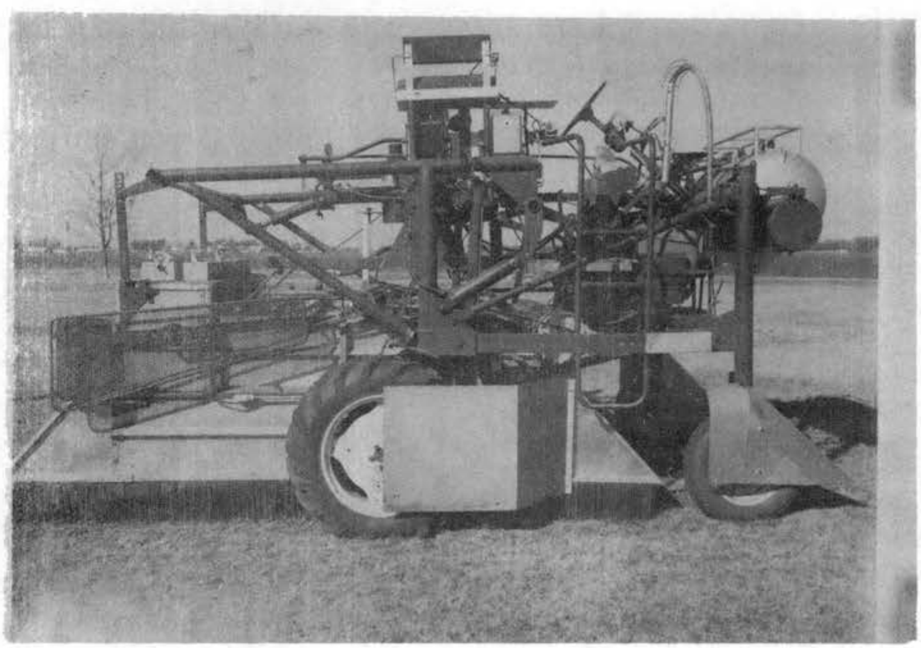

Figure 17. Right Side of Thermal Defoliation Machine

cool air from the exterior. Also, when the machine is operating in the field, plants to the side of the machine tend to contribute to this effect. Sash chain was chosen because it could be easily fastened to the lower edge of the heating chamber.

Movement of the fresh air inlet from the rear to the top of the burner fan was done simply because the rear air curtain fan was to be mounted on the rear. Air inlets for both fans would be close together, and possible disruption of flow in one fan or the other could result.

Removal of the front and rear doors of the heating chamber was part of the major change with which this study was concerned. In their place, two long transverse flow fans were used to develop air curtains which were placed so that they enclosed the ends, but presented an unobstructed entrance or exit for plants. This type of fan is unique in that it draws the air in radially on one side of the fan wheel and exhausts it radially at a different location, depending on the shroud arrangement. Each air curtain fan was approximately 80 inches long, using a single shaft of 5/8 inch diameter. Design of the fans was based on design procedure outlined by Whitney (17, 18), with 2000 feet per minute as the design velocity. According to the design procedure, six inch diameter fan wheels operating at 1200 rpm would give the desired results. The shrouding was constructed of 24 gauge galvanized sheet metal. A D- or Datwyler-type shrouding was used because it allowed the most desirable configuration of entrance and exit. No directional ducting or guide vanes were used on the exterior of the fan, as proper orientation of the fan itself would achieve the desired direction. Figure 16 shows the transverse flow fan used at the rear of the machine. Mounting on the front is similar.

In the model studies, the heat transfer coefficient decreased as the front angle increased until about 45 degrees was reached. For the rear angle, the heat transfer coefficient increased until about 45 degrees was reached. For angles greater than 45 degrees, the heat transfer coefficient was relatively constant, in both cases. In the model study, the front air curtain was defined as the air curtain on the end of the model facing into the wind, and the rear air curtain was the one on the end away from the wind. In field operation, it is extremely difficult to keep only one end of the machine facing into the wind. For example, the machine may be traveling in the same direction as the wind, but the wind may be moving at a faster rate than the machine. In this case, the air curtain on the rear of the machine would be the front air curtain, according to the model definition.

Considering these facts, that the front and rear air curtains have relatively constant heat transfer coefficients for angles of 45 degrees or greater, and that either the front or the rear air curtain on the machine could correspond to the front air curtain on the model, it was decided to mount both the front and rear air curtains to discharge at approximately 45 degrees downward and outward from the machine entrance and exit. This seemed to be the best compromise which could be reached. Also, as on the model, triangular shields were placed at each corner in an attempt to make a continuous boundary around the heating chamber.

On the original machine, a hydraulic pump and motor arrangement was used to provide a variable speed fan drive. On the modified machine, it seemed desirable to operate all three fans simultaneously. For this reason, the hydraulic fan drive was removed and in its place a mechanical drive arrangement was used. The mechanical drive consisted of a manually operated clutch on the front of the tractor motor, connected with double V-belts to a driveshaft. The driveshaft contained two universal joints, as the heating chamber could still be raised and lowered to facilitate road travel. The driveshaft was connected to a 90-degree gearbox mounted on the top of the heating chamber. A power shaft was run from the gearbox to the side of the heating chamber. V-belts and pulleys were used on the power shaft to connect it to the three fans, with the necessary pulley diameters to obtain correct speeds, and in the case of the rear air curtain, idler pulleys to obtain correct direction of rotation. In Figure 17, the V-belt and pulley arrangement is visible. In some tests performed on the prototype, it was necessary to stop one or both of the air curtain fans and continue to operate the burner fan. This was accomplished using bearings in the air curtain drive pulleys, with a key arrangement to transmit power when desirable.

CHAPTER VII

PROTOTYPE RESULTS

Three different types of tests were performed using the modified machine: (1) field tests to determine if plant response would show a significant difference between operation of the machine with the air curtains functioning and with the air curtains inoperative, (2) tests using air velocity and temperature readings to determine if an air curtain effect was achieved, and (3) tests to determine maximum temperature and elevated temperature period that the cotton plant experiences.

Field Tests

Two independent field tests were performed. One test was more severe in the treatment applied than the other. In the more severe test, a fuel pressure of 20 psi was used, the wind was blowing from the north at approximately 200 feet per minute, and a ground speed of 2 3/4 miles per hour was used. In the less severe test, fuel pressure was 20 psi, the wind was from the north at approximately 400 feet per minute, and a ground speed of 3 miles per hour was used.

Initial operation of the prototype showed that operation traveling in the same direction as the wind, or away from the wind, resulted in different performance than operation traveling in a direction opposite to the wind direction, or into the wind. It was fortunate that on the two days that field tests were performed, the wind was from the same direction. The four treatments considered were as follows: (1) with air curtain fans, into the wind, (2) without air curtain fans, into wind, (3) with air curtain fans, away from wind, and (4) without air curtain fans, away from wind. Each treatment was repeated seven times for each test. To evaluate each replicate, four plants were selected at random near the middle of each test block, two in each row, and the total number of leaves counted. Seven days after each test, the total number of leaves, and the total number of green leaves remaining were counted. The percent defoliation and percent kill were computed for each replicate according to the following formulae:

$$\% \text{ defoliation} = \frac{\left\{\begin{matrix}\text{Number leaves}\\ \text{before treatment}\end{matrix}\right\} - \left\{\begin{matrix}\text{Number leaves}\\ \text{after treatment}\end{matrix}\right\}}{(\text{Number leaves before treatment})} \times 100$$

$$\% \text{ kill} = \frac{\left\{\begin{matrix}\text{Number leaves}\\ \text{before treatment}\end{matrix}\right\} - \left\{\begin{matrix}\text{Number green leaves}\\ \text{after treatment}\end{matrix}\right\}}{(\text{Number leaves before treatment})} \times 100$$

Figure 18 is a graphical representation of the results of test one, the more severe test, showing percent kill and percent defoliation. Figure 19 is a graphical representation of test two, the less severe test. Table V presents the analysis of variance for test one, and table VI the analysis of variance for test two. Appendix B, Tables B-I and B-II present the original field data collected for tests one and two respectively. Also, presented at the bottom of Tables V and VI is the Duncan's multiple range test for significant difference of each test. The only detectable significant difference was between treatments two and three for percent kill of test two. This difference was indicated in only one of the four possible criterion.

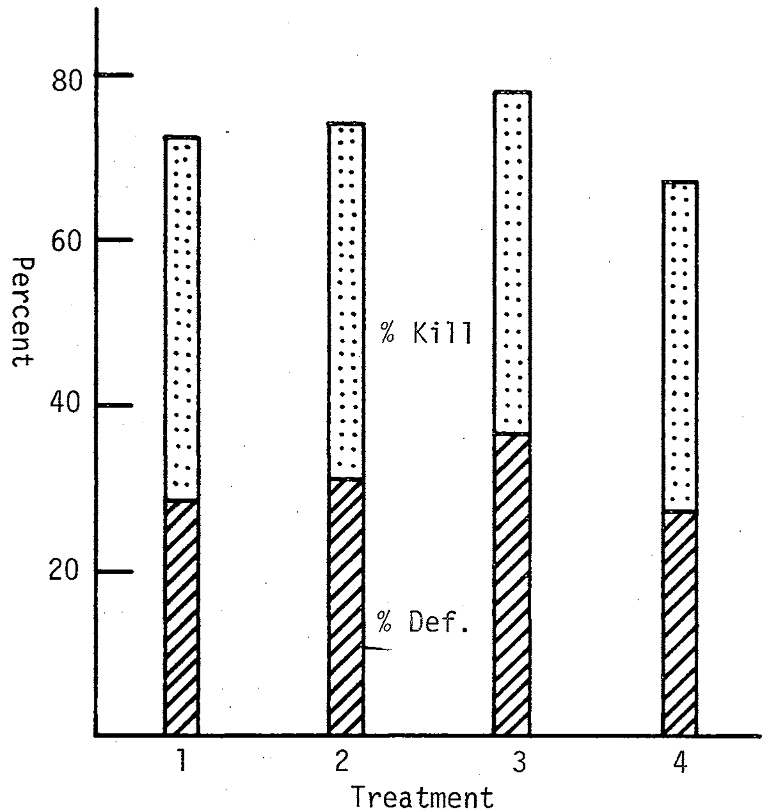

Figure 18. Field Test 1 Results

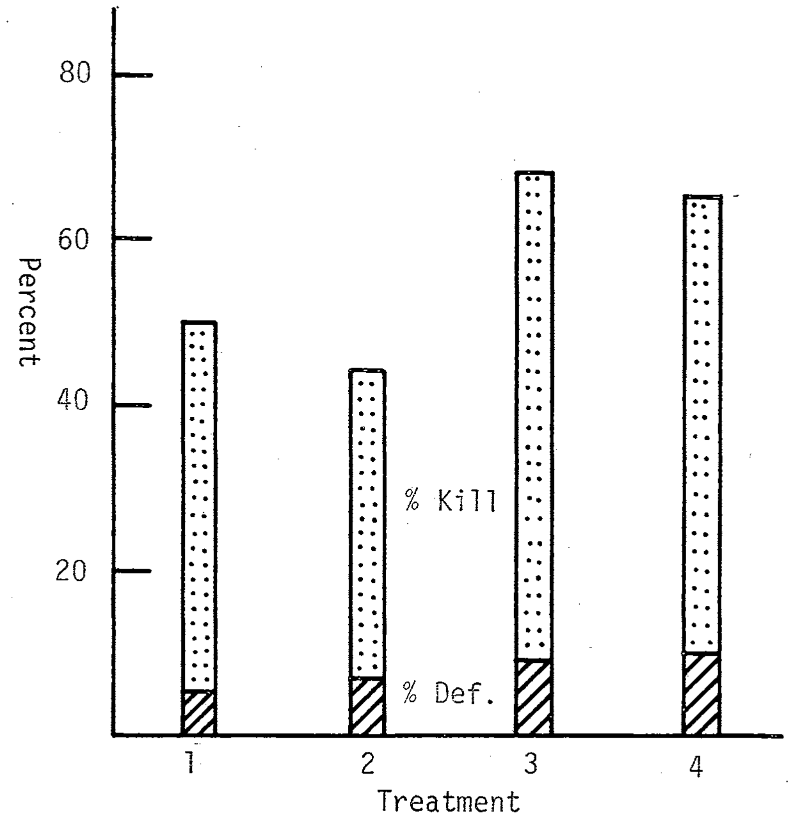

Figure 19. Field Test 2 Results

TABLE V

ANALYSIS OF VARIANCE, TEST 1

TREATMENTS		% Defoliation 1	2	3	4	% Kill 1	2	3	4
REPLICATE	I	36.42	29.75	37.90	35.10	68.30	71.00	63.62	54.80
	II	21.62	27.35	31.78	21.72	86.15	96.66	73.55	65.50
	III	36.62	41.10	52.58	41.05	65.00	86.57	79.75	91.07
	IV	31.15	19.55	23.62	22.80	87.15	69.72	92.77	59.62
	V	20.50	40.82	54.55	25.12	49.85	55.48	80.30	62.50
	VI	22.35	21.52	25.82	31.25	100.00	60.90	76.90	81.08
	VII	33.52	36.72	32.10	16.48	53.42	81.40	78.92	60.02

A. O. V. (% Defoliation)

SOURCE	df	ss	ms	f
TOTAL	27	2474.7795		
REPS	6	1126.0782	187.6797	3.40
TREATMENTS	3	354.1077	118.0359	2.14
ERROR	18	994.5936	55.2552	

A. O. V. (% Kill)

SOURCE	df	ss	ms	f
TOTAL	27	5253.1723		
REPS	6	1564.7154	260.7859	1.42
TREATMENTS	3	376.8333	125.6111	<1
ERROR	18	3311.6236	183.9790	

DUNCAN'S MULTIPLE RANGE

TREATMENT	4	1	2	3
MEAN	27.65	28.88	30.97	36.91

TREATMENT	4	1	2	3
MEAN	67.80	72.84	74.53	77.97

TABLE VI

ANALYSIS OF VARIANCE, TEST 2

		% Defoliation				% Kill			
TREATMENTS		1	2	3	4	1	2	3	4
REPLICATES	I	4.85	13.35	4.68	12.62	46.82	89.57	52.75	80.55
	II	2.92	16.18	7.85	11.42	53.38	45.87	75.28	83.78
	III	9.57	1.85	15.22	7.22	57.55	34.55	79.45	69.62
	IV	3.80	1.58	6.20	19.22	47.28	19.68	66.67	75.15
	V	6.60	4.50	12.67	7.32	33.97	44.17	71.72	57.02
	VI	8.75	8.17	8.80	5.48	58.50	50.00	77.75	38.37
	VII	3.27	5.70	6.57	3.25	54.10	42.78	53.77	51.98

A. O. V. (% Defoliation)

SOURCE	df	ss	ms	f
TOTAL	27	552.0589		
REPS	3	15.7338	2.6223	<1
TREATMENTS	6	60.9399	20.3133	<1
ERROR	18	475.3852	26.4102	

A. O. V. (% Kill)

SOURCE	df	ss	ms	f
TOTAL	27	7980.7497		
REPS	3	1065.9462	177.6577	<1
TREATMENTS	6	2409.8250	803.2750	3.21
ERROR	18	4504.9785	250.2765	

DUNCANS MULTIPLE RANGE

TREATMENT	1	2	3	4
MEAN	5.68	7.33	8.86	9.50

TREATMENT	2	1	4	3
MEAN	46.66	50.23	65.21	68.20

It seems reasonable that a difference would exist between these two treatments, since they were the least severe and the most severe treatments applied in this test. In treatment two, the wind was blowing into the front of the machine, and this would cause the machine speed to add to the wind speed for an increased relative wind velocity, and there was no air curtain to protect the heating chamber. In treatment three, the machine was traveling with the wind, so the relative velocity would be the difference of the wind speed and the machine speed, and the air curtains were in operation to protect the heating chamber. There are two possible reasons why the other tests did not show any further differences. The wind was blowing very lightly on both days that tests were performed, and perhaps the wind would have to be stronger to indicate any differences. A second explanation is that the air curtains were not able to provide effective protection for the heating chamber when they were in operation.

Surrounding Environment Tests

The surrounding environment tests were performed with the machine stationary, and operating inside the Agricultural Engineering shop at the Cotton Research Station. Several different conditions of operation were studied. A coordinate system was organized on the floor to facilitate the tests. The internal flow of the heating chamber was not included. Figure 20 shows a schematic representation of the coordinate system used. A thermocouple and the probe of a hot wire anemometer were mounted on a moveable stand.

Tests were conducted using only the air curtains with no internal heat or air circulation; with internal circulation, but no heat; and

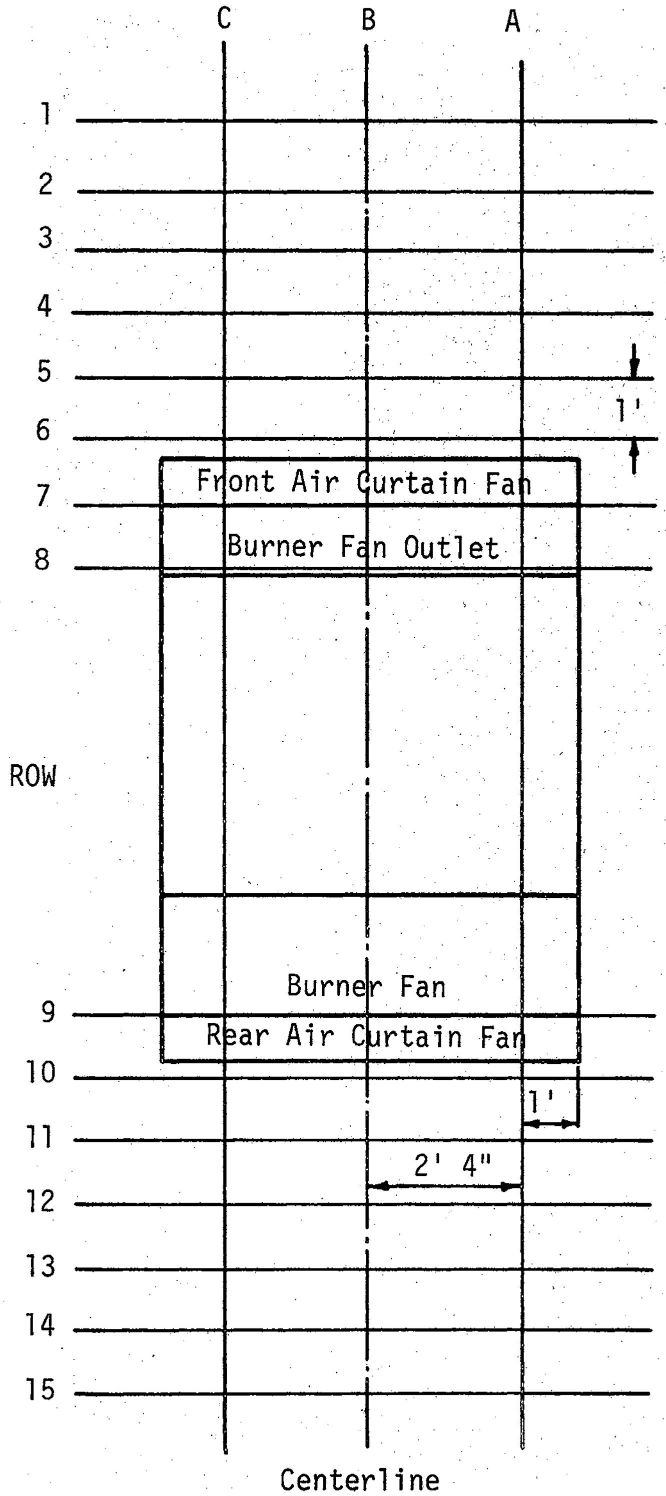

Figure 20. Coordinate Plot for
Air Velocity and Temperature Measurements

with internal circulation and heating. When heated air was used, attempts were made to maintain a maximum internal temperature of 300°F, although one test was conducted at considerably higher temperatures. Also, tests were conducted using only the front or rear fan, and various combinations of heat and internal circulation. The data from these tests is listed in Appendix C. Figures 21, 22, 23, and 24 present an overall view of the results of the environment tests. Figure 21 is the velocity profile of front and rear air curtain operation only. In Figure 22, internal circulation has been added. Figure 23 gives velocity and temperature profiles for operation at 300°F for the complete system. In Figure 24, the internal fan was slowed from 900 rpm to 700 rpm.

In Figure 21, it is readily apparent that the air curtains at both the front and rear are well developed. Both exhibit a well defined region of air with velocity greater than 500 feet per minute. The front curtain is aimed slightly higher than the rear air curtain. This may have been a significant factor in the operation of the machine.

With internal air flow added, as shown in Figure 22, a significant decrease in the air velocities had taken place. Some form of interaction between the fans occured so that the air curtains were greatly disturbed. This disturbance was possibly the form of an outward velocity component created by makeup air which was drawn into the heating chamber by the burner fan.

When internal heating was added to the internal flow pattern, as shown in Figure 23, further disruption occured. This may be attributed to the further increase in volume of a given mass of air as it was heated.

With internal heating, but with less internal circulation, an improvement of the air curtains was noted. Figure 24 shows this effect.

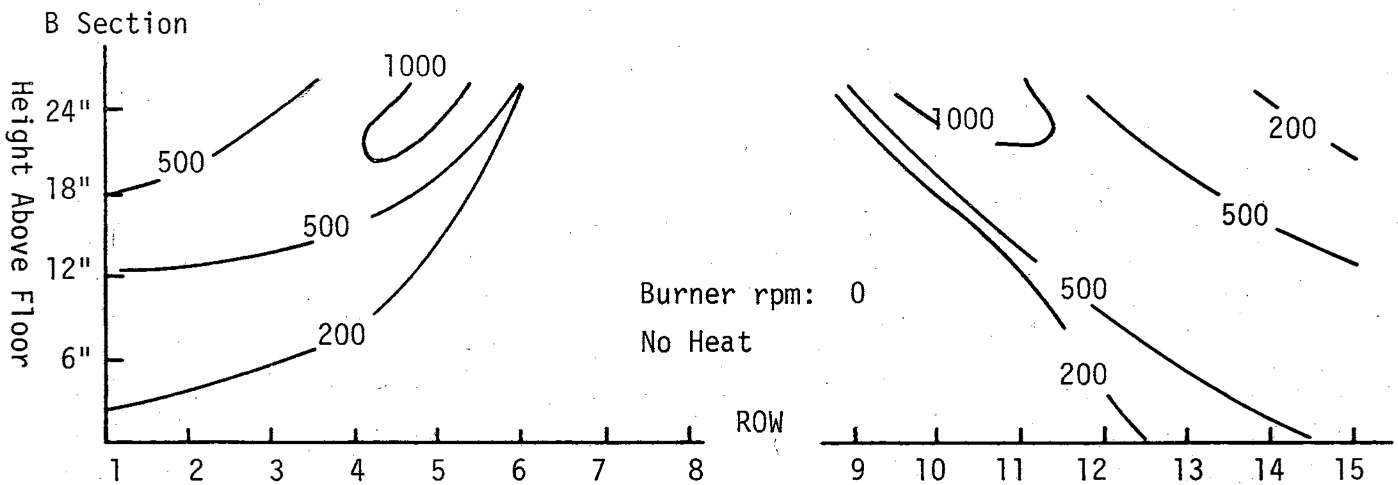

Figure 21. Velocity Profile (FPM) For Air Curtain

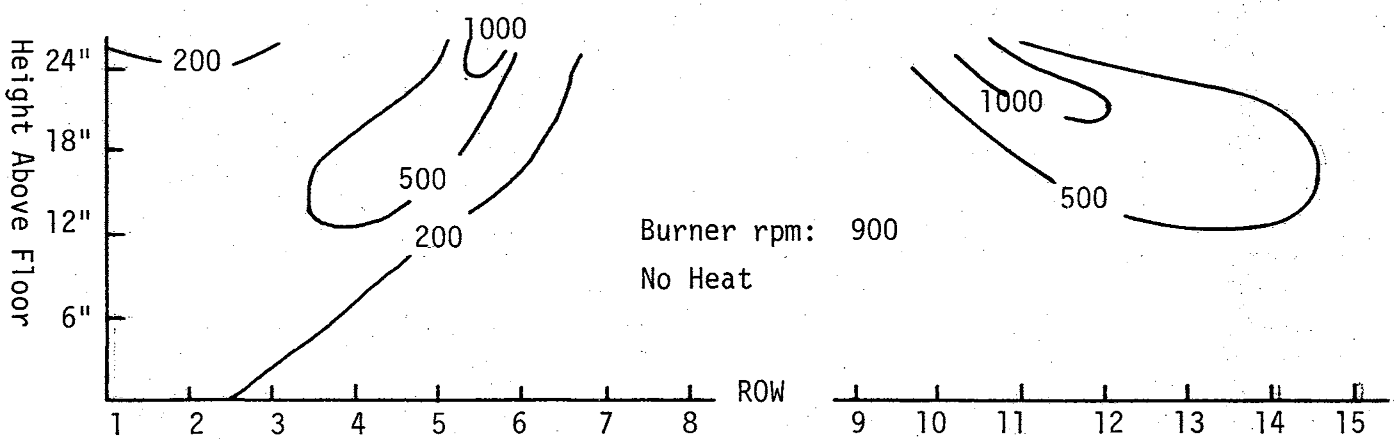

Figure 22. Velocity Profile (FPM) For Air Curtain

B Section

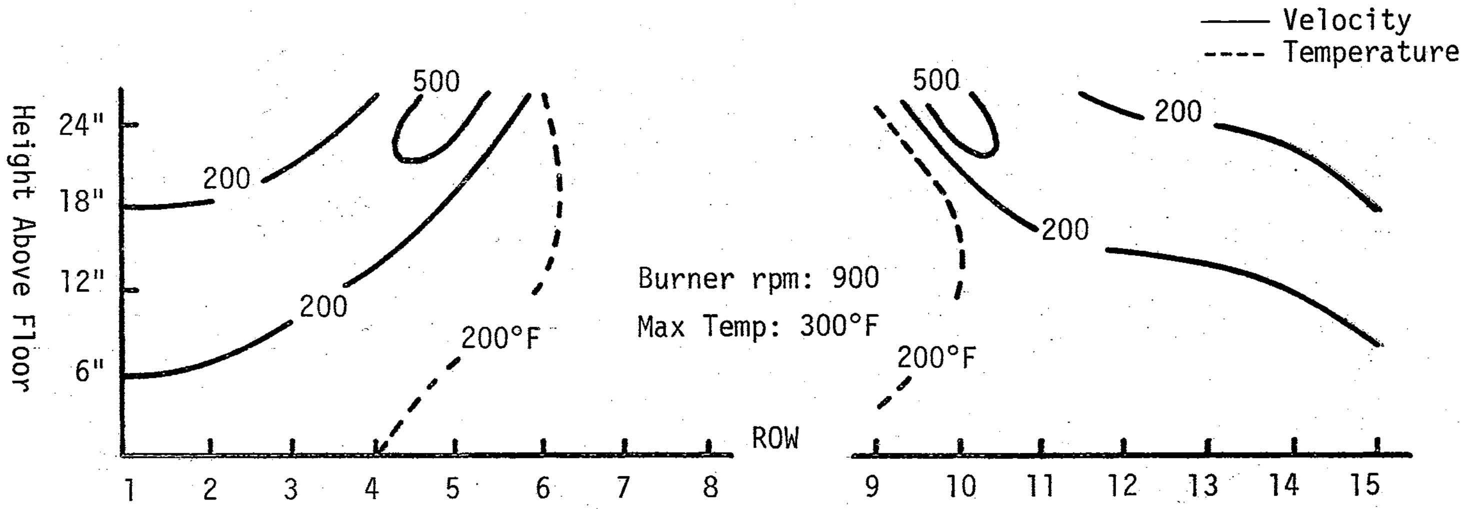

Figure 23. Velocity (FPM) and Temperature Profile for Air Curtain

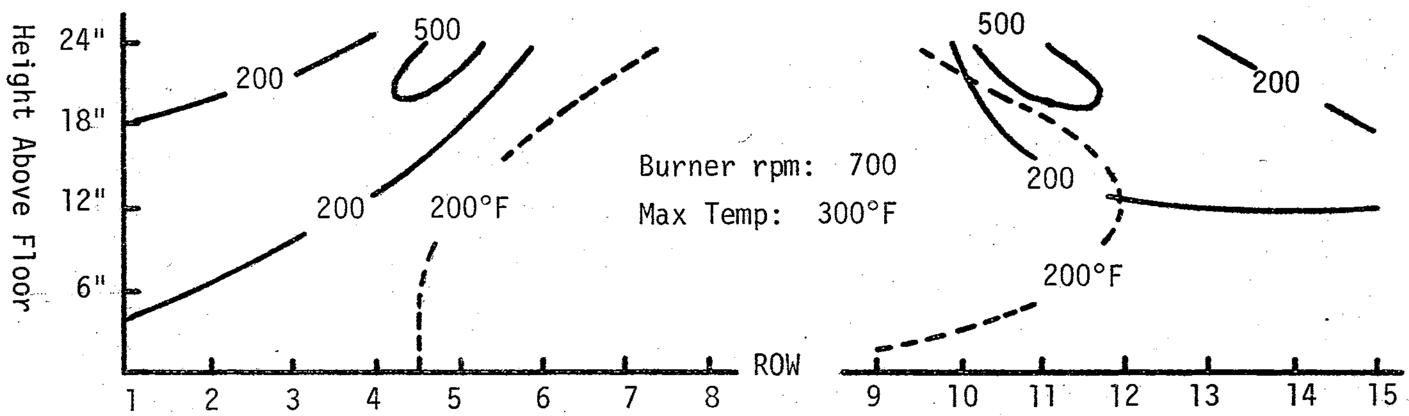

Figure 24. Velocity (FPM) and Temperature Profile for Air Curtain

There is a lengthening of the 200°F region of the chamber, and a lowering of the bottom 200 feet per minute velocity line. This improvement is attributable to the decreased volume of air being moved inside the heating chamber.

Plant Elevated Temperature Period And Maximum Temperature Tests

The final machine test consisted of placing a thermocouple in a row of cotton plants and driving the machine over the plants at various operating conditions. The temperature was recorded using a 0-800°F Leeds and Northrupp Speedomax H, Model S, temperature recorder. The data obtained from this test is presented in Table VII, along with the average values for the different operating conditions. Parameters varied were machine speed, fuel pressure, and height of thermocouple. The maximum temperature was recorded and read directly from the chart. The elevated temperature period was determined by measuring the distance on the chart that the thermocouple remained at a temperature greater than 200°F. Knowing the chart speed allowed calculation of the elevated temperature period.

Figures 25, 26, and 27 are graphical representations of the average maximum temperature and elevated temperature period for ground speed, fuel pressure, and height, respectively. In each figure, the shape of both curves is similar. This was to be expected because as the air temperature increases, a longer time would be required to both heat and cool the air surrounding the plant.

The ground speed of the prototype was similar to the model Reynolds number of the model study. For increasing Reynolds number, the wind

TABLE VII

ORIGINAL DATA FOR MAXIMUM TEMPERATURE AND ELEVATED TEMPERATURE PERIOD

900 rpm Burner Fan 1200 rpm Air Curtain Fan

TEST	SPEED (mph)	FUEL PRESSURE (psi)	HEIGHT (in.)	MAXIMUM TEMPERATURE (°F)	APPROXIMATE ELEVATED TEMPERATURE PERIOD (sec)*
1	2	15	4	415	1.41
2	3			355	1.17
3	4			290	.94
4	2	25		535	1.41
5	3			505	1.64
6	4			415	1.17
7	2	15	12	345	.94
8	3			300	.70
9	4			270	.47
10	2	25		450	1.41
11	3			420	1.17
12	4			370	.94
13	2	15	18	315	.94
14	3			290	.70
15	4			245	.47
16	2	25		435	1.17
17	3			400	.94
18	4			355	.94
19	2	15	24	320	.94
20	3			270	.70
21	4			255	.47
22	2	25		370	.94
23	3			350	.47
24	4			325	.70
25	-	15	hover	425	----
26	-	25	hover	575	----

* Based on chart speed of 30 in./hr. .47 sec./(1/64 in.)

AVERAGE VALUES OF ENVIRONMENT TESTS

GROUND SPEED	MAXIMUM TEMPERATURE	ELEVATED TEMP PERIOD
2 mph	398	1.14
3 mph	361	.94
4 mph	315	.76
PRESSURE		
15	306	.82
25	411	1.08

HEIGHT	MAXIMUM TEMPERATURE	ELEVATED TEMP PERIOD
4"	419	1.29
12"	359	.94
18"	340	.86
24"	315	.70

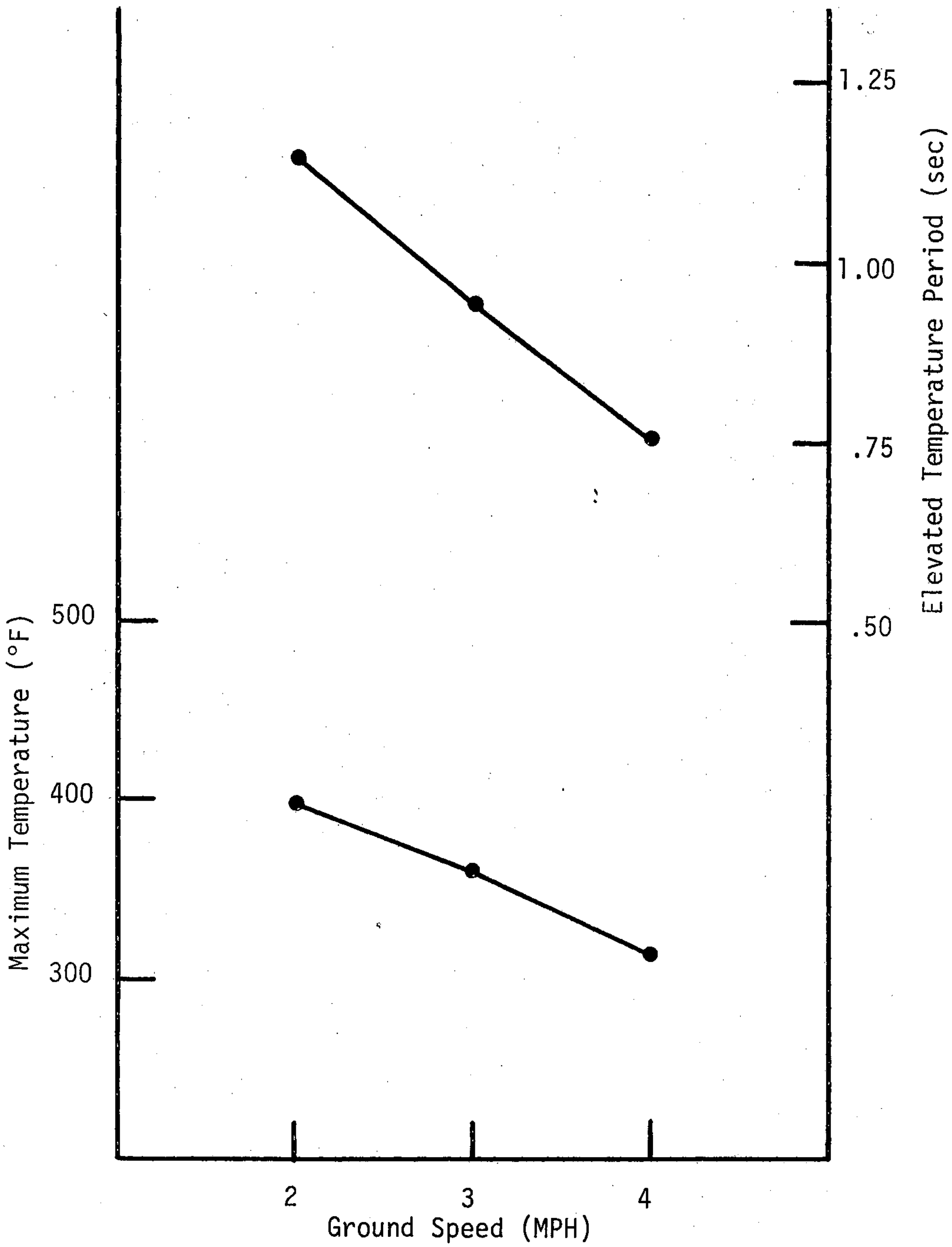

Figure 25. Maximum Temperature and Elevated Temperature Period vs. Ground Speed

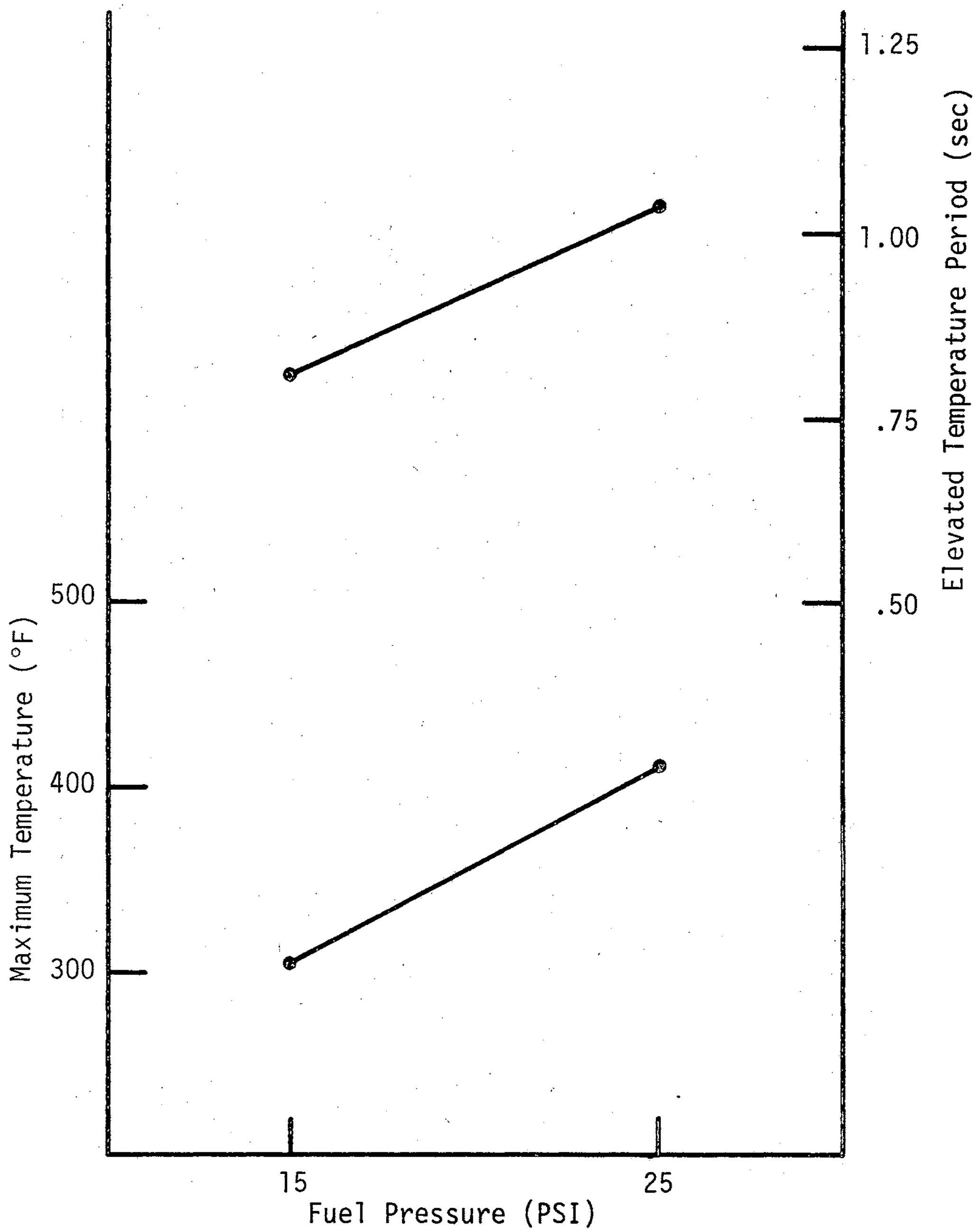

Figure 26. Maximum Temperature and Elevated Temperature Period vs. Fuel Pressure

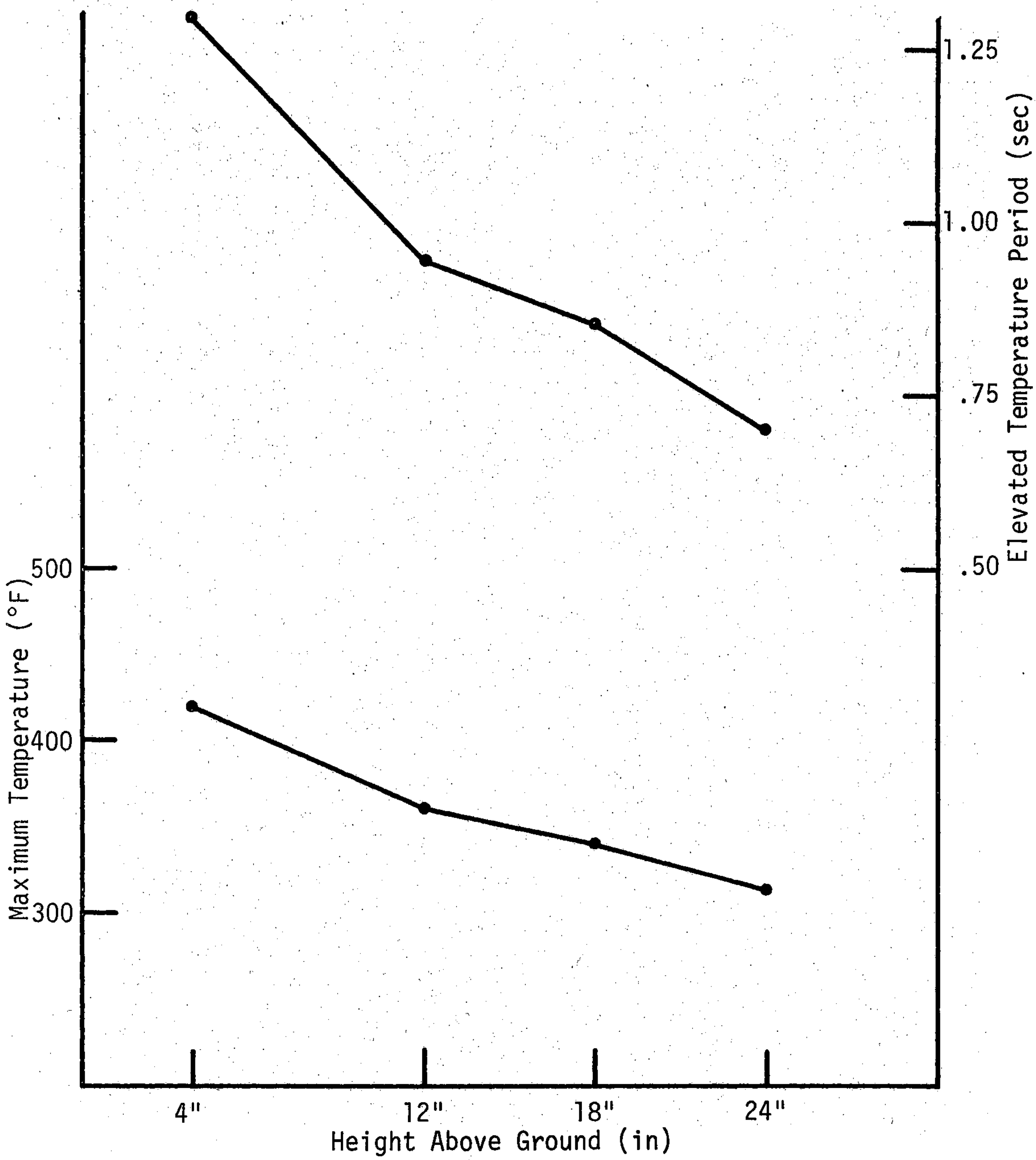

Figure 27. Maximum Temperature and Elevated Temperature Period vs. Height

tunnel air velocity was increasing. As the model Reynolds number increased, the heat transfer coefficient increased, indicating an increased amount of heat loss. This was reflected in a decrease in the maximum temperature and elevated temperature period of the prototype for increasing ground speeds.

Fuel pressure was a measure of the amount of heat supplied within the heating chanber. As long as there was sufficient oxygen to support combustion, an increasing fuel pressure would result in more heat supplied. This was reflected by an increase of both elevated temperature period and maximum temperature.

The variation of elevated temperature period and maximum temperature with variable height was more complicated. The flow patterns developed within the heating chamber were responsible for the variation shown in Figure 27. The heated air was blown downward over the plants and struck the ground. Because of the forward motion, the return intake at the rear of the heating chamber, and decreased density due to heating, the air moved to the rear and began to rise. By this time, the heated air was mixed with enough cooler air that the temperature decreased considerably. Also, heated air emerging from the ends of the machine would be blown downward because of the air curtain action. All of these factors contributed to increased elevated temperature period and maximum temperature at the lower levels and decreasing temperatures and elevated temperature period at the upper levels.

Using a computer program with a logarithmic transformation, a prediction equation was obtained which related elevated temperature period and maximum temperature to ground speed, fuel pressure, and height. The form of the equation was:

$$Y = C_o \times X_1^{A_1} \times X_2^{A_2} \times X_3^{A_3}.$$

Y was the dependent variable, elevated temperature period or maximum temperature; X_1 was ground speed in miles per hour; X_2 was fuel pressure in pounds per square inch; X_3 was height in inches of the thermocouple above the ground. The values obtained are given in Table VIII, with the correlation coefficient for each equation.

TABLE VIII

EQUATION CONSTANTS FOR MAXIMUM TEMPERATURE AND ELEVATED TEMPERATURE PERIOD

	MAXIMUM TEMPERATURE	ELEVATED TEMPERATURE PERIOD
$C_{(o)}$	124.4	3.041
$A_{(1)}$	-0.3125	-0.6875
$A_{(2)}$	+0.5835	+0.6499
$A_{(3)}$	-0.1455	-0.4199
Correlation Coefficient	.9791	.9495

CHAPTER VIII

SUMMARY AND CONCLUSIONS

This study involved the adaptation of the air curtain on the front and rear of the thermal defoliation machine to replace the doors used on previous machines. The first part of the study consisted of a model study performed using wind tunnel facilities at the Agricultural Engineering Research Laboratory at Oklahoma State University. The second part was a prototype test performed with the existing thermal defoliation machine, after modifications to include the air curtain principle. The second part was conducted at the Oklahoma Cotton Research Station at Chickasha, Oklahoma.

The model used in the model study was a one-eighth scale reproduction of the heating chamber on the prototype. No attempt was made to duplicate the internal flow or the high temperatures achieved in the prototype. The heating device used was a one thousand watt resistance electrical heater which was placed inside the model. Voltage and amperage were measured to determine heat input. The air curtains were developed using a series of small holes in a one-inch diameter thinwalled pipe. Air was supplied from the compressed air system in the building. Factors considered in the model study were direction of front and rear air curtains, air curtain velocity, wind velocity, and wind direction.

On the prototype machine, transverse flow fans were used to develop the air curtains. Three different types of tests were performed using the prototype: field tests to determine plant response, tests to evaluate the effectiveness of the air curtain, and tests to determine elevated temperature period and maximum temperature which the plant is subjected to.

Prediction equations were developed to predict the heat transfer coefficient from the model study and the elevated temperature period and maximum temperature for the prototype.

Conclusions

1. The model study indicated that direction of the air curtains was important to successful operation. The air curtain direction was measured outward from the vertical. The heat transfer coefficient decreased for increasing front air curtain angle, and increased for increasing rear air curtain angle.
2. For wind tunnel speeds of less than 1000 feet per minute, using the air curtains resulted in a considerable decrease in the heat transfer coefficient. For increasing wind tunnel velocities, the heat transfer coefficient of the model increased, and for increasing air curtain velocities, the heat transfer coefficient decreased.
3. Wind direction was a definite factor in operation of the model Much more uniform performance was obtained using the corner shields to protect the heating chamber from a side wind.
4. It is possible to predict the heat transfer coefficient for the model as a function of front air curtain angle, rear air curtain angle, air curtain velocity, and wind tunnel velocity. A correlation coefficient of .9420 was achieved.

5. Air velocity readings indicated that both the front and rear air curtains were well developed with no internal heating on the prototype. With internal heating, however, the air curtain flow is much less pronounced.
6. For the prototype, field tests indicated by plant response a statistically significant difference in performance in only one of four possible indications. Operation of the machine away from the wind and with air curtains in operation was different than operation into the wind and without air curtains in the less severe test for percent kill.
7. It is possible to predict the elevated temperature period and maximum temperature which the cotton leaf is subjected to as a function of fuel pressure, ground speed, and height of the leaf. A correlation coefficient of .9791 was achieved for maximum temperature, and .9495 for elevated temperature period.

Suggestions for Future Study

1. Due to the natural tendency of the heated air to rise, and the side protection afforded by the cotton plants, it may be possible to remove the sash chain completely and operate with the heating chamber raised above the ground.
2. Air velocity and temperature readings indicated a definite improvement in the air curtain patterns when the burner fan was slowed to 700 rpm, rather than 900 rpm. Further tests at reduced burner fan speeds might indicate improved performance.

3. The cotton used in the field test did not achieve full growth. This may have allowed the wind to be a more important factor in performance than if the plants had been full grown.
4. Modifications to the internal flow arrangement of the heating chamber might allow more effective operation of the air curtains. If the front fan were located ahead of the hot air outlet farther, the front curtain might be more effective.

BIBLIOGRAPHY

(1) Perry, B. D., "Thermal Applicator for Field Crops," Master's Thesis, Oklahoma State University Library, 1967.

(2) Henderson, S. M., and R. L. Perry. Agricultural Process Engineering. New York: John Wiley & Sons, 1955, 272-299.

(3) Stickler, F. C., F. N. Reece, L. E. Anderson, and G. H. Larson, "Production of Grain Sorghum Using Flame for Weed Control and Dessication," Proceedings - Second Annual Symposium, Use of Flame in Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1965, 7-10.

(4) Lien, R. M. and D. R. Sisson, "Non Selective Flaming for Drainage Ditch Maintenance," Proceedings - Fourth Annual Symposium on Thermal Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1967, 28-29.

(5) Wilson, H. P., and R. D. Ilnicki, "Weed and Crop Responses to Flame Applications in Cole Crops," Proceedings - Fourth Annual Symposium on Thermal Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1967, 25-28

(6) Lalor, W. F., and W. F. Buchele, "Progress in Development of a Selective Flame Weeder," Proceedings - Fourth Annual Symposium on Thermal Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1967, 45-51.

(7) Matthews, E. J. "Flame Cultivation in Mechanized Weed Control," Proceedings - Third Annual Symposium on Thermal Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1966, 14-15.

(8) Mendenhall, W. T., "Cultural and Mechanical Control of Pink Bollworms," Proceedings - Third Annual Symposium on Thermal Agriculture, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1966, 12-14.

(9) Kent, J. D., "Thermal Defoliation of Cotton," Master's Thesis, Oklahoma State University Library, 1966.

(10) Batchelder, D. G., and J. G. Porterfield, "Thermal Defoliation of Cotton," *Proceedings - Second Annual Symposium Use of Flame in Agriculture*, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1965, 18-21.

(11) Reifschneider, D. E., and R. C. Nunn, "Infrared Cotton Defoliation or Dessication," *Proceedings - Third Annual Symposium on Thermal Agriculture*, Sponsored by Natural Gas Processors Association and National LP-Gas Association, 1966, 6-10.

(12) Norton, Walter, "Where to Use a Curtain of Air," *Consulting Engineer*, Vol. 12, No. 3, March 1959, 108-113.

(13) Herndon, C. L., "The Closed Open Door," *Heating, Piping, and Air Conditioning*, Vol. 38, No. 5, May 1966, 105-108.

(14) Zehnder, N., "Wind Tunnel Tests Help Design Air Curtain Entrance," *Heating, Piping, and Air Conditioning*, Vol. 32, No. 12, December 1960, 119-122.

(15) Albertson, M. L., et. al., "Diffusion of Submerged Jets," *Transactions of the American Society of Civil Engineers*, Vol. 115, 1950, 639-697.

(16) Merz, K. E., "An Unusual Type of Blower," *Product Engineering*, Vol. 34, No. 7, April 1, 1964, 51-55.

(17) Whitney, C. E., "Transverse Flow Improves Air Movement, Part 1," *Design Engineering*, Vol. 12, No. 12, December, 1966, 64-66.

(18) Whitney, C. E., "Transverse Flow Improves Air Movement, Part 2," *Design Engineering*, Vol. 13, No. 1, January, 1967, 52-53.

(19) Hetsroni, G., "Heat Transfer Properties of an Air Curtain," *Transactions of the American Society of Agricultural Engineers*, Vol. 6, No. 4, 1963, 328-331 & 334.

APPENDIX A

ORIGINAL MODEL TEST DATA

TABLE A-I

AIR VELOCITY (fpm) vs. PRESSURE FOR MODEL AIR CURTAINS

23 HOLES 18" LENGTH .0225" DIAMETER

Holes 1-7

Pressure (psi)	Distance from Pipe 1"	2"	3"	5"
2	650	500	0	0
4	1100	850	250	100
6	1400	1050	550	250
8	1800	1250	750	500
10	2200	1450	900	600
12	2500	1650	1100	650
14	2800	1850	1250	700
16	3100	2000	1300	800
18	3300	2250	1400	875
20	3500	2450	1550	950

Holes 9-15

Pressure	Distance from Pipe 1"	2"	3"	5"
2	500	200	0	0
4	1000	750	350	100
6	1400	1000	600	200
8	1800	1200	800	400
10	2250	1450	900	500
12	2600	1650	1000	600
14	2900	1900	1100	650
16	3200	2200	1250	750
18	3450	2300	1350	800
20	3650	2600	1400	900

Holes 17-23

Pressure	Distance from Pipe 1"	2"	3"	5"
2	550	100	0	0
4	650	600	200	0
6	800	800	550	300
8	1100	950	650	400
10	1350	1100	800	650
12	1600	1300	900	700
14	1700	1500	950	800
16	1900	1600	1100	950
18	2100	1700	1250	1000
20	2250	1800	1300	1050

TABLE A-II

AIR VELOCITY (fpm) vs. PRESSURE FOR MODEL AIR CURTAINS

45 HOLES 18" LENGTH .0225" DIAMETER

Holes 3-15

Pressure (psi)	Distance from Pipe 1"	2"	3"	5"
2	600	200	0	0
4	1200	550	400	0
6	1600	700	600	100
8	2100	950	700	100
10	2450	1100	900	500
12	2750	1250	1000	600
14	3000	1500	1100	700
16	3250	1600	1250	700
18	3500	1800	1350	800
20	3700	2050	1500	900

Holes 18-29

Pressure	Distance from Pipe 1"	2"	3"	5"
2	600	100	0	0
4	1100	500	400	0
6	1500	750	600	200
8	1900	900	800	300
10	2200	1050	950	500
12	2600	1300	1025	550
14	2900	1450	1100	850
16	3150	1600	1300	850
18	3450	1800	1450	900
20	3700	2100	1600	975

Holes 33-45

Pressure	Distance from Pipe 1"	2"	3"	5"
2	650	100	100	0
4	1150	650	300	100
6	1600	850	550	250
8	2150	1000	700	525
10	2500	1250	850	600
12	2800	1450	1000	700
14	3050	1700	1150	800
16	3350	1850	1300	950
18	3550	2200	1400	1000
20	3850	2350	1600	1050

TABLE A-III

AIR VELOCITY (fpm) vs. PRESSURE FOR MODEL AIR CURTAINS

48 HOLES 12" LENGTH .0225" DIAMETER

Pressure	Velocity
3	1600
6	2400
9	3200
12	4000
15	4800

TABLE A-IV

MODEL STUDY RESULTS

TEST NUMBER	AMBIENT TEMP (°F)	FRONT TEMP (°F)	MEAN REAR TEMP (°F)	AIR CURTAIN VELOCITY (FPM)	WIND VELOCITY (FPM)	FRONT AIR CURTAIN ANGLE	REAR AIR CURTAIN ANGLE	LEAD ANGLE
3-1	96.4	97.6	102.4	4000	806	0°	45°	0°
3-2	96.8	102.8	106.6	4000	806	15°	45°	0°
3-3	96.6	103.6	111.0	4000	806	30°	45°	0°
3-4	96.8	104.7	116.5	4000	806	45°	45°	0°
3-5	95.1	102.5	117.2	4000	806	60°	45°	0°
3-6	94.6	108.3	117.4	4000	806	75°	45°	0°
3-7	90.2	94.6	108.6	0	585	---	---	0°
3-8	91.4	94.5	107.0	0	692	---	---	0°
3-9	92.5	94.6	105.4	0	806	---	---	0°
3-10	93.7	95.4	104.8	0	943	---	---	0°
3-11	94.8	95.7	103.4	0	1132	---	---	0°
3-12	95.2	95.9	102.3	0	1307	---	---	0°
3-13	95.8	96.1	100.8	0	1547	---	---	0°
3-14	94.1	104.9	117.6	4000	806	45°	0°	0°
3-15	86.4	93.9	103.8	4000	806	45°	15°	0°
3-16	88.1	94.0	105.2	4000	806	45°	30°	0°
3-17	89.6	104.0	106.5	4000	806	45°	45°	0°
3-18	90.9	98.5	107.4	4000	806	45°	60°	0°
3-19	92.1	98.9	109.0	4000	806	45°	75°	0°
3-20	93.1	99.9	109.5	4000	806	45°	90°	0°
3-21	94.3	99.8	109.5	4000	806	45°	105°	0°
3-22	97.7	99.7	110.1	1600	806	45°	45°	0°
3-23	97.9	100.8	111.0	2400	806	45°	45°	0°
3-24	97.9	104.3	115.6	3200	806	45°	45°	0°
3-25	85.1	89.5	102.0	4800	806	45°	45°	0°
3-26	95.8	100.7	116.3	4000	585	45°	45°	0°

TABLE A-IV (CONTINUED)

MODEL STUDY RESULTS

TEST NUMBER	AMBIENT TEMP (°F)	FRONT TEMP (°F)	MEAN REAR TEMP (°F)	AIR CURTAIN VELOCITY (FPM)	WIND VELOCITY (FPM)	FRONT AIR CURTAIN ANGLE	REAR AIR CURTAIN ANGLE	LEAD ANGLE
3-27	96.3	101.3	115.7	4000	692	45°	45°	0°
3-28	96.8	100.4	108.7	4000	952	45°	45°	0°
3-29	97.1	98.1	105.5	4000	1140	45°	45°	0°
3-30	97.3	97.9	105.0	4000	1307	45°	45°	0°
3-31	86.4	90.4	100.6	4000	806	45°	45°	22 ½°
3-32	87.4	90.1	103.4	4000	806	45°	45°	45°
3-33	88.4	113.5	111.1	4000	806	45°	45°	67 ½°
3-34	89.4	111.9	109.3	4000	806	45°	45°	90°
3-35*	91.7	96.0	110.3	4000	806	45°	45°	22 ½°
3-36*	92.8	97.9	110.7	4000	806	45°	45°	45°
3-37*	93.9	99.5	113.2	4000	806	45°	45°	67 ½°
3-38*	94.7	105.6	108.5	4000	806	45°	45°	90°

* Tests 3-35 through 3-38 were performed with corner shields in place.

APPENDIX B

ORIGINAL FIELD TEST DATA

TABLE B-I

FIRST TEST SERIES

OCTOBER 22, 1968

PRESSURE: 20 psi   AIR VELOCITY: 200 fpm   GROUND SPEED: 2 3/4 mph

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT* DEFOLIATION	PERCENT** KILL
I-1	1	26	13	0	50.0	100.0
	2	20	5	0	75.0	100.0
	3	55	50	48	9.1	12.7
	4	43	38	17	11.6	60.5
I-2	1	18	14	0	22.2	100.0
	2	40	15	0	62.5	100.0
	3	49	48	41	2.0	16.3
	4	31	21	10	32.3	67.7
I-3	1	17	15	9	11.8	47.1
	2	27	25	25	7.4	7.4
	3	15	6	0	60.0	100.0
	4	29	8	0	72.4	100.0
I-4	1	23	22	20	4.3	13.4
	2	34	34	32	0.0	5.8
	3	36	11	0	69.4	100.0
	4	30	10	0	66.7	100.0
II-1	1	30	27	0	10.0	100.0
	2	32	26	0	18.7	100.0
	3	38	31	10	18.4	73.6
	4	38	23	11	39.4	71.0
II-2	1	40	25	0	37.5	100.0
	2	14	11	0	21.3	100.0
	3	32	26	13	18.7	59.3
	4	22	15	0	31.8	100.0
II-3	1	70	53	53	24.2	24.2
	2	20	9	6	55.0	70.0
	3	50	39	0	22.0	100.0
	4	54	40	0	25.9	100.0
II-4	1	66	61	55	7.5	16.7
	2	64	52	35	18.7	45.3
	3	77	50	0	35.0	100.0
	4	35	26	0	25.7	100.0

TABLE B-I (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT* DEFOLIATION	PERCENT** KILL
III-1	1	17	5	0	70.5	100.0
	2	24	13	3	45.8	87.5
	3	36	36	30	0.0	16.7
	4	43	30	19	30.2	55.8
III-2	1	34	25	11	26.4	67.6
	2	25	18	2	30.4	92.0
	3	46	20	2	56.5	95.6
	4	45	22	4	51.1	91.1
III-3	1	21	21	17	0.0	19.0
	2	23	8	0	65.2	100.0
	3	31	6	0	80.6	100.0
	4	31	11	0	64.5	100.0
III-4	1	61	55	12	9.8	80.3
	2	25	16	4	36.0	84.0
	3	25	14	0	44.0	100.0
	4	43	11	0	74.4	100.0
IV-1	1	74	65	35	12.1	52.7
	2	49	35	2	28.5	95.9
	3	45	33	0	26.6	100.0
	4	54	23	0	57.4	100.0
IV-2	1	76	58	0	23.6	100.0
	2	50	38	17	24.0	66.0
	3	31	28	9	9.6	70.9
	4	19	15	11	21.0	42.0
IV-3	1	34	22	3	35.2	91.1
	2	30	22	6	26.7	80.0
	3	52	45	0	13.4	100.0
	4	26	21	0	19.2	100.0
IV-4	1	43	41	38	4.6	11.6
	2	26	22	19	15.3	26.9
	3	18	11	0	38.8	100.0
	4	40	27	0	32.5	100.0
V-1	1	30	26	0	13.3	100.0
	2	25	10	8	60.0	68.0
	3	57	52	47	8.7	17.5
	4	43	43	37	0.0	13.9

TABLE B-I (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT* DEFOLIATION	PERCENT** KILL
V-2	1	36	6	0	83.3	100.0
	2	44	20	4	54.5	90.9
	3	30	24	24	20.0	20.0
	4	36	34	32	5.5	11.0
V-3	1	38	6	0	84.2	100.0
	2	47	40	37	14.8	21.2
	3	40	18	0	55.0	100.0
	4	56	20	0	64.2	100.0
V-4	1	16	13	8	18.7	50.0
	2	24	24	24	0.0	0.0
	3	23	8	0	65.2	100.0
	4	24	20	0	16.6	100.0
VI-1	1	31	19	0	38.7	100.0
	2	21	19	0	9.5	100.0
	3	36	28	0	22.2	100.0
	4	21	17	0	19.0	100.0
VI-2	1	31	22	0	29.0	100.0
	2	19	18	7	5.2	63.1
	3	47	36	27	23.4	42.5
	4	42	30	26	28.5	38.0
VI-3	1	31	18	0	41.9	100.0
	2	26	26	24	0.0	7.6
	3	42	27	0	35.7	100.0
	4	35	26	0	25.7	100.0
VI-4	1	25	21	17	16.0	32.0
	2	13	6	1	53.8	92.3
	3	29	23	0	20.6	100.0
	4	26	17	0	34.6	100.0
VII-1	1	25	24	21	4.0	16.0
	2	40	36	27	10.0	32.5
	3	41	21	4	48.7	90.2
	4	28	8	7	71.4	75.0
VII-2	1	56	34	0	39.2	100.0
	2	12	6	0	50.0	100.0
	3	39	29	29	25.6	25.6
	4	28	19	0	32.1	100.0

TABLE B-I (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT* DEFOLIATION	PERCENT** KILL
VII-3	1	38***	34	32	10.5	15.7
	2	35***	9	0	74.2	100.0
	3	37***	27	0	27.0	100.0
	4	60***	50	0	16.7	100.0
VII-4	1	50***	46	29	8.0	42.0
	2	45***	38	31	15.6	31.1
	3	75***	62	6	17.3	92.0
	4	60***	45	15	25.0	75.0

* Percent Defoliation = $\frac{\text{Initial Count - Final Count}}{\text{Initial}}$

** Percent Kill = $\frac{\text{Initial - Green}}{\text{Initial}}$

*** Estimated

TABLE B-II

Second Test Series

OCTOBER 24, 1968

PRESSURE: 20 psi AIR VELOCITY: 400 fpm GROUND SPEED: 3 mph

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT DEFOLIATION	PERCENT KILL
I-1	1	42	39	23	7.1	45.2
	2	26	26	1	0.0	96.1
	3	35	32	31	8.5	11.4
	4	26	25	17	3.8	34.6
I-2	1	24	24	0	0.0	100.0
	2	25	21	0	16.0	100.0
	3	30	20	0	33.3	100.0
	4	48	46	20	4.1	58.3
I-3	1	37	37	31	0.0	16.2
	2	32	32	28	0.0	12.5
	3	34	34	6	0.0	82.3
	4	16	13	0	18.7	100.0
I-4	1	43	41	0	4.6	100.0
	2	44	39	0	11.3	100.0
	3	17	13	0	23.5	100.0
	4	18	16	14	11.1	22.2
II-1	1	62	62	7	0.0	88.7
	2	36	34	10	5.5	72.7
	3	28	28	23	0.0	17.8
	4	32	30	21	6.2	34.3
II-2	1	42	37	28	11.9	33.3
	2	35	22	11	37.1	68.5
	3	29	29	19	0.0	34.4
	4	19	16	10	15.7	47.3
II-3	1	47	43	6	8.5	87.2
	2	43	42	37	2.3	13.9
	3	23	20	0	13.0	100.0
	4	65	60	0	7.6	100.0
II-4	1	47	40	0	14.8	100.0
	2	36	36	0	0.0	100.0
	3	36	31	11	13.8	69.4
	4	35	29	12	17.1	65.7

TABLE B-II (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT DEFOLIATION	PERCENT KILL
III-1	1	42	36	0	14.2	100.0
	2	42	33	5	21.4	88.0
	3	19	19	12	0.0	36.8
	4	37	36	35	2.7	5.4
III-2	1	54	50	43	7.4	20.3
	2	42	42	26	0.0	38.0
	3	30	30	16	0.0	46.6
	4	27	27	18	0.0	33.3
III-3	1	24	22	9	8.3	62.5
	2	24	23	8	4.1	66.7
	3	43	34	4	25.5	90.6
	4	52	40	1	23.0	98.0
III-4	1	27	23	0	14.8	100.0
	2	36	32	2	11.1	94.4
	3	33	32	28	3.0	15.1
	4	42	42	13	0.0	69.0
IV-1	1	42	38	20	9.5	52.3
	2	52	49	5	5.7	90.3
	3	44	44	31	0.0	29.5
	4	41	41	34	0.0	17.0
IV-2	1	36	36	29	0.0	19.4
	2	24	24	21	0.0	12.5
	3	47	44	34	6.3	27.6
	4	26	26	21	0.0	19.2
IV-3	1	32	32	26	0.0	18.7
	2	25	23	13	8.0	48.0
	3	38	34	0	10.5	100.0
	4	16	15	0	6.3	100.0
IV-4	1	31	23	6	25.8	80.6
	2	36	36	0	0.0	100.0
	3	55	48	44	12.7	20.0
	4	26	16	0	38.4	100.0
V-1	1	30	30	21	0.0	30.0
	2	24	24	8	0.0	33.3
	3	37	30	24	18.9	35.1
	4	40	37	25	7.5	37.5

TABLE B-II (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT DEFOLIATION	PERCENT KILL
V-2	1	61	50	14	18.0	77.0
	2	28	28	7	0.0	75.0
	3	50	50	46	0.0	8.0
	4	18	18	15	0.0	16.7
V-3	1	34	30	3	11.7	91.1
	2	23	19	0	17.3	100.0
	3	20	19	15	5.0	25.0
	4	24	20	7	16.7	70.8
V-4	1	34	33	28	2.9	17.6
	2	38	38	34	0.0	10.5
	3	32	29	0	0.3	100.0
	4	35	29	0	17.1	100.0
VI-1	1	38	33	1	13.1	97.3
	2	26	22	0	15.3	100.0
	3	34	34	34	0.0	0.0
	4	30	28	19	6.7	36.7
VI-2	1	39	36	2	7.6	94.8
	2	33	31	20	6.4	39.3
	3	32	26	18	18.7	43.7
	4	36	36	28	0.0	22.2
VI-3	1	36	34	32	5.5	11.0
	2	17	14	0	17.6	100.0
	3	30	28	0	6.6	100.0
	4	54	51	0	5.5	100.0
VI-4	1	43	37	23	13.9	46.5
	2	25	23	3	8.0	88.0
	3	61	61	54	0.0	11.4
	4	26	26	24	0.0	7.6
VII-1	1	38	33	13	13.1	65.7
	2	26	26	22	0.0	15.3
	3	21	21	4	0.0	80.9
	4	22	22	10	0.0	54.5
VII-2	1	31	30	21	3.2	32.2
	2	33	31	12	6.0	63.6
	3	69	62	49	10.1	28.9
	4	28	27	15	3.5	46.4

TABLE B-II (CONTINUED)

TEST NUMBER	PLANT NUMBER	INITIAL COUNT	FINAL COUNT	GREEN LEAVES	PERCENT DEFOLIATION	PERCENT KILL
VII-3	1	50*	46	0	8.0	100.0
	2	38*	34	3	10.5	92.1
	3	48*	47	42	2.0	12.5
	4	85*	80	76	5.8	10.5
VII-4	1	40*	38	0	5.0	100.0
	2	80*	78	0	2.5	100.0
	3	41*	41	40	0.0	2.4
	4	36*	34	34	5.5	5.5

* Estimated

APPENDIX C

AIR TEMPERATURE AND VELOCITY DATA FOR PROTOTYPE AIR CURTAINS

TABLE C-I

TEST 1

TEMPERATURE (°F) FRONT FAN-0 RPM

BURNER FAN-700 RPM REAR FAN 1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	70	70	70	70	70	70	70	70	70	70	60	70
2	70	70	70	70	70	70	70	70	70	70	70	65
3	70	90	70	70	75	80	70	70	70	80	70	70
4	80	85	70	70	80	80	70	70	70	80	70	70
5	170	80	70	75	130	80	70	80	70	80	70	80
6	110	150	90	75	110	100	100	100	80	80	70	80
7	150	130	110	100	210	130	130	100	180	120	80	110
8	220	310	100	190	220	280	130	300	200	285	110	280
9	210	260	180	200	250	270	280	240	220	240	160	170
10	140	180	200	115	210	250	260	150	135	150	145	115
11	155	185	140	100	200	230	200	130	160	150	120	110
12	160	150	130	100	180	220	210	150	230	140	110	105
13	145	140	110	110	170	200	170	140	135	140	120	110
14	120	120	110	110	170	170	150	145	115	130	130	120
15	130	120	110	100	160	160	150	150	120	130	140	130

TABLE C-II

TEST I

AIR VELOCITY (FPM) FRONT FAN-0 RPM

BURNER FAN-700 RPM REAR FAN 1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	50	40	10	60	10	10	10	10	10	10	10	10
2	20	10	10	10	10	50	10	50	10	10	10	10
3	10	10	10	50	10	10	10	50	20	30	10	10
4	10	10	10	30	25	30	10	30	30	10	10	10
5	10	10	30	20	10	10	20	30	20	10	10	20
6	10	10	10	10	10	40	20	50	10	30	20	20
7	40	20	30	35	50	10	20	20	30	20	10	50
8	30	40	30	80	50	40	10	70	20	20	40	70
9	20	40	50	20	100	30	20	10	125	90	75	30
10	10	20	40	600	100	60	75	500	75	50	70	600
11	40	100	450	400	75	150	250	500	40	200	500	150
12	10	350	450	200	40	125	275	275	150	250	150	50
13	250	350	250	75	30	100	300	275	250	175	200	50
14	300	300	150	30	30	100	250	200	250	225	175	100
15	275	200	120	20	75	150	200	200	200	150	125	100

TABLE C-III

TEST II

TEMPERATURE (°F) FRONT FAN-1200 RPM

BURNER FAN-700 RPM REAR FAN-1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	100	100	100	100	130	120	110	100	110	100	100	100
2	100	100	100	95	130	115	120	110	110	100	100	100
3	110	105	100	90	125	140	120	110	100	105	110	110
4	100	125	110	95	135	160	125	110	100	100	110	105
5	200	150	120	95	230	220	150	110	140	140	130	110
6	240	180	160	130	280	260	200	150	130	150	130	120
7	240	190	170	150	280	245	220	190	220	160	170	160
8	290	330	310	280	310	320	320	280	250	300	310	270
9	95	235	170	190	250	270	260	250	150	240	160	180
10	120	165	195	120	210	250	250	160	160	170	170	120
11	160	170	130	100	170	220	200	130	165	150	115	105
12	150	150	115	95	150	200	165	115	145	125	120	110
13	125	120	90	95	150	185	140	135	145	130	120	130
14	120	120	105	100	155	170	140	130	130	140	135	130
15	120	110	100	90	140	150	140	135	140	145	140	120

TABLE C-IV

TEST II

AIR VELOCITY (FPM) FRONT FAN-1200 RPM

BURNER FAN-700 RPM REAR FAN-1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	175	150	75	10	200	275	200	100	75	175	200	100
2	150	200	100	20	100	300	200	150	75	225	250	175
3	150	250	200	30	50	300	250	175	50	150	250	200
4	70	225	200	125	30	175	400	225	20	50	125	450
5	30	30	200	350	30	20	200	500	40	50	40	330
6	40	75	40	75	40	30	50	100	50	75	150	75
7	150	30	50	40	100	30	30	50	75	50	50	60
8	60	60	60	75	100	50	60	75	60	50	50	75
9	10	40	70	10	150	50	30	20	150	200	100	40
10	30	30	30	650	100	100	150	500	75	50	50	900
11	40	150	575	400	50	100	300	500	50	375	500	150
12	200	500	450	50	50	75	400	250	200	400	200	140
13	400	400	40	50	30	150	325	200	250	250	175	140
14	400	250	75	20	50	150	200	150	275	250	200	100
15	275	150	75	20	100	200	200	175	200	200	150	75

TABLE C-V

TEST III

AIR VELOCITY (FPM) FRONT FAN-1200 RPM

BURNER FAN-700 RPM REAR FAN-0 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	350	400	200	100	450	550	350	250	300	550	350	200
2	400	500	350	150	350	500	400	300	350	550	450	250
3	450	500	350	300	400	550	450	275	300	450	450	400
4	200	500	450	100	200	400	600	250	200	500	550	450
5	200	250	700	400	200	150	700	800	150	100	150	950
6	400	175	200	200	350	150	200	300	250	200	200	120
7	800	200	150	150	500	125	100	100	650	150	175	200
8	150	180	200	250	200	200	200	150	200	200	200	225
9	150	300	200	150	200	150	100	100	200	500	200	200
10	100	60	150	125	150	200	90	150	200	250	75	50
11	10	50	100	100	75	150	120	150	100	150	30	75
12	10	10	75	50	75	100	75	75	150	100	50	75
13	10	20	30	40	50	50	50	75	10	100	30	100
14	10	50	30	20	50	10	10	50	100	75	30	50
15	10	10	20	20	40	175	50	30	50	20	10	40

TABLE C-VI

TEST III

TEMPERATURE (°F) FRONT FAN-1200 RPM

BURNER FAN-700 RPM REAR FAN-0 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	160	140	130	110	170	150	130	100	150	120	120	100
2	180	140	110	100	200	160	120	110	160	120	100	100
3	170	140	120	100	210	160	120	115	165	140	100	100
4	230	160	120	100	260	180	120	110	200	110	100	100
5	260	160	120	90	300	210	140	110	250	150	110	110
6	280	260	110	140	310	290	250	180	220	200	130	140
7	285	280	220	210	320	280	260	250	280	200	200	180
8	330	380	370	310	330	330	330	300	290	310	310	250
9	95	120	180	230	220	260	240	150	95	80	100	110
10	100	110	200	190	140	220	230	220	90	130	90	90
11	85	110	190	150	120	150	180	220	100	95	100	120
12	85	90	120	120	85	140	140	120	85	90	85	150
13	85	80	90	100	80	85	120	160	80	85	90	100
14	85	80	90	80	80	80	90	100	80	80	90	90
15	80	60	90	80	80	80	85	85	80	80	90	85

TABLE C-VII

TEST IV

VELOCITY (FPM) FRONT FAN-0 RPM

BURNER FAN-0 RPM REAR FAN-1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	400	300	75	50	350	500	500	300	150	300	350	250
2	500	400	350	30	200	500	500	300	150	350	550	450
3	400	550	400	200	100	450	650	500	130	300	550	650
4	200	750	800	250	100	250	600	700	150	100	250	750
5	75	150	250	1050	75	50	100	1200	100	100	150	400
6	100	20	10	150	100	30	20	75	150	250	150	150
7	10	10	10	50	20	40	50	30	75	200	75	150
8	10	10	20	30	30	30	20	50	50	10	20	150
9	10	30	30	10	50	40	50	10	10	50	50	75
10	20	10	30	1500	75	50	75	1300	50	20	30	1400
11	75	250	100	75	175	150	750	1300	100	100	750	600
12	250	1300	750	300	150	700	900	400	150	750	300	250
13	700	950	350	75	400	950	550	250	300	550	300	300
14	650	550	250	30	600	650	450	250	350	400	300	250
15	600	350	50	30	700	550	350	150	350	400	300	200

TABLE C-VIII

TEST V

AIR VELOCITY (FPM)   FRONT FAN-1200 RPM

BURNER FAN-900 RPM   REAR FAN-1200 RPM   NO INTERNAL HEATING

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	300	250	100	40	400	450	300	250	275	400	400	200
2	350	300	100	30	400	450	350	200	250	550	500	300
3	350	350	100	40	300	450	350	250	150	500	600	500
4	300	400	200	75	150	450	600	250	100	350	600	650
5	150	450	650	250	150	200	600	1000	150	150	200	1100
6	250	250	300	250	250	150	300	250	250	200	300	250
7	450	175	200	175	500	125	120	150	450	150	200	200
8	175	250	275	300	225	250	275	300	225	250	275	300
9	350	450	250	150	400	175	150	100	350	250	250	200
10	275	150	200	1900	350	250	300	700	325	150	200	1900
11	200	950	1400	550	225	350	650	1450	200	650	850	400
12	700	900	550	70	200	450	950	700	450	650	300	200
13	750	550	150	50	200	500	700	500	400	600	450	300
14	600	300	75	50	200	500	550	450	400	450	500	250
15	550	350	200	50	400	450	450	350	400	450	300	250

TABLE C-IX

TEST VI

TEMPERATURE (°F)   FRONT FAN-1200 RPM

BURNER FAN-900 RPM   REAR FAN-1200 RPM

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	95	95	85	90	110	100	95	90	110	110	95	100
2	95	90	90	90	115	100	95	90	110	110	100	100
3	100	90	95	85	120	110	100	90	110	110	110	95
4	120	100	90	90	150	120	110	100	130	140	110	100
5	160	110	100	90	220	180	140	95	150	180	160	110
6	190	150	140	150	240	200	190	200	200	200	150	140
7	180	160	210	200	240	210	250	260	180	180	200	230
8	220	210	190	210	250	240	220	210	230	240	240	220
9	130	120	150	160	200	220	230	220	140	140	150	170
10	140	150	150	100	160	200	220	120	140	150	140	90
11	120	110	110	95	140	180	190	120	130	120	110	100
12	120	110	110	100	140	160	160	120	120	130	110	100
13	110	95	100	100	140	140	150	130	120	120	120	110
14	110	100	110	100	130	140	130	130	130	120	120	110
15	100	100	100	100	130	130	140	130	120	120	120	110

TABLE C-X

TEST VI

VELOCITY (FPM) FRONT FAN-1200 RPM

BURNER FAN-900 RPM REAR FAN-1200 RPM WITH HEATING

ROW	A-SECTION				B-SECTION				C-SECTION			
	4"	12"	18"	24"	4"	12"	18"	24"	4"	12"	18"	24"
1	200	150	50	50	175	250	200	50	75	150	150	150
2	150	150	50	10	150	300	200	75	75	150	200	200
3	150	150	75	20	75	250	300	100	30	75	200	250
4	50	175	200	50	20	150	300	250	30	20	150	350
5	30	30	75	450	30	40	50	550	40	50	50	200
6	75	75	100	50	100	20	30	30	50	30	75	50
7	150	75	75	110	150	50	90	140	75	30	75	100
8	75	75	75	100	175	75	75	75	60	70	75	100
9	50	30	30	20	150	75	40	20	120	75	40	30
10	75	30	200	700	100	100	150	550	50	50	50	650
11	75	400	400	150	50	75	250	400	50	250	200	30
12	300	350	175	50	40	100	300	200	200	200	100	50
13	350	100	100	40	75	150	200	200	200	150	100	50
14	300	150	75	50	150	175	150	200	200	250	150	75
15	325	100	60	50	200	200	175	150	150	175	100	75

VITA

Glen Felix Moore

Candidate for the Degree of

Master of Science

Thesis: USING AIR CURTAINS TO SIMPLIFY THE THERMAL DEFOLIATION MACHINE

Major Field: Agricultural Engineering

Biographical:

Personal Data: Born at Enid, Oklahoma, September 10, 1945, the son of Mr. and Mrs. C. A. Moore, jr.

Education: Graduated from Nash High School, Nash, Oklahoma, in 1963; attended Central State College, Edmond, Oklahoma, in 1963 and 1964; received the Bachelor of Science degree from Oklahoma State University in 1968; completed requirements for Master of Science in May, 1969.

Professional Experience: Graduate Research Assistant for one year for the Agricultural Engineering Department, Oklahoma State University.