EXPERIMENTAL AND THEORETICAL STUDY OF FROST PROTECTION BY WATER APPLICATION UNDER SIMULATED RADIATION FROST CONDITIONS

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E H Kidder
Major professor

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EXPERIMENTAL AND THEORETICAL STUDY OF FROST PROTECTION BY WATER APPLICATION UNDER SIMULATED RADIATION FROST CONDITIONS

by

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A THESIS

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INTRODUCTION

A leaf during a typical radiation frost is being cooled by negative radiation to the sky as well as by the cooler air surrounding it. If this leaf is sprayed intermittently with water, 144 BTU per hour for each pound of water frozen is released in the vicinity of the leaf. This leaf therefore will remain at approximately the freezing point of water as long as there is sufficient water freezing to offset the various heat demands of the leaf and ice coat. This principle has been used to advantage in Michigan as well as in other areas to protect crops against damaging agricultural frosts.

In recent years water application has become a popular method of frost control in areas where spring and autumn frosts can and have destroyed the entire crop in localized areas. Most of the interest in this method is shown by growers of high value crops who are using sprinkler irrigation as a regular practice. These individuals expand their irrigation system to completely cover an area for frost control work.

In Michigan most of the frost protection by water application is carried out on crops which are not overly susceptible to damage by the ice load on the plant. Strawberries are a crop which lends itself to frost control in this manner. Also considerable interest in this method of frost control has been shown by growers of tree fruits.

Although water application for frost control has been used for a long time, only recently has there been much experimental work done on this particular method. The difficulties in studying the problem are many. The major problem is that of being unable to control the conditions under which our damaging agricultural frosts occur. During one crop season, the number of nights when one can study frost control are limited and the temperature of the air around the plant is not controllable. These difficulties are what have stimulated this thesis work on the study of frost control in the laboratory.

The objectives of this investigation are as follows:

1) Design and develop a refrigerated cabinet which will provide an environment for a plant which approaches the plant's environment in the field during a typical radiation frost.

2) Obtain data on the plant temperatures and plant damage for various combinations of water application rates, repeat frequencies and minimum air temperatures. These data will be obtained under the conditions of negative radiation and very low wind speed.

3) Develop theoretical and empirical expressions for the heat balance of a flat plate, sphere and cylinder of various sizes under different conditions of wind speed and direction.

important factor. They note that small one-nozzle sprinklers turning once every 12 - 20 seconds have given satisfactory results but that a larger sprinkler with one nozzle plugged, turning once every 90 seconds failed to give adequate protection to a tomato crop in a 24° F frost.

Keszler and Kaemfert (6) protected plants against a frost of 26.6° F. They suggested precipitation rates of .04 to .08 inches per hour.

Rogers (13) states the most important factor in frost damage is the temperature of the plant tissue itself, and certain plant tissues such as blossoms during a typical radiation frost can be 1 - 3° F colder than the air temperature. He indicates frost damage occurs when ice forms within the plant tissues and is due either to mechanical damage or physio-chemical changes. He noted that supercooling does exist but if the cell content does not freeze, frost damage is not apparent.

The condition of the plant with respect to previous exposure to frost and stage of growth was an important factor in frost resistance, Rogers indicated. He also found it not necessary to continue sprinkling once the air temperature has risen above 32° F.

Rogers reports good protection of apple blossoms and buds at a screen temperature of 27° F (exposed thermometer temp 25½°F) with a precipitation rate of one-tenth inch per hour and sprinkler rotation time of two minutes. He stressed the importance of continuous application.

Rogers (14) also carried out laboratory experiments, using slices of potato tuber. The laboratory experiments were with continuous water spray and controlled dropping of water on the potato disk.

The only reported experimental work on frost control by water application in this country is that of Bilanski (1) in 1955. His work consisted partly of a survey of about thirty farmers in Michigan who used their irrigation system to protect crops from strawberries and tomatoes to certain kinds of flowers. The farmers protected from 1 to 50 acres of crops using triangular spacing of sprinklers. Many indicated the irrigation equipment paid for itself in one year when also used for frost protection.

The second phase of Bilanski's work consisted of field studies of frost control. He noticed the air temperature in the sprinkled area to be 3 to 5° F higher than in the unsprinkled area.

Businger (3) in Holland carried out field studies on the effect of wind speed and precipitation rate on frost control. He relates some of the variables mathematically. Indicating wind effects are beneficial in that they usually bring warm air, Businger also said wind is harmful in that it distorts the spray pattern and causes uneven distribution.

A theoretical approach to the heat balance of the plant is made by Niemann (10), from the Meteorological Institute in

Hannover, Germany. He discusses the various phases of heat transfer, radiation, convection and evaporation, and how they affect the heat balance of the plant. The author discusses how various shapes of ice and plant forms affect the heat transfer but indicates that further work needs to be done along this line.

Fairly recent laboratory as well as field work has been carried on by von Pogrell (16). His laboratory apparatus consisted of a small spray nozzle (.02 inch diameter) and magnetic valve actuated by a time clock. The clock was adjustable to give a good range of application rates and interruption periods. Potted plants were then sprayed in a cold chamber where the air temperature could be controlled. The air and plant temperatures were sensed with thermocouples and recorded on single point recording galvanometers.

Von Pogrell's field and laboratory results showed a trend of increasing protection with an increase in application rate holding the interval constant and an increase in protection with a decrease in sprinkling interval holding the application rate constant.

A summary of von Pogrell's field and laboratory work for low wind speeds is in the following table:

TABLE I

MINIMUM AIR TEMPERATURE FOR FROST PROTECTION AT LOW WIND SPEEDS

Precipitation rate-in/hr.	Intervals in Minutes			
	1	2	3	4
.06	23.9	24.8	26.6	27.5
.12	21.2	22.1	23.0	23.9
.18	18.5*	20.3*	22.1*	23.0*
.24	15.3*	16.7*	17.8*	21.2*

Von Pogrell also investigated the effects of supercooling of the plant tissue. He noticed that there appeared to be more damage at the outer periphery of the rotating sprinkler pattern. This he attributed partly to the effect of the larger drops disturbing the supercooled state and thereby freezing the plant tissue. He suggests that a plant is able to supercool to a greater degree if ice is not formed on the surface. This is perhaps due to intracellular innoculation when ice is present.

The effect of wind speed on the amount of precipitation required to protect plants was investigated by Witte and von Pogrell (17). Their findings are as follows:

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*Frost protection to at least this level of temperature.

TABLE II

MINIMUM AIR TEMPERATURES FOR FROST PROTECTION AT VARIOUS WIND SPEEDS

Air Temperature °F	Wind Speed (m.p.h.)	Precipitation Required (in/hr)
25.7-27.5	1.1 3.1-5.6	0.04-0.06 0.06-0.10
22.9-23.5	1.1 3.1-5.6	0.10-0.14 0.14-0.18
16.2-19.0	1.1 3.1-5.6	0.14-0.18 0.26

APPARATUS

The problem of studying this method of frost control can be broken down into two different phases. The first is that of studying the plant's environment on a large scale. This entails a study of what factors in the local environment affect the air temperature and wind movement. How does the effect of heat transfer from the soil effect the temperature of the air? How does the freezing water from the sprinkler affect the air temperature around the plant? Such things as air drainage and topographic conditions are important in this type of study. This particular phase of the problem of frost control does not lend itself to study in the laboratory.

The second phase of the problem is concerned more with the plant and the environment immediately surrounding it as well as any radiation source or sink which might be affecting the heat balance of the plant and ice coat. This phase of the problem lends itself to study in the laboratory on a small scale. Essentially, the problem is reduced to applying the notions of a heat balance to the item whose temperature is to be controlled, namely the plant, leaf, blossom or bud. The following sketch and description will illustrate more clearly what is intended.

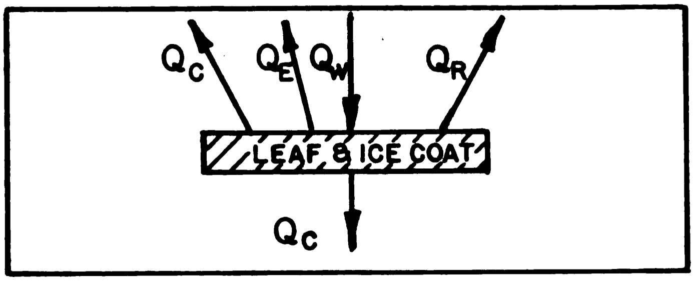

Fig. 1 Heat balance of leaf and ice coat.

where:

Qr = net radiant heat transfer

Qc = convective heat loss

Qe = evaporation loss

Qw = heat added to leaf by the sensible and latent heat release of the water which is applied (and frozen)

Once equilibrium conditions have been established, the condition which must exist is:

$$Qw = Qc + Qr + Qe$$

In view of the foregoing discussion, a refrigerated cabinet was constructed which would provide a constant rate of radiant heat transfer from the plant and ice coat. This apparatus also supplied cold air parallel to a leaf at low air velocities. A water application system capable of applying water at varying intensities and frequencies was assembled.

The remainder of the apparatus discussion will be divided into the major items of control necessary for this study.

General Construction

In attempting to decide what manner of refrigeration would be practical to provide a radiation sink in the regrigerated cabinet, dry ice was chosen because of the low equipment cost involved compared to vapor compression refrigeration.

The overall dimensions of the refrigerated cabinet are shown in Figure 2. The cooling element consists of a shallow enclosed pan. The pan, made of galvanized steel, is constructed gas tight with a vent to carry away the gaseous carbon dioxide. It is equipped with a steel cover plate as an access cover. With the cover removed solid carbon dioxide (dry ice) is placed in the pan with methyl alcohol. The alcohol tends to provide a uniform temperature heat transfer surface as well as preventing the formation of an insulating gas film between the dry ice and the bottom of the pan. This pan forms the ceiling of the cabinet. The pan then serves the dual purpose of cooling the air and providing the radiation sink. This ceiling is insulated from the galvanized steel inner sidewalls by one inch of styrofoam to minimize conduction down the walls.

A slow speed scraper was provided to maintain a constant frost thickness on the ceiling. The scraper was driven by a motor and gear reducer external to the cabinet. The frost was collected by the scraper and dumped at the sides of the

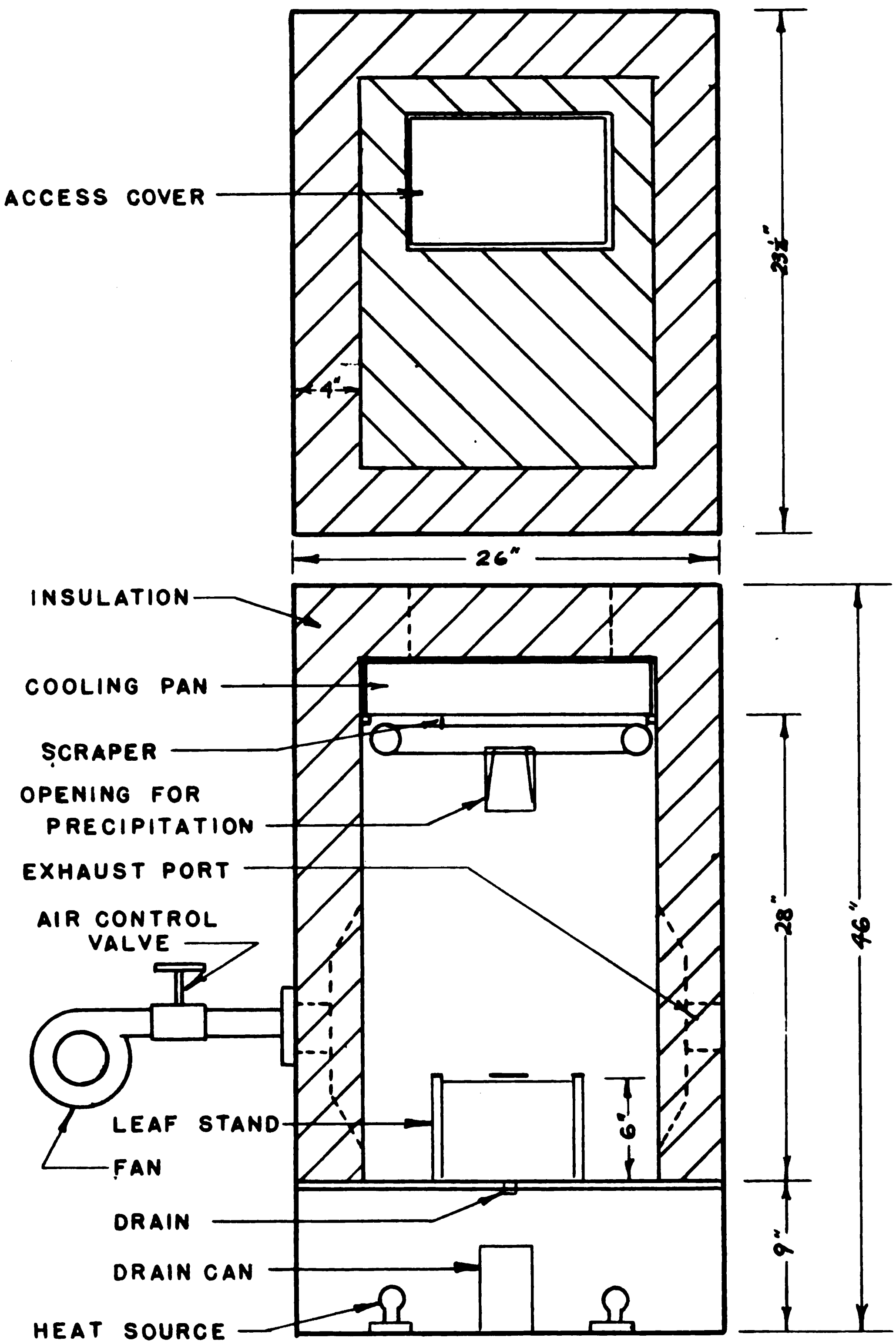

FIG. 2 PLAN AND ELEVATION VIEW OF FREEZING CABINET.

enclosure.

The cabinet is insulated with four inches of styrofoam on the three walls, door, and top above the cooling pan. The bottom of the cabinet is uninsulated and is provided with a heat source to prevent it from cooling below 32° F.

Air Temperature and Velocity Control:

The air temperature was regulated by bringing into the cabinet varying amounts of air from the walk-in refrigerator within which the cabinet was operated. The air was blown in by the fan shown in Figure 2, and mixed with the cooler air in the chamber. The flow rate was controlled by the hand valve. The refrigerated ceiling cooled the air inside by a combination of forced and natural convection. Bringing in lesser amounts of 40°F air made the resultant temperature of the air at the leaf location less.

The air velocity was predominatly parallel to the leaf and was measured by a hot wire anemometer. An auxiliary air recirculating system was installed to maintain the wind speed at about one and one fourth miles per hour for a limited series of tests.

The relative humidity in the cabinet was not controlled. It was measured at various air temperatures with an aspirating psycrometer.

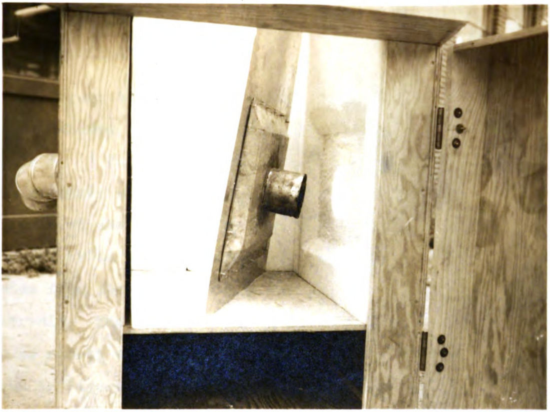

Fig. 3. Freezing cabinet during construction showing side-wall insulation and exhaust headder.

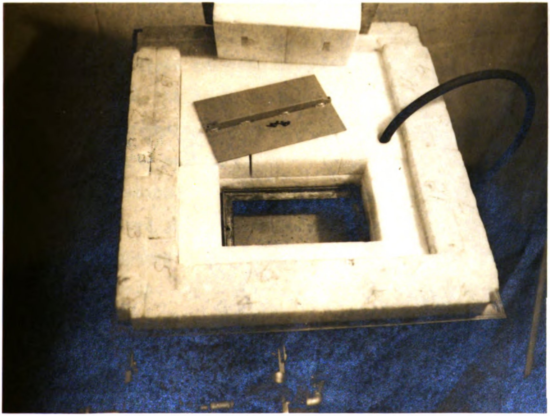

Fig. 4. Top view of freezing cabinet showing cooling pan access and gas vent hose.

Water Application:

In order to prevent freezing of the nozzle and piping, it was decided to place them outside the box as shown in Figure 5. A full cone nozzle with an orifice opening of .027 inch diameter was used to spray the water through a hole in the wall of the cabinet onto the plant. The solenoid was actuated by an industrial time clock capable of yielding cycles from twenty seconds to two minutes in twenty second intervals. The dwell or "on period" could be adjusted from two to ninety-eight per cent of the cycle. The waste line shown in Figure 5 between the pressure gage and the solenoid was needed to rapidly remove the pressure from the discharge side of the solenoid so it would snap shut soon after the valve was turned off.

A conical shaped galvanized section was installed in the wall at the point the spray entered the chamber. This section was wrapped with thermotape to prevent ice build-up from clogging the spray opening.

For calibration of the water application rate, a six ounce concentrated juice can was used because the area of the can opening compared very well with the size of the leaf used in the experiments. When precipitation tests were run at cold temperatures, nicrome wire was wrapped around the upper edge of the can to prevent water from freezing on the rim.

It was found necessary to provide a positive pressure

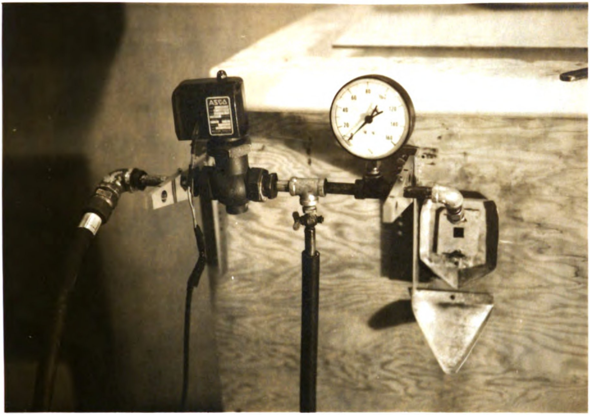

Fig. 5. Water application system showing solenoid, waste line, pressure gage and nozzle.

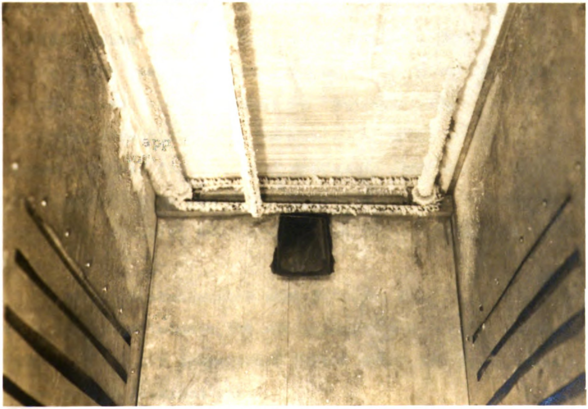

Fig. 6. Ceiling of cabinet showing frost scraper and conical section through sidewall for spray water entry.

control valve to eliminate the effect of pressure variations in the water line.

Several extra lengths of pipe were added to the water line inside the walk-in refrigerator. This acted as a heat exchanger to cool the spray water.

Temperature Measurement:

Two No. 40 s.w.g. copper-constantan thermocouples were used to measure leaf temperatures. No. 24 s.w.g. thermocouple wire was used as the lead wire for the fine wire thermocouples. No. 24 s.w.g. copper-constantan thermocouples were used as the air measuring thermocouples as shown in Figure 7.

The temperature was recorded outside the walk-in refrigerator on a 16 point print-type potentiometer. A common constantan system was used for all the thermocouples except one of the plant measuring thermocouples. Since the two plant thermocouples were not insulated from each other, one was connected directly to the recorder. A common constantan system was used for the other measuring junctions as it allowed the recorder to be easily disconnected from the system for the purpose of checking the temperatures with a manual potentiometer.

The air thermocouples were shielded with an aluminum and wood roof to eliminate the radiation effect as well as any effect on the air temperature in the event the spray water froze on the aluminum cap.

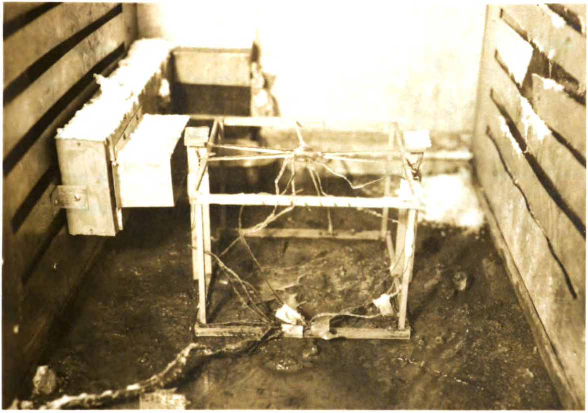

Fig. 7. Leaf stand and leaf after test showing air temperature thermocouples to the left, right and under the leaf.

Fig. 8. From left - manual potentiometer, net thermal radiometer, 16 point print potentiometer.

A wooden frame with a thread net held the leaf in place during the tests (Figure 7).

Radiation Measurement:

Both the outdoor and the refrigerated cabinet radiation measurements were made with a net radiometer.

PROCEDURE

Plant and Air Temperature Measurement

It was thought that the observation of plant damage alone was not a sufficient indicator of when there was insufficient precipitation for frost protection. Since the most important factor affecting frost damage is the temperature of the plant itself, it was decided to use fine wire thermocouples.

There were two methods of locating the thermocouple junctions. The first method was taping the thermocouple with cellophane tape in the center of the underside of the leaf. The other method was to tape the thermocouples to the center of the upper side of the leaf. This would place the thermocouple at the leaf-ice interface.

A second thermocouple was provided to measure leaf edge temperatures. This thermocouple was always placed on the upper side edge facing the direction from which the air was blowing.

As shown in the Apparatus, the air temperature was measured in three locations. If any difference between thermocouples was recorded, an average was used.

The leaf thermocouple wires were originally uninsulated. To prevent short circuiting before the measuring junction, these lead wires were insulated with spar varnish.

All thermocouples were calibrated against a certified thermometer calibrated to 0.2° F. It was not found necessary to draw calibration curves for any of the thermocouples.

Calibration of Precipitation

It was found most desirable to determine the water application rate immediately after a test run. However, when it was desired to accurately know the precipitation during the test, an additional precipitation test was run first. To do this the can was placed in exactly the same location as the leaf had been placed during the test. This was accomplished by placing the can and leaf directly over the drainage hole on the floor of the apparatus. This insured the same location in every test. The can was placed on a block to bring the rim to the same height as the leaf was placed during the test.

Precipitation tests were run for at least an hour. For very low precipitation rates, periods greater than one hour were needed to give sufficient water for accurate measurement.

At the end of the precipitation test, the contents of the can were poured in a 25 ml. graduated cylinder for measurment. The amount of water collected was always converted to an hourly basis.

The following calculations show the derivation of the conversion factor from ml. per hour to inches per hour:

Diameter of opening of can = D = 2.05 in.

1 milliliter of water = 1 cubic centimeter of water

1 cubic inch = $(2.54)^3$ cubic centimeters = 16.4 c.c.

= 16.4 ml. of water

let w = amount of water collected (ml/hr)

$$\text{Precipitation (in/hr)} = \frac{w \text{ (ml/hr)}}{16.4 \text{ (ml/in}^3) \times \text{area of can (in.}^2)}$$

$$= \frac{w}{16.4 \cdot \frac{(3.142)(2.05)^2}{4}}$$

$$= 0.0185w$$

The proper location of the nozzle with respect to the orifice through which the water entered the cabinet was determined by trial and error. The same nozzle was used throughout the tests.

The nozzle pressure was controlled by the setting of the pressure regulator as well as by the setting of the stopcock on the waste line.

Critical Plant Temperature

The procedure followed for determining the damaging plant temperature was to cool a series of leaves to various temperature levels. The temperature of the leaf would be maintained for five minutes, after which the leaf would be removed from the freezing chamber and the damage noticed. Leaves with and without an ice coat were tested in this manner.

Negative Radiation Measurements

The surface material was placed in a shallow pan and the pan was placed under the thermal net radiometer. The radiometer was located in an open area to eliminate radiation effects from buildings. The output of the radiometer was continuously recorded on a potentiometer. The temperature of the radiating surface was measured with a mercury in glass thermometer.

The net radiation from the soil to the ceiling was checked at the beginning of each series of tests in the same manner as described before for the outdoor radiation measurements. The standard radiation surface was the wet soil.

Wind Speed Measurement

A calibration of the average wind speed versus the air temperature was made. The anemometer was placed at the same location as the leaf during the actual tests.

Test Procedure

One object of this study is to develop a workable procedure for determining the damaging air temperature for various conditions. Three procedures were tried.

In all three procedures, a fresh,greenhouse grown, bean leaf was first clipped off the plant and its stem dipped in melted paraffin to seal it against moisture loss. The leaf was then placed on the test stand and the thermocouples taped to the surface. The plant was placed in the freezing chamber and the water applicator turned on.

The first procedure was to slowly lower the air temperature (by adjusting the hand valve - Figure 2) to a level which was thought to be near but slightly above the critical level for the frequency and precipitation rate being used. This temperature was reached within twenty minutes. The test would be run at this temperature for at least an hour. If the plant temperature remained at a safe level, the air temperature was then decreased by one degree and the test continued for another hour. When the plant temperature was just below the damaging level, the air temperature was noted; the test was stopped, and the temperature one degree higher used as the minimum air temperature for frost protection.

The second method used was to quickly lower the air temperature to about 19° F after the plant was in the chamber. This was a damaging temperature for the range of application rates studied. The test would then be continued until there was a constant difference in temperature between the air and plant. This method, of course, kills the plant. The difference between the minimum plant temperature and the air temperature subtracted from the damaging plant temperature is then used as the minimum air temperature for frost protection under the conditions of frequency and precipitation being studied.

The third procedure tried was to first lower the air temperature several degrees below the expected level for frost protection. After the difference between air and plant temperature was constant and it was obvious that the air temperature was too low for protection, the air temperature would be raised in increments to bring the plant temperature to the minimum level for protection. After each air temperature was established, the test was run for one half hour to allow the system to come to equilibrium.

In all cases when the minimum air temperature was reached, the plant temperature was checked at intervals with the precision manual potentiometer. This was because the range of the print potentiometer was too large to permit accurate reading of the plant temperature. At this point, the air temperature was also checked with the manual potentiometer.

Water Temperature Measurement

The temperature of the water was measured by strapping a thermometer to the water line about five feet before the spray nozzle. At various times the temperature of the spray water was measured at the nozzle.

Leaf Selection

To eliminate any possible effect of leaf size, leaves which were about an inch and a quarter wide by an inch and a half long (measured at the widest parts of the leaf) were selected for the tests.

APPARATUS RESULTS & DISCUSSION

Radiation

The radiation measurements with the net radiometer are shown below. The maximum range of radiation is also shown. All radiation measurements are from a very wet dark loamy soil at a temperature of 32° F ± 2 degrees and were taken while the apparatus was in operation.

TABLE III

RADIATION MEASUREMENTS IN FREEZING CABINET

Radiometer Reading (millivolts)	Net Radiation BTU/hr ft²
1.05	25
1.15	27

This net radiation agrees well with experimental values obtained outside on a clear night in East Lansing.

Temperature

The air temperature control was good. Frequent adjusting of the hand valve was necessary at the beginning of a test when a particular temperature was being established. Maximum variation among the three air measuring thermocouples was 1° F. Once the air temperature valve was set after the apparatus had cooled, the temperature remained quite steady.

It was important, however, to be sure the apparatus had an ample supply of dry ice to prevent the radiation as well as the air temperature from decreasing. Once during the tests the alcohol in the cooling pan was changed because it had imbibed sufficient water to lower its effectiveness as a heat transfer medium.

The soil temperature remained very close to 32° F throughout all tests. The heat source (two 60 watt light bulbs) were on throughout all tests.

It was felt the three air temperature thermocouples gave an accurate measurement of the air temperature around the plant. As the air thermocouples were shielded, they were unaffected by freezing spray water as well as radiation.

Precipitation

The water application system was satisfactory, but a calibration of the precipitation for each test was necessary. It was difficult to precalibrate the precipitation system because of the accuracy of setting the time clock and because of the effect of the amount of air flow through the apparatus on the precipitation. For these reasons precipitation tests were always run immediately after the tests under the same conditions on which the test was run.

The range of the water temperatures leaving the nozzle was 42-46° F. The variation in water temperature as measured by the thermometer strapped to the water line was 48 - 52° F.

The pressure regulator maintained a water pressure of eleven pounds per square inch throughout all tests. Once the proper location of the nozzle was determined, it was not changed in order to keep the instantaneous application rate constant.

Wind Speed

The temperature control of the apparatus depended upon bringing in varying amounts of air from the walk-in refrigerator. This caused the average wind speed to be dependent upon temperature. The following table shows the variation in air flow with temperature.

TABLE IV

AIRFLOW FOR VARIOUS TEMPERATURES IN FREEZING CABINET

Temperature (°F)	Airflow Range (f.p.m.)	Airflow Ave. (f.p.m.)	Airflow Ave. (m.p.h.)
28	40-100	75	0.85
25	-	57	0.67
22	-	44	0.50
19	20-50	30	0.34

These airflow measurements were made at the leaf location in a direction parallel to the leaf.

If one considers two or three ranges of temperatures, the variation in airflows is not too large. However, if one compares the data at temperatures around 28° F to that of temperatures around 18° F, it is felt the difference in airflow has some effect. This is definitely one of the limitations of this apparatus. A way to overcome this difficulty in any future work would be to control the temperature of the incoming air, keeping the flow rate constant. It would be quite difficult to provide a radiation panel and eliminate turbulent convective air movement in a small cabinet. However, with some redesign of the air control system, the variations in air movement could be decreased.

Relative Humidity

The relative humidity was measured at various air temperatures. However, no readable wet bulb depression was obtained. This indicates a relative humidity very close to 100 per cent.

Instrumentation

The print potentiometer (range -20° F to 120° F) was very reliable. A calibration against a certified thermometer showed it to be accurate to ± 0.5° F. However, when one is dealing with such a small range of plant temperatures, an instrument with a much narrower range would be desirable. For recording temperature fluctuations in the leaf, a continuous recording galvanometer is necessary. The only temperature which could be determined with the instrument used was an average leaf temperature. Fluctuations in leaf temperature were very hard to determine except at long spraying frequencies.

Plant Temperature Measurement

The location of the thermocouples on top of the leaf was considered satisfactory and required less time than inserting the thermocouple in the plant tissue as is sometimes done. It was felt important to provide a thermocouple for the edge of the leaf facing the wind. This placement is especially important at higher wind speeds. Although no work was carried out on buds or blossoms, it was concluded that one thermocouple with lead wires insulated would be sufficient for plant temperautre measurement in this instance.

Refrigerant Consumption

The average consumption of dry ice was about four pounds per hour.

EXPERIMENTAL RESULTS

Negative Radiation

Limited negative radiation data from various surfaces were taken between 10 p.m. and 2 a.m. at East Lansing on two separate evenings. There was clear sky in both cases. The following table shows these results.

TABLE V

NEGATIVE RADIATION MEASUREMENTS AT EAST LANSING*

Date	Surface	Temp. °F	Air Temp. °F	Radiation (BTU/hr ft²)
Dec. 26, 1958	very wet dark soil	32	20	29.4
Dec. 26, 1958	dark soil frozen	31	20	28.3
Dec. 26, 1958	winter grass	20-22	20	23.5
Dec. 26, 1958	thin film of ice	32	20	28.3
Dec. 27, 1958	water	38	25	31.8
Dec. 27, 1958	dark soil frozen	32	25	26.0

Damaging Leaf Temperatures

The limited data taken on damaging leaf temperatures indicate the following results:

---

*Absolute humidity: 10-20 grains of moisture per lb. dry air.

1) Without an ice coat, the center temperature of the bean leaves reached 29° F before some fringe damage was noticed.

2) With an ice coat, the center temperature was 30.5° F before fringe damage was noticed.

3) With an ice coat, the center temperature reached 29.5° F before center damage was noticed.

It must be pointed out these are approximate determinations because of equipment limitations and because few tests were run.

The conditions under which these tests were run favor supercooling of the plant tissue. There is little shaking of the leaves to disturb a supercooled state if one exists. This would point out that the above observations do not necessarily indicate the freezing point of the cell sap but rather a supercooled temperature capable of being achieved without actual cell sap freezing.

The leaves without an ice coat were able to supercool to a greater degree than those with one. This bears out observations made by other investigators to the extent that there might be an effect of ice innoculation in the latter case.

Without specific knowledge of the freezing point of the cell sap of the beans used and in view of the above results, a plant temperature of 31° F was chosen as the minimum allowable plant temperature for frost protection. In other words when the air temperature level was such that the average plant temperature was 31° F, the minimum air temperature for frost protection was said to have been reached.

Minimum Air Temperature for Frost Control

Table VI shows the experimental results obtained for the minimum air temperature for frost protection. These were determined on the basis of a minimum leaf temperature of 31° F with a net radiant heat transfer from the ice surface of 27 BTU/hr ft^2. The wind speed parallel to the leaf varied from .34 to .85 miles per hour. The relative humidity was about 100 per cent. These results are plotted in Figure 9.

Effect of Increased Wind Parallel to Leaf

A centrifugal fan was installed in the cabinet which gave a steady wind speed parallel to the leaf of 1.25 miles per hour. The purpose of the tests thus run was to observe if there was any marked effect on the edge temperature of the leaf for this increased wind speed.

Four tests were run. In all four cases the entire leading edge of the leaf was damaged. When there was not much ice on the leaf, leading edge temperatures of 26° F were observed even when the center temperature was above 31° F.

TABLE VI

MINIMUM AIR TEMPERATURE FOR FROST PROTECTION

Spraying Frequency (seconds)	Application Rate (inches/hr)	Air Temperature (°F)	Airflow (m.p.h.)
20	.053	27.5	.85
20	.062	25.0	.67
20	.093	21.0	.50
20	.098	21.0	.50
20	.123	19.0	.34
20	.148	18.5	.34
60	.070	26.0	.67
60	.076	26.0	.67
60	.111	21.0	.50
120	.060	28.5	.85
120	.065	28.0	.85
120	.114	22.0	.50
120	.123	21.5	.50
120	.123	20.5	.40

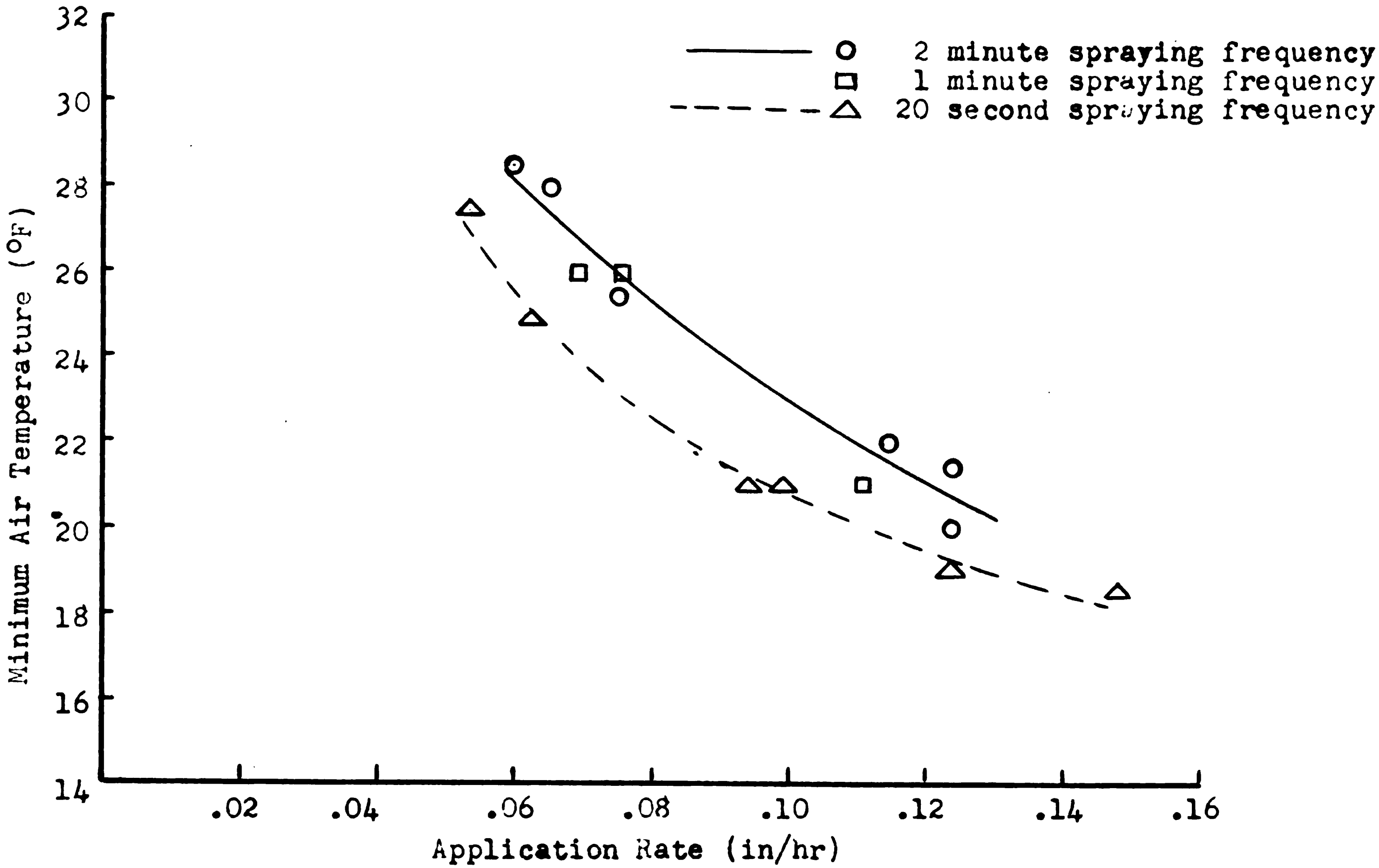

Fig. 9. Minimum temperature for frost protection versus water application rate.

THEORETICAL ANALYSIS OF HEAT TRANSFER IN FROST PROTECTION USING GEOMETRIC ANALOGIES

Many film coefficients have been determined both analytically and empirically for various shaped bodies. Analogies can be made of these shapes to the shapes of the vegetation we are protecting from frost by water application.

In this section various empirical relationships are employed to arrive at film coefficients for the shapes involved. A heat balance can then be set up and the amount of water needed to maintain the temperature of the body at 31.5° F can be computed. These figures are all for 100 per cent efficiency of the water. This means all the water which is applied stays on the surface and is available upon cooling down and freezing. In all cases it is assumed the water strikes the surface at 38° F.

Heat Balance of a Leaf Using Flat Plate Analogy

Treating a leaf and ice coat as a homogenious body, a heat balance can be applied for various conditions:

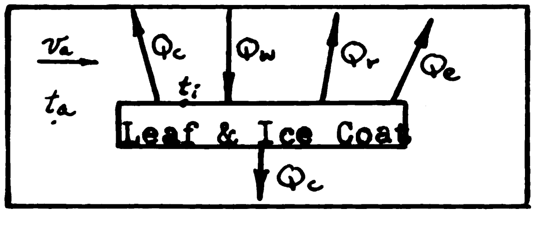

Fig. 10. Heat balance of leaf and ice coat.

where:

Q_e = heat loss by evaporation (BTU/hr ft^2)

Q_r = heat loss by radiation (BTU/hr ft^2)

Q_c = heat loss by convection (BTU/hr ft^2)

Q_w = heat gain from sensible and latent heat of water (BTU/hr ft^2)

v_a = velocity of air parallel to leaf (ft/min)

t_a = temperature of air (°F)

t_i = temperature of leaf and ice coat (°F)

The equation which must be satisfied in the above instance is:

$$Q_w = Q_c + Q_r + Q_e \qquad (1)$$

Based on the radiation measurements made at East Lansing (page 33) and on the reports of other investigators (2) (10), a net negative radiation of 28 BTU/ hr ft^2 from a horizontal, wetted surface will be assumed.

The heat loss by convection is expressed by the following:

$$Q_c = h(t_i - t_a) \qquad (2)$$

where:

h = film coefficient from leaf to air (BTU/hr ft^2 °F)

Based on the differential equation of motion and energy, Geidt (4) suggests the following dimensionless formula for fluid flow parallel to a flat plate in the laminar region ($N_{Re} < 320{,}000$).

$$N_{Nu} = 0.664\, N_{Pr}^{\frac{1}{3}}\, N_{Re}^{\frac{1}{2}} \qquad (3)$$

where:

$$N_{Nu} = hL/k$$

$$N_{Pr} = \mu C_p/k$$

$$N_{Re} = VL\rho/\mu$$

h = film coefficient (BTU/hr ft^2 ^{o}F)

L = plate length (ft)

k = thermal conductivity of fluid (BTU/hr ft ^{o}F)

μ = dynamic viscosity (lb/hr ft)

C_p = specific heat (BTU/lb ^{o}F)

ρ = density of fluid (lb/ft^3)

V = fluid velocity (ft/hr)

For air in the range of temperatures 15 - 30^o F, the physical constants from the standpoint of this treatment can be assumed to remain constant as follows:

C_p = 0.24 BTU/lb ^{o}F

μ = 0.0415 lb/hr ft

ρ = 0.081 lb/ft

k = 0.014 BTU/hr ft ^{o}F

Therefore:

$$N_{Pr} = \frac{(0.0415)(0.24)}{(0.014)} = 0.712$$

$$N_{Re} = \frac{(0.081)\,VL}{(0.0415)} = 1.95\,VL \qquad (4)$$

$$N_{Nu} = \frac{hL}{(0.014)}$$

substituting in equation (3)

$$hL = (0.0083)\, N_{Re}^{\frac{1}{2}}$$

$$h = (0.0116)\left(\frac{V}{L}\right)^{\frac{1}{2}} \qquad (5)$$

If V is expressed in feet per minute and L is expressed in inches, equation (5) becomes the following:

$$h = (0.311)\left(\frac{V}{\ell}\right)^{\frac{1}{2}} \quad Btu./hr.\ ft^{2}\ {}^{\circ}F \qquad (6)$$

The film coefficients computed from this formula are shown in Tables VII and VIII.

The estimation of the evaporation loss is difficult. The following equations taken from Jacob (5) seem to represent the situation. It is assumed in these calculations that the evaporating surface has free moisture on it at all times.

$$\dot{m}'' = b\,\frac{1}{RT_M}\,(p_s - p_o) \qquad (7)$$

where:

$\dot{m}''$ = rate of mass flow per unit area (lbm/ft^2 hr)

b = coefficient of mass transfer (ft/hr)

R = gas constant of diffusing gas (lbf/lbm ^{o}R)

T_M = mean absolute temperature (^{o}R)

p_s = partial pressure at saturation according to surface temperature (lb/ft^2)

p_o = partial pressure of vapor in flowing fluid (lb/ft^2)

Furthermore:

$$b = \frac{\delta}{\alpha}\cdot\frac{h}{\rho C_p}$$

TABLE VII

FILM COEFFICIENT FOR FLAT PLATE OF LENGTH ONE INCH PARALLEL TO AIRFLOW OF VARIOUS INTENSITIES

Wind Speed (m.p.h.)	Wind Speed (ft/min)	$(\frac{v}{x})$	$(\frac{v}{x})^{\frac{1}{2}}$	h (BTU/hr ft^2 °F)
0.10	8.8	8.8	2.96	0.92
0.25	22	22	4.69	1.46
0.50	44	44	6.60	2.06
1.00	88	88	9.37	2.91
2.00	176	176	13.33	4.14
5.00	440	440	20.95	6.53

TABLE VIII

FILM COEFFICIENT FOR AIRFLOW PARALLEL TO FLAT PLATES OF VARIOUS LENGTHS AT A CONSTANT WIND SPEED OF ONE HALF MILE PER HOUR

Length (inches)	$(\frac{v}{x})$	$(\frac{v}{x})^{\frac{1}{2}}$	h (BTU/hr ft^2 °F)
0.5	88	9.4	2.93
1.0	44	6.6	2.06
2.0	22	4.7	1.47
3.0	18	4.2	1.31
4.0	11	3.2	1.00

where:

α = heat diffusivity (ft^2/hr)

δ = mass diffusivity (ft^2/hr)

Jacob (5) gives the following for water vapor diffusing into air.

$$\frac{\alpha}{\delta} = 0.92$$

Therefore:

$$b = \frac{1}{(0.92)} \cdot \frac{h}{(0.081)(0.24)}$$

where:

$$R = 85 \ lbm/lbf \ ^\circ R$$

$$T = 460 + 25 = 485 \ ^\circ R$$

substituting:

$$\dot{m}'' = \frac{56h(70.7)(p_s - p_o)}{(85.8)(485)} = .095h(p_s - p_o) \qquad (8)$$

where p_s and p_o are expressed in inches of mercury.

The latent heat of vaporization of water at 32° F equals 1070 BTU/lb.

then:

$$q'' = (1070)(.095)h(p_s - p_o) = 102h(p_s - p_o) \qquad (9)$$

The quantity $102(p_s - p_o)$ for various air temperatures is shown in Table IX.

Assuming an average leaf and ice coat temperature of 31.5° F (31° F bottom surface, 32° F top surface), the heat balance equation can be written and the required amount of sensible and latent heat from the water can be calculated. These calculations assume the same film coefficient on the upper and lower sides of the leaf.

$$Q_w = Q_r + Q_c + Q_e \quad (\text{BTU/hr ft}^2\,°) \qquad (1)$$

$$Q_w = 28 + h(31.5 - t_a) + h(31.5 - t_a) + 102\,h(p_s - p_o)$$

$$Q_w = 28 + h\left[2(31.5 - t_a) + 102(p_s - p_o)\right] \qquad (10)$$

Assuming the water is applied to the flat plate at a temperature of 38° F, the amount of heat available from each pound of water which freezes is as follows:

$$q_w = 144 + 6 = 150 \text{ BTU./lb.}$$

This figure can be converted to the heat available per square foot of area for various application rates. In equation form the available heat can be expressed as follows:

$$Q_w = \frac{w(150)(62.4)}{(12)} = 780\,w \quad (\text{BTU/hr. ft}^2) \qquad (11)$$

where:

w = water application rate (in/hr)

The required application rate for frost protection can now be found:

$$w = \frac{28 + h\left[2(31.5 - t_a) + 102(p_s - p_o)\right]}{780} \quad \text{in./hr.} \qquad (12)$$

Table X shows the theoretical application rate computed from equation (12) for certain conditions.

The heat loss by convection when there is no air movement is called heat loss by natural convection. In this instance, the equations have a different form than for forced convection because the degree of fluid motion depends upon the difference in temperature between the fluid and the body.

For natural convection cooling of low vertical plates,

TABLE IX

THE QUANTITY 102 $(P_s - P_o)$ FOR VARIOUS AIR TEMPERATURES

Temp. °F	P Partial Pressure (in of Hg) of water vapor at 100% R.H.	$P_s - P_o$	102 $(P_s - P_o)$
32	.180	0.000	0.0
30	.165	0.015	1.5
28	.150	0.030	3.1
26	.137	0.043	4.4
24	.124	0.056	5.7
22	.113	0.067	6.8
20	.103	0.077	7.9
18	.093	0.087	8.9
16	.085	0.095	9.7

TABLE X

THEORETICAL APPLICATION RATE FOR WIND SPEED PARALLEL TO PLATE OF ONE HALF M.P.H., LENGTH OF PLATE OF 1 INCH, R.H. 100 PERCENT

Air Temperature (°F)	Application Rate - w (in/hr)
30	.048
28	.063
26	.076
24	.091
22	.104
20	.118
18	.131
16	.143

N_{Nu} can be obtained as a function of the product, $(N_{Gr}\ N_{Pr})$ from Saunders' experimental curve (9). At a film temperature of 28° F, the quantity $(N_{Gr}\ N_{Pr})$ can be represented sufficiently accurately for the purpose of this presentation as follows:

$$N_{Gr} N_{Pr} = 1.31 \times 10^{3} (D^{3})(\Delta t) \qquad (13)$$

where:

D is expressed in inches

Δt is expressed in °F

Table XI shows the film coefficient computed at various air temperatures for a plate of one inch length.

Fishenden and Saunders (5) recommend the following for the film coefficient for the upper side of a plate cooled in air:

$$h_u = 0.27 \left(\frac{\Delta t}{L}\right)^{0.25} \qquad (14)$$

For horizontal plates facing downward, they recommend:

$$h_l = 0.12 \left(\frac{\Delta t}{L}\right)^{0.25} \qquad (15)$$

For the purposes of this discussion an average film coefficient can be defined thus:

$$h_{ave.} = \frac{0.27+0.12}{2} \cdot \left(\frac{\Delta t}{L}\right)^{0.25} = 0.2 \left(\frac{\Delta t}{L}\right)^{0.25} \qquad (16)$$

Table XII shows the film coefficients computed from equation (16)

In the field the leaves which approach a flat plate in shape are oriented in all directions with respect to the wind. Film coefficients were shown for the plate oriented parallel to the wind. For basis of comparison, the film coefficient

TABLE XI

FREE CONVECTION COEFFICIENTS FOR A VERTICAL PLATE OF LENGTH 1 INCH AND APPLICATION RATE USING THESE COEFFICIENTS.

t (°F)	$(N_{Gr}\ N_{Pr})$	$\log (N_{Gr}\ N_{Pr})$	$\log N_{Nu}$	N_{Nu}	h (BTU/hr ft^2 °F)	w (in/hr)
30	1.97×10^3	3.29	0.65	4.47	0.745	0.040
26	7.21×10^3	3.86	0.74	5.50	0.925	0.054
22	1.24×10^4	4.09	0.77	5.89	0.990	0.069
18	7.65×10^4	4.25	0.80	6.31	1.060	0.085

TABLE XII

NATURAL CONVECTION COEFFICIENT FOR A 1 INCH HORIZONTAL PLATE COOLING IN AIR AND APPLICATION RATE USING THESE COEFFICIENTS.

Air Temperature °F	h (BTU/hr ft^2 °F	w (inches/hr)
30	.413	.038
26	.570	.047
22	.653	.058
18	.715	.069

for the plate perpendicular to the wind can be shown for a plate of three inch length.

Hilpert (5) represented forced convection perpendicular to cylinders by the following equation:

$$N_{Nu} = B N_{Re}^{n} \qquad (17)$$

For flow perpendicular to a flat plate, Reiher (5) suggests 0.205 for the constant B and 0.731 for the exponent n. This is valid in the region $4{,}000 < N_{Re} < 15{,}000$ when the characteristic length used is the diameter of the cylinder which has the same surface area per unit length as the flat plate.

Therefore:

3 inch plate - surface area = 6 in^2 per inch of length.

or:

$$6 = \pi D$$

$$D = \frac{6}{3.14} = 1.9 \text{ in.}$$

substituting in equations (4) and (5) for a windspeed of 3 miles per hour:

$$N_{Re} = \frac{(1.95)(3)(88)(60)(1.9)}{(12)} = 4900$$

and

$$h = \frac{(0.014)(12)(0.205)(4900)^{.731}}{1.9} = 9.1 \text{ BTU./hr. ft}^2 \text{ °F}$$

The film coefficient for air flow parallel to a 3 inch plate at a wind speed of 3 m.p.h. is as follows:

$$h = (0.311)\left[\frac{(3)(88)}{(3)}\right]^{\frac{1}{2}} = 3 \text{ BTU/hr. ft}^2 \text{ °F}$$

Figures 11 through 14 show curves of application rate versus air temperature for a flat plate under various conditions.

Heat Balance of an Exposed Blossom or Bud Using a Sphere Analogy

The problem here is slightly more difficult than the case before where the area over which the water was applied was just one half the area of the shape. In the case of a sphere or cylinder, this is not the case.

A sphere intercepts the falling water according to its projected area or the area of a circle of the same diameter as the sphere. This is important with respect to frost protection as will be shown.

Assume:

A = surface area of sphere (square inches)

D = diameter of the sphere. (inches)

ω = application rate (inches per hour)

φ_ω = heat available from water which impinges on the sphere (BTU/hr)

Solving for φ_ω:

$$\varphi_\omega = \frac{\omega \pi D^2 (62.4)(150)}{(4)(1728)} = 4.27 \omega D^2 \text{ BTU/hr.}$$

The heat balance equation for a sphere can now be written:

$$\varphi_\omega = \varphi_r + \varphi_c + \varphi_e \qquad (1)$$

For the horizontal flat plate, the upper surface "sees"

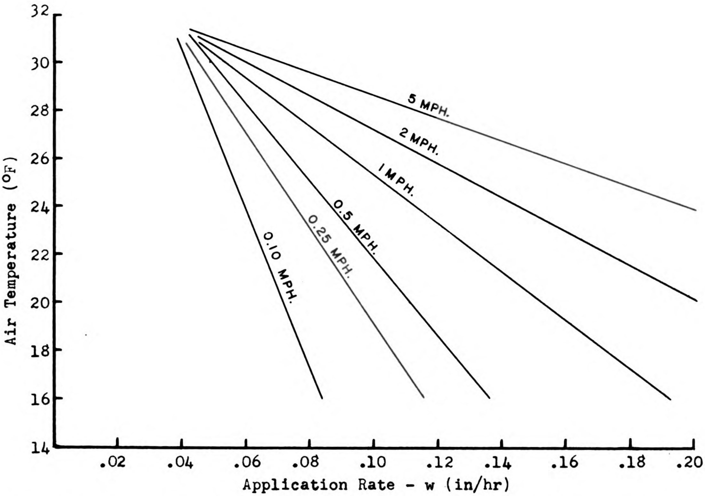

Fig. 11. Theoretical application rate for a flat plate of length one inch for various air temperatures and wind speeds parallel to plate.

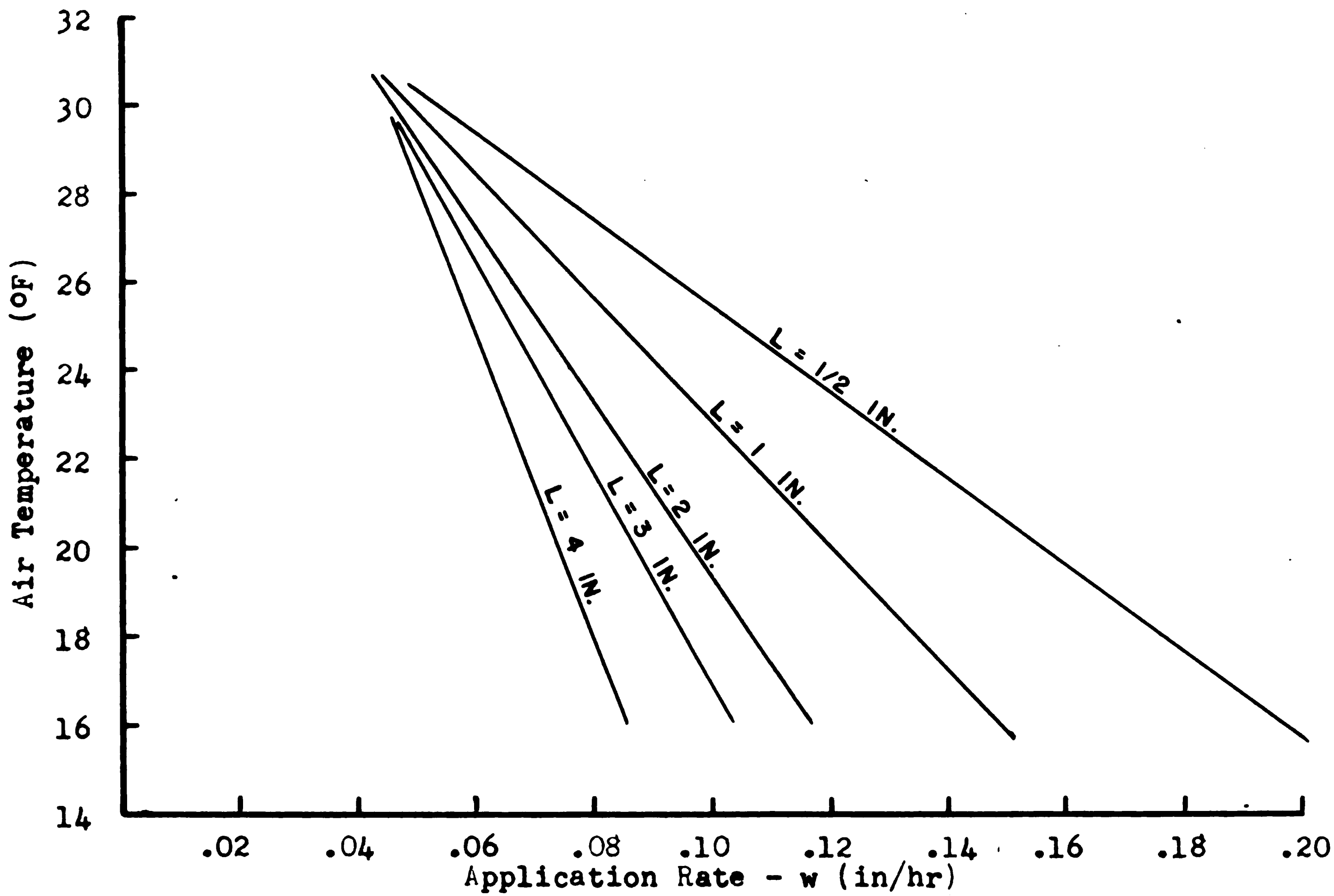

Fig. 12. Theoretical application rate versus air temperature for flat plates of various lengths at a wind speed of one half mile per hour parallel to plate.

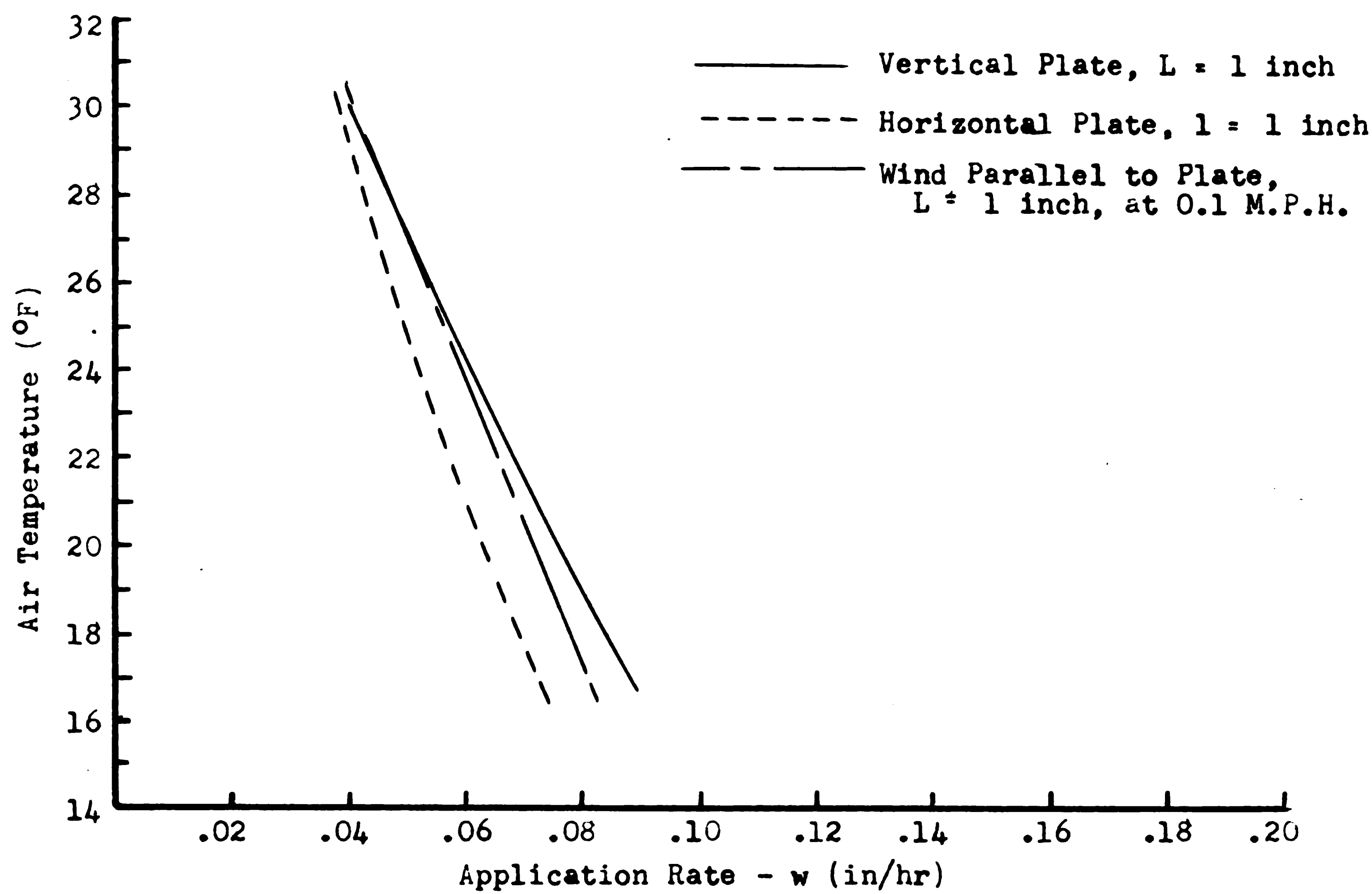

Fig. 13. Theoretical application rate versus air temperature for horizontal and vertical flat plates of one inch length under natural convection cooling. Curve also shown for forced convection at 0.1 miles per hour parallel to a one inch plate.

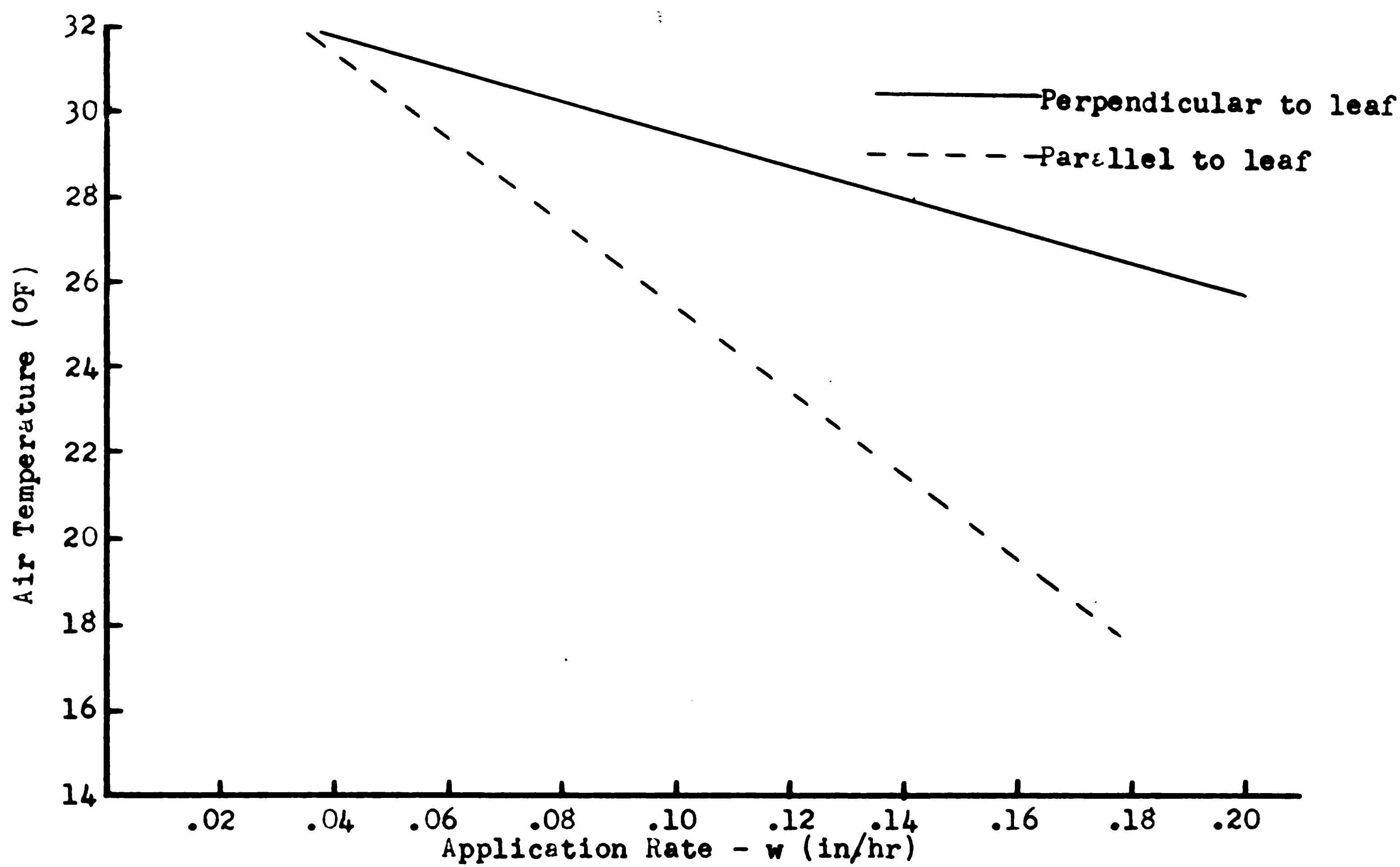

Fig. 14. Theoretical application rate for wind of three miles per hour parallel and perpendicular to a three inch plate.

only the sky. Hence, the net radiation is from a surface of a certain emissivity to a black body. If we treat a sphere as a body whose lower half does not "see" the sky but whose upper half "sees" both the sky and the ground, each infinitesimal area on the upper hemisphere exchanges radiation from its surroundings. The net radiant heat transfer from this infinitesimal area depends upon what percentage of the sky it "sees" in relation to the ground. The total net radiant heat transfer is then a summation of the heat transfer from these small areas.

In this discussion, the shape factors from the sphere and cylinder to the sky and ground are not going to be shown. Rather it will be assumed that the sphere radiates at the net rate of 28 BTU/hr ft^2 from six tenths of its upper hemisphere, and the cylinder radiates at the net rate of 28 BTU/hr ft^2 from eight tenths of its upper surface (because a cylinder geometrically lies between a sphere and a flat plate).

The net radiation from the sphere is therefore:

$$Q_r = \frac{(28)(0.6)(\pi)(D^2)}{(144)(2)} = 0.184 D^2 \text{ (BTU/hr.)}$$

The convection and evaporation loss can be represented by:

$$Q_{cre} = \frac{\pi D^2 h}{144}\left[(31.5 - t_a) + 102(p_s - p_o)\right] \text{ BTU/hr.}$$

Writing the heat balance and substituting:

$$4.27 \omega D^2 = 0.184 D^2 + \frac{3.14 h D^2}{144}\left[(31.5 - t_a) + 102(p_s - p_o)\right]$$

$$\omega = \frac{0.184 + .0218\, h\left[(31.5 - t_a) + 102(p_s - p_o)\right]}{4.27} \qquad (17)$$

For forced convection heat transfer from a sphere, McAdams (13) shows data correlated by G. C. Williams. From $20 < N_{Re} < 150{,}000$, the following empirical equation can be used:

$$N_{Nu} = 0.33\, N_{Re}^{\,0.6} \tag{18}$$

$$h = \frac{(0.014)(12)(0.33)}{D}\left[\frac{(1.95)(60)(D)(v)}{12}\right]^{0.6}$$

where:

v is expressed in feet per minute

D is expressed in inches

This simplifies to:

$$h = 0.218\,\frac{v^{0.6}}{D^{0.4}} \quad Btu/hr.\,ft^2\,{}^\circ F \tag{19}$$

Film coefficients computed from equation (19) are shown in Tables XIII and XIV.

The coefficient of natural convection from a sphere and cylinder can be well represented by W. J. King's (5) curves representing his experiments on natural convection from horizontal cylinders, vertical surfaces, blocks and spheres.

To use King's data, first the quantity log $(N_{Gr}\, N_{Pr})$ must be determined:

From equation (13) the quantity $(N_{Gr}\, N_{Pr})$ is:

$$N_{Gr}\, N_{Pr} = 1.31 \times 10^3\,(D^3)(\Delta t)$$

TABLE XIII

FILM COEFFICIENT FOR FORCED CONVECTION FOR A SPHERE ONE INCH IN DIAMETER FOR VARIOUS WIND SPEEDS

Wind Speed (m.p.h.)	Wind Speed (ft/min)	h (BTU/hr ft^2 °F)
0.10	8.8	0.805
0.25	22	1.40
0.50	44	2.17
1.00	88	3.23
2.00	176	4.85
5.00	440	8.47

TABLE XIV

FILM COEFFICIENT FOR FORCED CONVECTION FOR A SPHERE OF VARIOUS DIAMETERS WITH A CONSTANT WIND SPEED OF ONE HALF M.P.H.

Diameter of Sphere (inches)	h (BTU/hr ft^2 °F)
0.5	2.86
1.0	2.17
2.0	1.65

Once the Nusselt number has been determined, the film coefficient is:

$$h = \frac{k N_{Nu}}{d}$$

$$h = \frac{(0.014)(12) N_{Nu}}{D}$$

$$h = \frac{0.168 N_{Nu}}{D} \qquad (20)$$

The natural convection coefficients from King's curves and equation (20) are shown in Tables XV and XVI. Figures 15 through 17 show theoretical application rate versus air temperature for spheres under various conditions.

Heat Balance of a Branch with Buds or Blossoms Using a Cylinder Analogy

As discussed before, it will be assumed that the net radiation from a horizontal cylinder is 28 BTU/ hr ft^2 from eight tenths of its upper surface.

Therefore:

$$Q_r = \frac{28 \pi D L (0.8)}{(2)(144)}$$

where:

D = diam of cylinder in inches

L = length of cylinder in inches

The heat loss by convection and evaporation is:

$$Q_{c+e} = \pi D L h \left[(31.5 - t_a) + 102 (p_s - p_o) \right]$$

As in the case of the sphere, the amount of water which will strike the cylinder depends upon its projected area. The

TABLE XV

THEORETICAL APPLICATION RATE AND NATURAL CONVECTION FILM COEFFICIENTS FOR A ONE HALF INCH DIAMETER SPHERE
(from King's curves)

t (^oF)	$(N_{Gr}\ N_{Pr})$	log $(N_{Gr}\ N_{Pr})$	log N_{Nu}	N_{Nu}	h (BTU/hr ft^2 ^oF)	w (in/hr)
30	2.46×10^2	2.39	0.4	2.51	0.85	.056
26	9.00×10^2	2.95	0.5	3.16	1.06	.097
22	1.56×10^3	3.19	0.5	3.31	1.11	.135
18	2.21×10^3	3.34	0.6	3.98	1.38	.201

TABLE XVI

THEORETICAL APPLICATION RATE AND NATURAL CONVECTION FILM COEFFICIENTS FOR A ONE INCH DIAMETER SPHERE
(from King's curves)

t (^oF)	$(N_{Gr}\ N_{Pr})$	log $(N_{Gr}\ N_{Pr})$	log N_{Nu}	N_{Nu}	h (BTU/hr ft^2 ^oF)	w (in/hr)
30	1.97×10^3	3.29	0.55	3.55	0.596	.052
26	7.21×10^3	3.86	0.70	5.02	0.843	.086
22	1.24×10^4	4.09	0.73	5.37	0.902	.118
18	7.65×10^4	4.25	0.80	6.31	1.060	.164

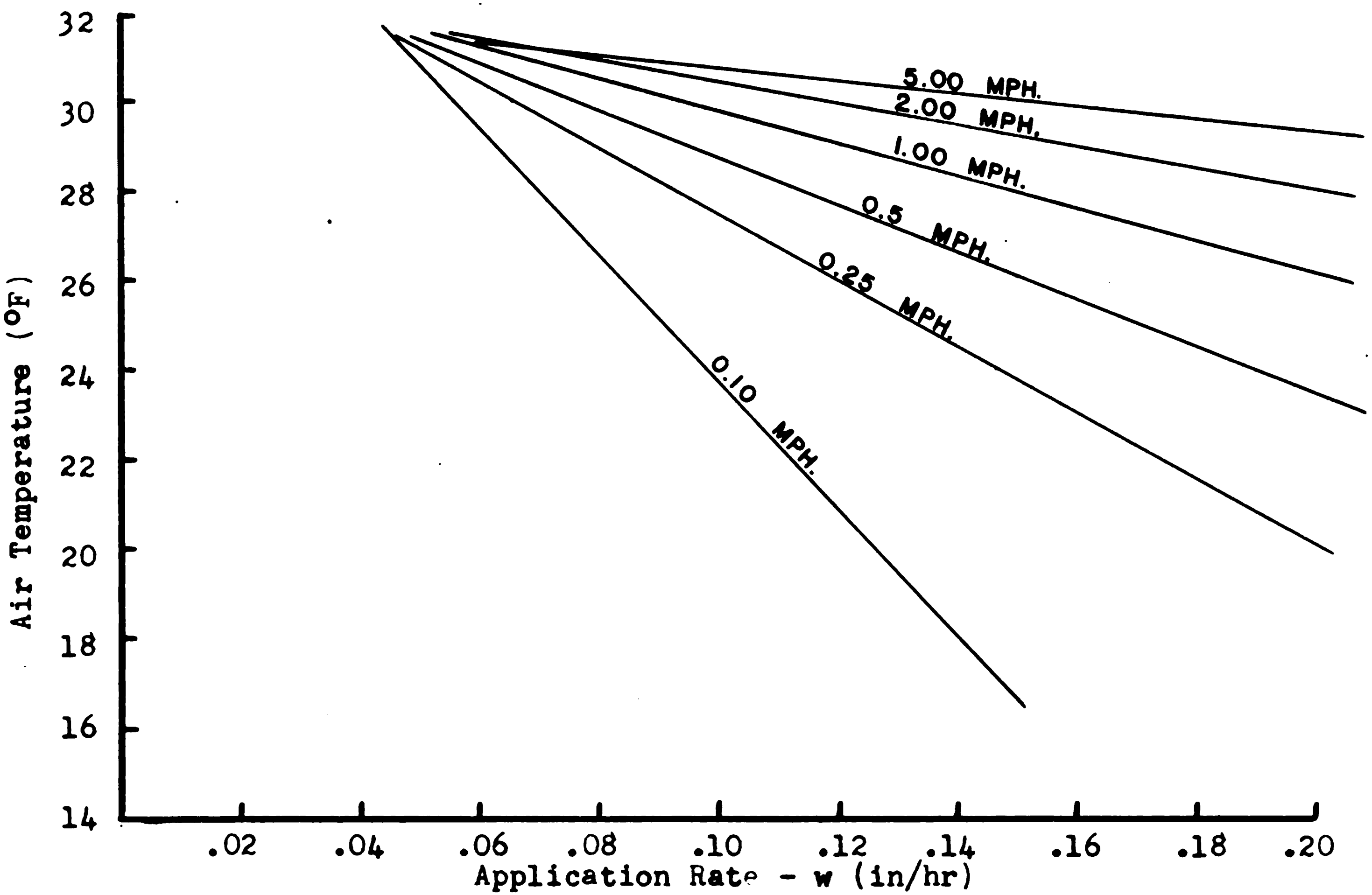

Fig. 15. **Theoretical application rate for a one inch sphere at various wind speeds.**

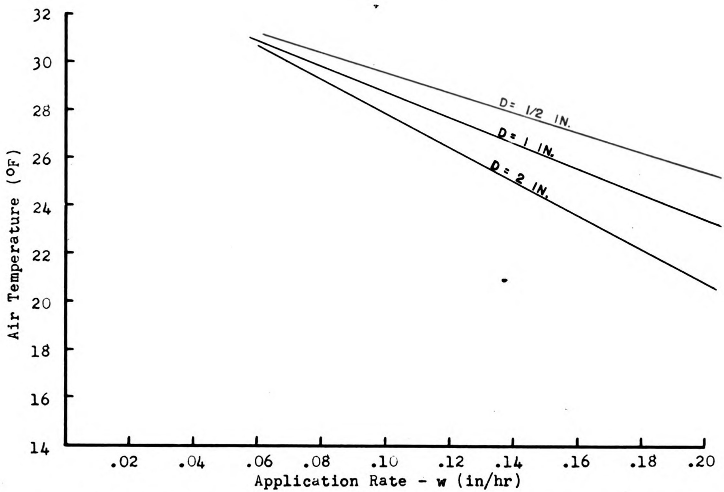

Fig. 16. Theoretical application rate for a sphere of various diameters at a constant wind speed of three miles per hour.

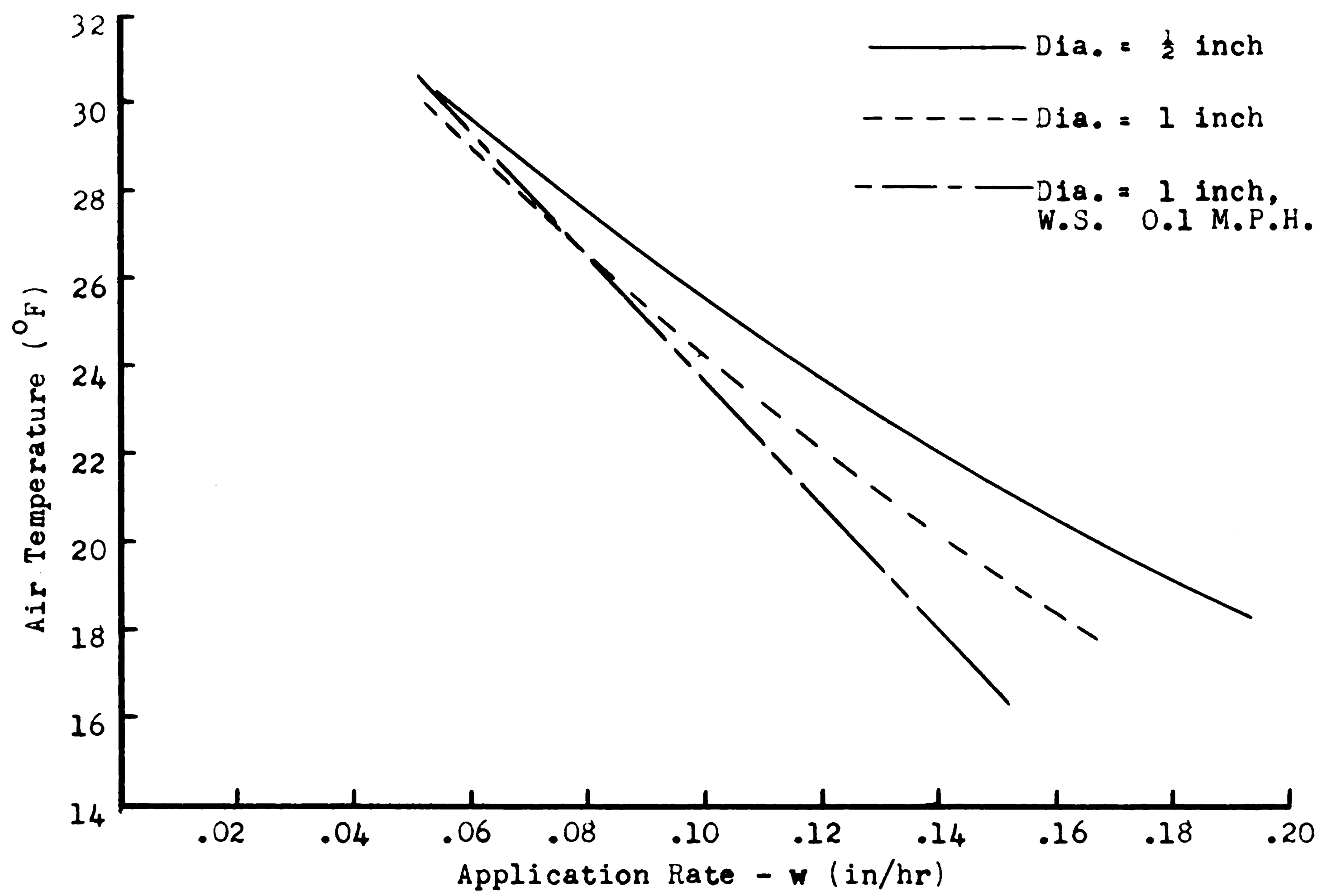

Fig. 17. Theoretical application rate for spheres of various diameters under natural convection conditions. Also shown is forced convection cooling of 0.1 miles per hour for a sphere one inch in diameter.

heat available from the water can be written as follows:

$$Q_w = \frac{150 w DL (62.4)}{1728} = 5.42 w DL \qquad (\text{BTU./hr.})$$

The heat balance equation may now be written:

$$5.42 w DL = 0.245 DL + \frac{\pi DL h}{144}\left[(31.5 - t_a) + 102(p_s - p_o)\right]$$

$$w = \frac{0.245 + .0218 h\left[(31.5 - t_a) + 102(p_s - p_o)\right]}{5.42} \quad \text{in./hr.} \qquad (22)$$

As mentioned before, Hilpert (5) represented his empirical results for forced convection perpendicular to cylinders by the following expression:

$$N_{Nu} = B N_{Re}^{\,n} \qquad (17)$$

In the range of Reynolds numbers $40 < N_{Re} < 40{,}000$, he suggests 0.466 for the exponent n and 0.615 for the coefficient B. Applying these constants yields:

$$\frac{hD}{k} = (12)(0.615)(9.75\, v D)^{.466}$$

$$h = (0.014)(12)(2.89)\, v^{.466} / D^{.534}$$

$$h = 0.486\, v^{0.466} / D^{0.534} \qquad (23)$$

where:

v is expressed in ft/min

D is expressed in inches

These film coefficients are shown in Tables XVII and XVIII.

For unconfined air flow parallel to a cylinder in the laminar range, Pohlhausen's (5) theoretical equation for air, which has been born out experimentally by Jacob and Dow, is as follows:

TABLE XVII

FILM COEFFICIENT FOR FLOW PERPENDICULAR TO A CYLINDER ONE INCH IN DIAMETER

Wind Speed (m.p.h.)	Wind Speed (ft/min)	h (BTU/hr ft^2 °F)
0.10	8.8	1.34
0.25	22.0	2.05
0.50	44.0	2.84
1.00	88.0	3.93
2.00	176.0	5.14
5.00	440.0	7.52

TABLE XVIII

FILM COEFFICIENT FOR AIRFLOW OF ONE HALF MILE PER HOUR PERPENDICULAR TO CYLINDERS

Diameter of cylinder (inches)	h (BTU/hr ft^2 °F)
0.5	4.12
1.0	2.84
2.0	1.96

$$N_{Nu} = 0.592 (N_{Re})^{\frac{1}{2}}$$

$$= 0.592 (117 vd)^{\frac{1}{2}}$$

$$h = (.0894) v^{\frac{1}{2}} / d^{\frac{1}{2}}$$

where:

d is expressed in feet

v is expressed in ft/min

For natural convection film coefficients from horizontal cylinders, King's data that was used for spheres can also be used. The theoretical application rate will be different however, because the ratio of projected area to total area is different for cylinders than spheres. Tables XIX and XX show the theoretical application rate computed from equation (22).

Figures 18 through 20 show the application rate versus air temperature for cylinders under various conditions. Figures 21 through 23 show comparisons of the various shapes.

Effect of Angle of Surface with Respect to Direction of Water Motion

If one considers a sphere, it does not matter from what direction it is viewed because it always looks the same. This, however, is not true of a cylinder or a flat plate. From certain directions a cylinder looks like a long rectangle, from others a shorter rectangle, and from even others a circle. So also a rectangle can appear to be of different size depending upon from what angle it is viewed.

The only thing that is important concerning frost

TABLE XIX

WATER APPLICATION RATE FOR A HORIZONTAL CYLINDER ONE HALF INCH IN DIAMETER UNDER NATURAL CONVECTION CONDITIONS

Air Temperature (°F)	w (in/hr)
30	.056
26	.088
22	.118
18	.169

TABLE XX

WATER APPLICATION RATE FOR A HORIZONTAL CYLINDER ONE INCH IN DIAMETER UNDER NATURAL CONVECTION CONDITIONS

Air Temperature (°F)	w (in/hr)
30	.052
26	.079
22	.104
18	.141

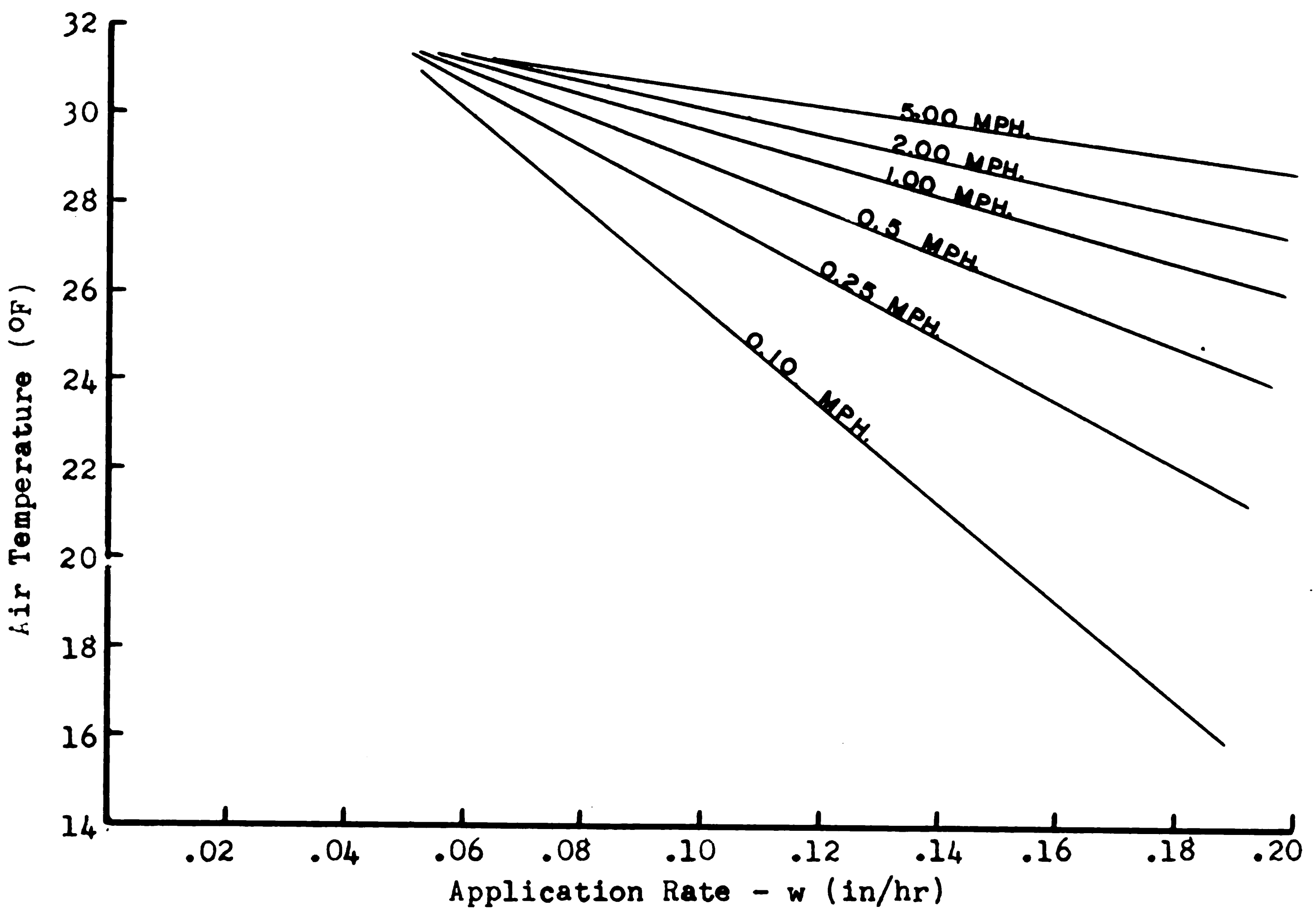

Fig. 18. Theoretical application rate for airflow perpendicular to cylinders of one inch diameter at various wind speeds.

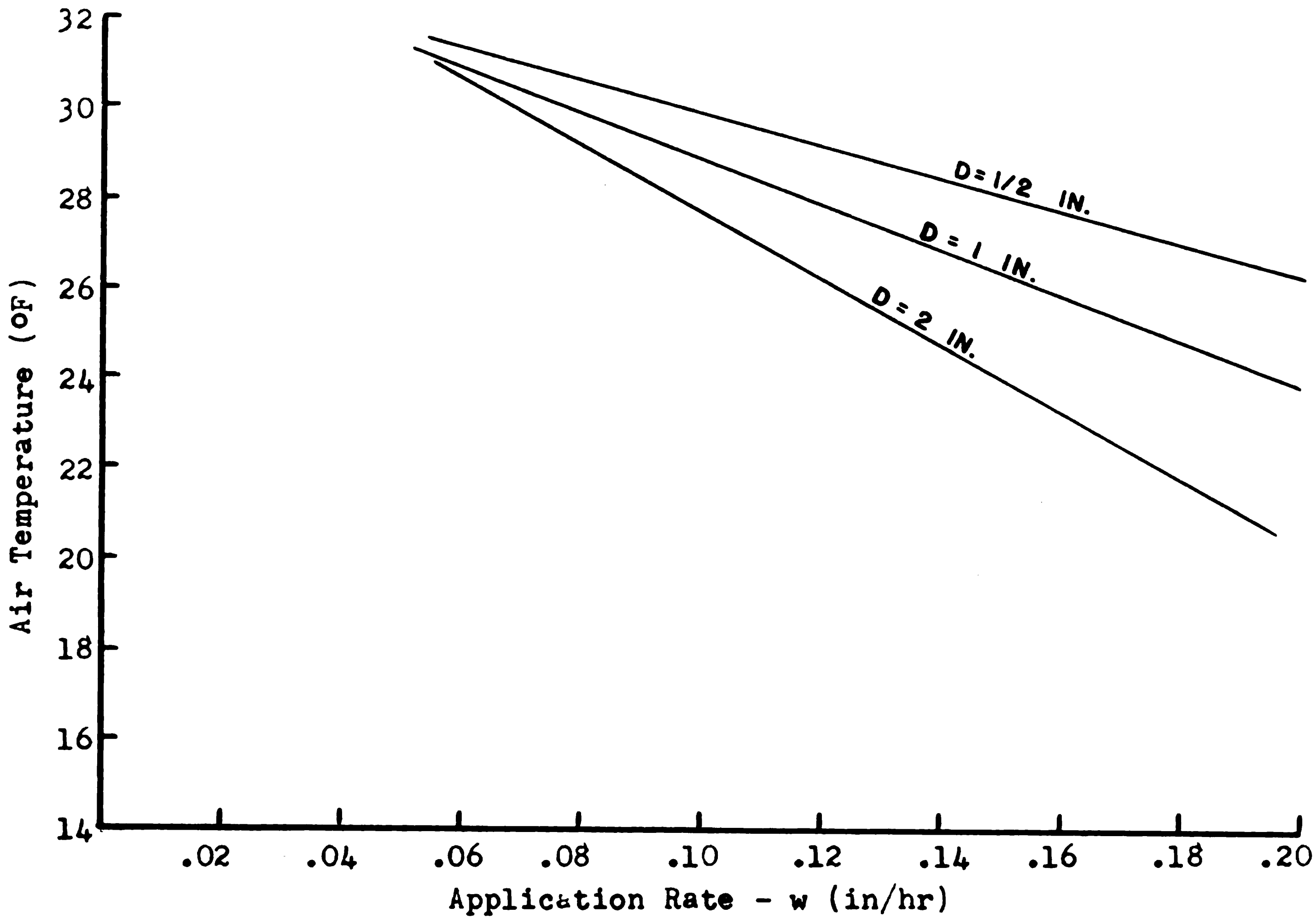

Fig. 19. Theoretical application rate for airflow perpendicular to cylinders of various diameters at a constant wind speed of one half mile per hour.

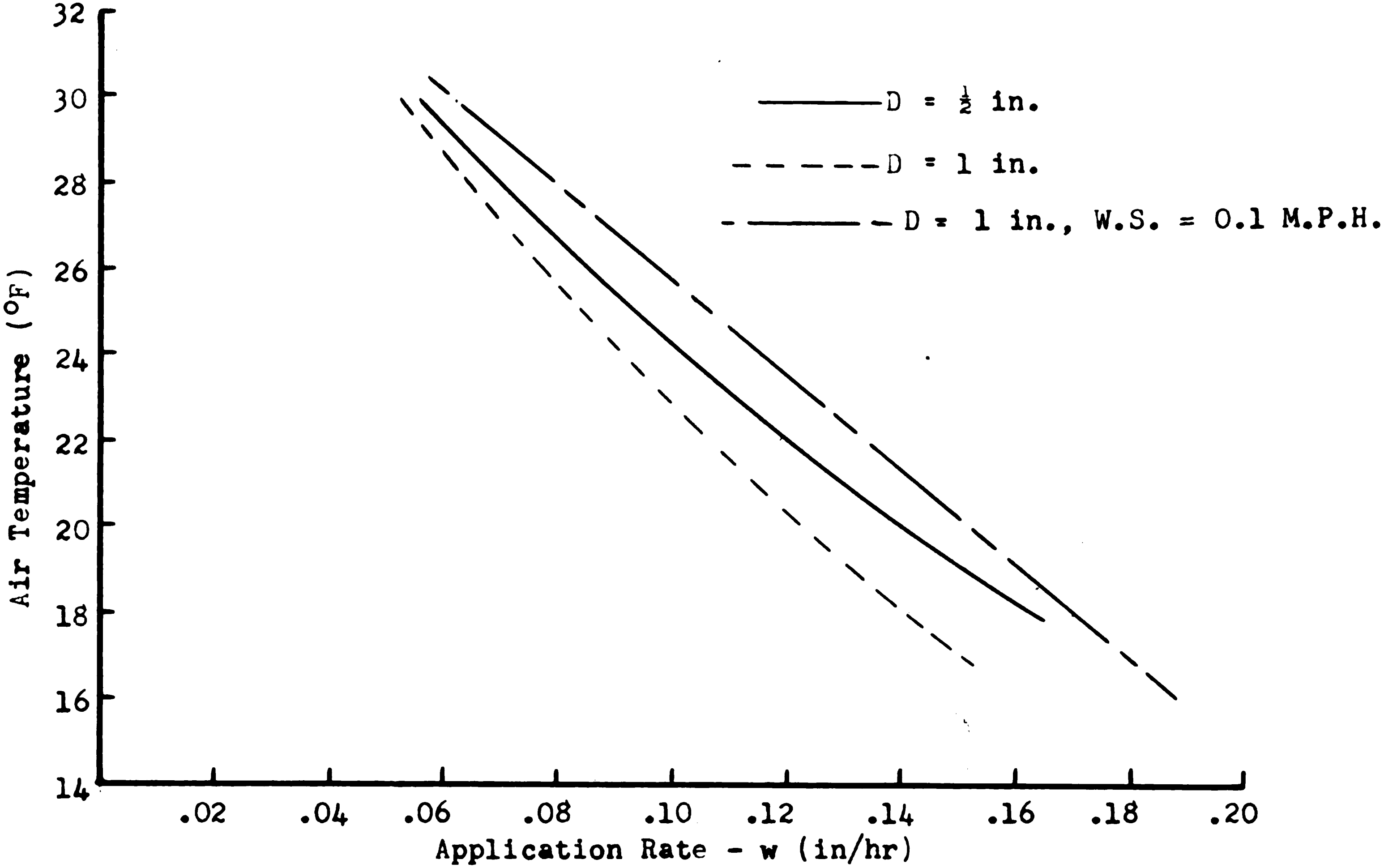

Fig. 20. **Theoretical application rate for cylinders of various diameters under natural convection conditions. Also shown is the curve for forced convection perpendicular to cylinders at a wind speed of 0.1 miles per hour.**

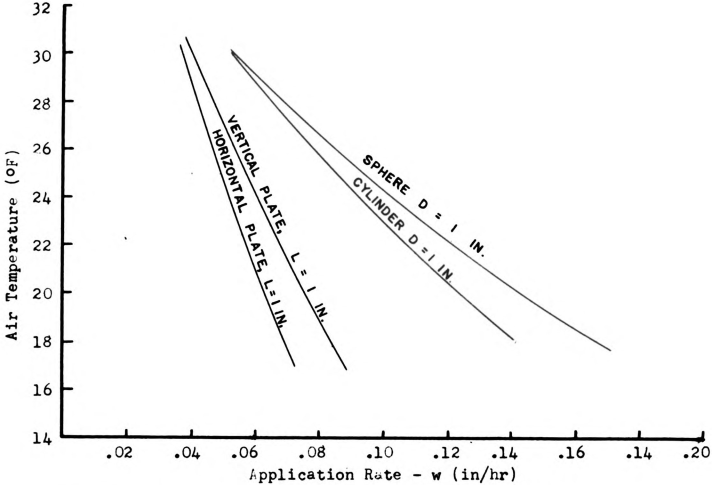

Fig. 21. Comparison of theoretical application rate under natural convection conditions for various shapes.

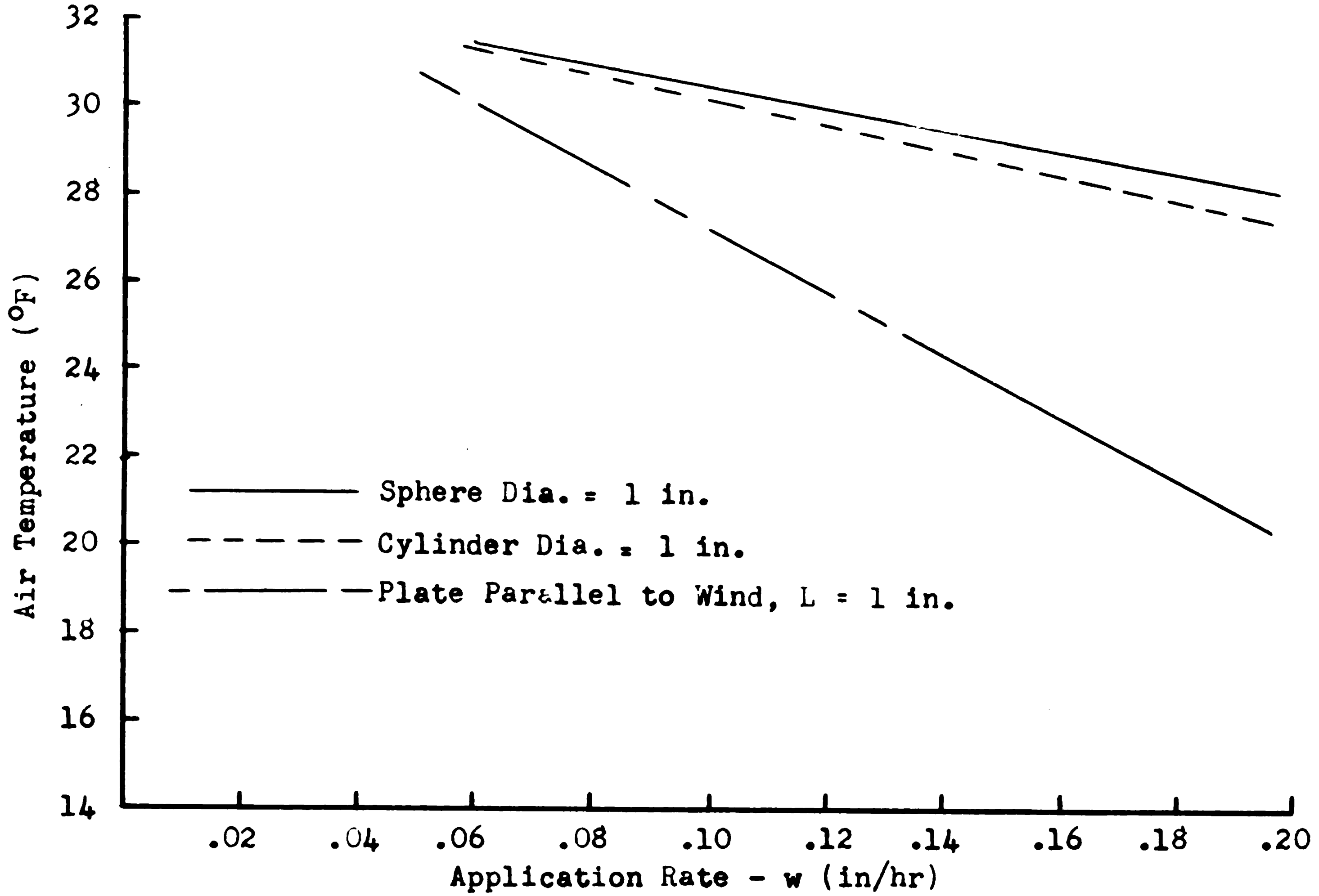

Fig. 22. Comparison of theoretical application rate for various shapes at a wind speed of two miles per hour.

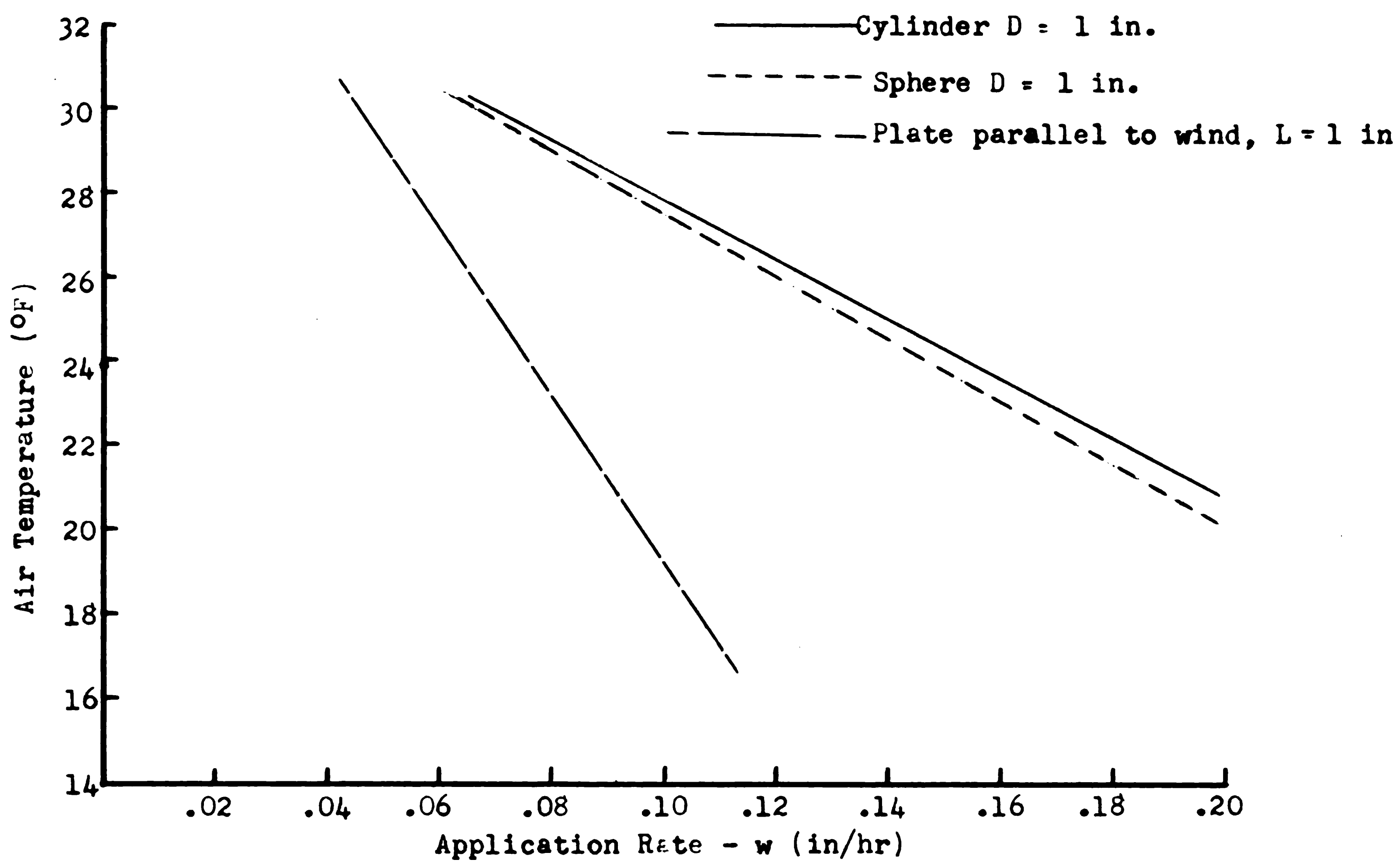

Fig. 23. Comparison of theoretical application for various shapes at a wind speed of 0.25 miles per hour.

protection is the local application rate on the vegetation we are trying to protect. The precipitation rate from a sprinkler is computed from the sprinkler discharge and the area of the wetted circle. It can be measured by placing cans on the ground and measuring the depth of water collected after a given period. However, if these cans are tilted at an angle of 45° F the depth of water collected in them would be nowhere near as great as before because the projected area perpendicular to the direction of the water is less. The following is a simple geometric development showing the magnitude of this effect.

Let a rectangular plate of area A square inches be perpendicular to the direction of the falling water. This plate is being sprinkled at the rate of w inches per hour.

When the plate is perpendicular to the direction of water, the volume of water intercepted is (A)(W) cubic inches per hour. However, when theplate is at an angle from the horizontal the volume of water intercepted is as follows:

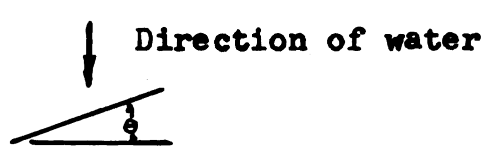

Fig. 24. Plate at an angle from the horizontal.

Let the projected area perpendicular to direction of water equal A'

Then:

$$A' = A \cos \theta$$

Therefore the volume of water collected, V', is:

$$V' = w A \cos \theta$$

The significance of this lies in the fact that leaves at various angles intercept varying volumes of water, but the only part of their cooling load which could be theoretically decreased would be the radiation because the leaf now "sees" more of the ground than it did before.

The ratio of the volume of water intercepted per unit time to the volume intercepted by a surface perpendicular to the direction of the water can be plotted against the angle θ. This can be expressed as follows:

$$\frac{V'}{V} = \frac{\omega A \cos \theta}{\omega A} = \cos \theta$$

This also can be shown graphically as in Figure 25.

Effect of Ice Thickness

Niemann (10) suggests that there are three different situations in frost control. The first is where all the water has not frozen before the sprinkler passes the surface again. This represents a waste of water. The second is where all the water freezes just before the next sprinkler pass is made. The third is when all the water freezes some time before the next sprinkler pass is made. Assuming the water is applied to the vegetation at 32° F, the first two conditions lead to constant temperature conditions.

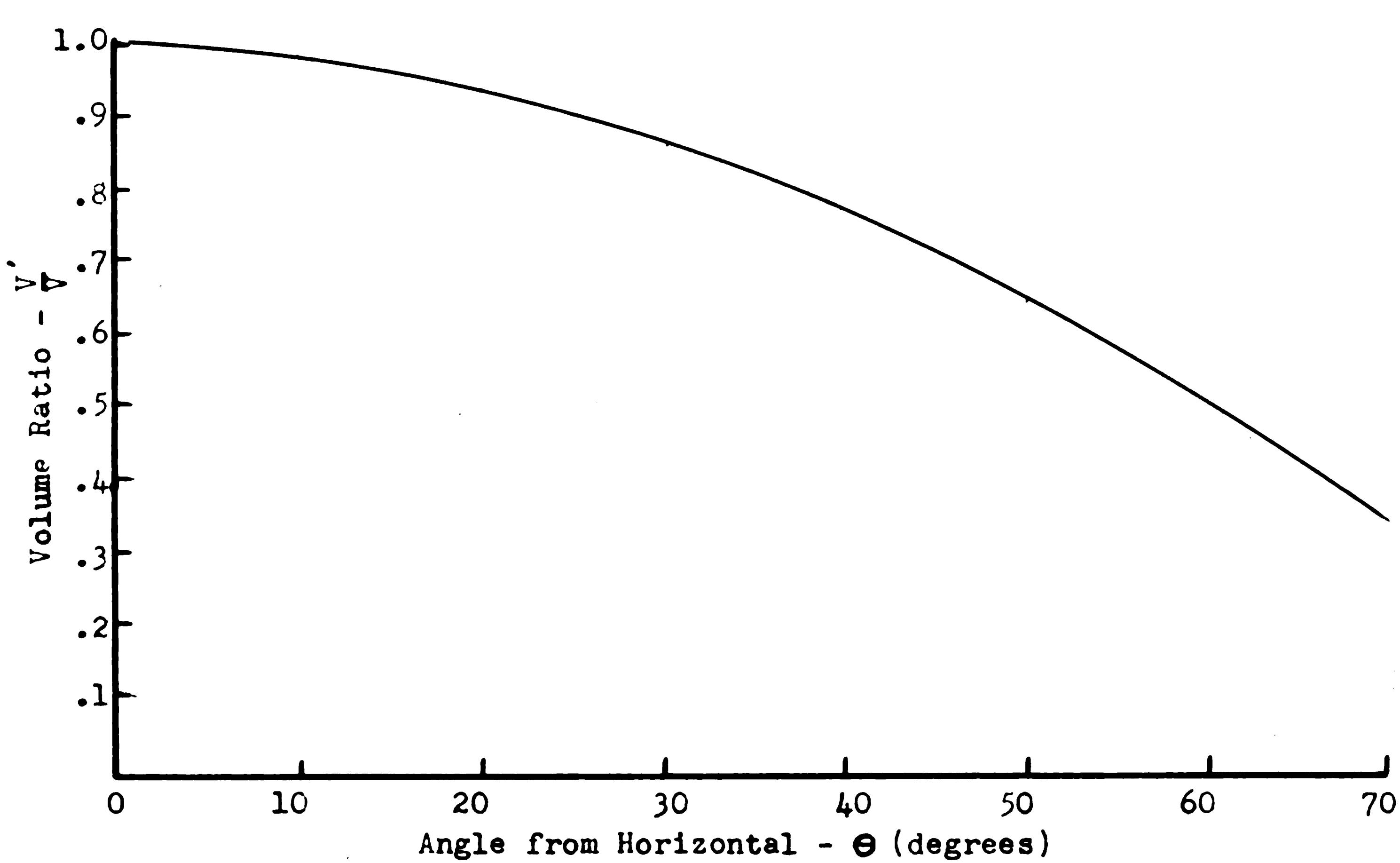

Fig. 25. Effect of angle θ on ratio of volume of water intercepted to volume intercepted for horizontal plate.

The third condition leads to fluctuations in ice and plant temperature which will be discussed later. For the first two steady state cases an interesting effect of ice thickness can be shown as follows:

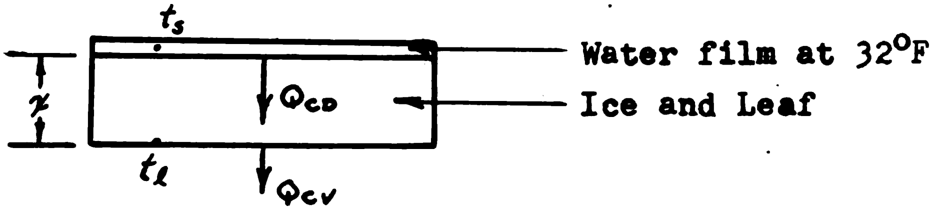

Fig. 26. One dimensional steady-state heat flow through a body with a constant temperature source at one side.

Notation:

t_l = temperature of the leaf

t_a = temperature of the air

h = film coefficient from leaf to air

Q_{CD} = conduction through the leaf and ice coat

Q_{CV} = convection from bottom of leaf

t_s = temperature of surface of ice = 32° F

x = ice thickness

k_i thermal conductivity of ice (leaf considered part of ice for practical purposes)

For the case of water freezing on the surface at all times, the following must be true:

$$Q_{CD} = Q_{CV}$$

This can also be written:

$$\frac{k_i (32 - t_l)}{x} = h (t_l - t_a)$$

Simplifying and solving for the temperature of the leaf:

$$t_l = \frac{32k_i + h t_{air}}{h_r + k_i}$$

This result is plotted in Figure 27 for air temperatures of 20^{o} F and 25^{o} F and for film coefficients of 1 BTU/ hr ft^2 °F and 4 BTU/hr ft^2 °F.

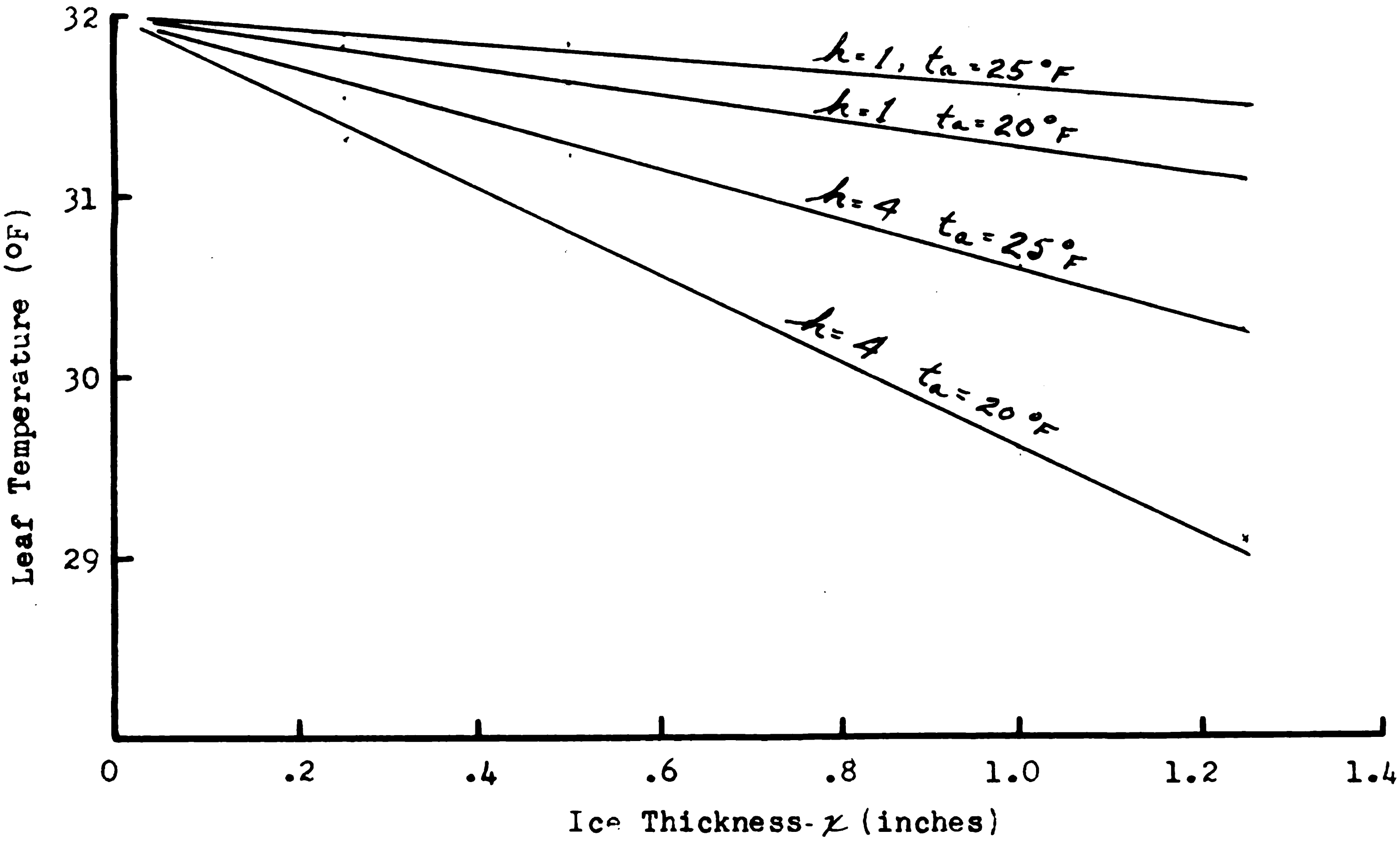

Fig. 27. Leaf temperature versus ice thickness for various air temperatures and film coefficients.

Discussion

It would be naive to accept the presentation in the previous section as a realistic representation of the actual conditions that occur in a particular culture of our agricultural crops. The ice shapes which occur during sprinkling for frost protection represent combinations of the shapes discussed. Also the exact behavior of the wind in a culture cannot be represented as simply as was shown in the previous section. There is always some water which runs off the leaf or bud and the radiant heat load on one bud might be quite different from that of another.

The previous section was rather intended as a guide to show the magnitude and trends of various effects. It is hoped that from information such as this, research under actual field conditions or simulated laboratory conditions can be guided in the proper direction.

Obvious trends which can be shown by the previous section are effects of the various shapes, the wind, and the various sizes within shapes.

The overall effect of increasing the size of any one shape was to decrease the amount of water needed for frost protection for a given air temperature and wind speed.

Spheres and cylinders were quite similar from a heat balance viewpoint when the wind is perpendicular to the cylinder. If the wind is parallel to a cylinder, it acts like a flat plate as far as its film coefficient is concerned. A flat plate with airflow parallel to it definitely has the lowest convection loss of any of the shapes studied. However, with airflow perpendicular to a flat plate, its convection loss is similar to the other two shapes. One effect which is inherent in the flat plate analogy is the temperature drop through the ice thickness due to the heat loss from the under side of the plate under steady state conditions. What possible effect this has in the field is discussed later.

From Figures 11, 13, 15, 17, 18, and 20 in the previous section, the effect of wind is very apparent. Increasing the wind, increases the amount of water required for protection to a given air temperature.

Evaporation was assumed to occur from the entire area of the sphere and cylinder and from the top side of the flat plate. However, in the limiting case of frost protection, the surfaces may not be wet the entire period. This would lead to evaporation losses which might be higher than actual. Formula (7) used for the evaporation loss gives results higher than some other formulas which would also seem to apply. It is felt more review is needed to discover a good relationship for the evaporation loss where the heat for evaporation does not come from the flowing fluid but comes from the heat of fusion during the change of phase. The evaporation loss, although it does not represent the greatest heat load on the plant, needs more study. It is felt the evaporation might be important at the beginning of sprinkling when the leaves or blossoms are first wetted.

Although the Figures 11 through 23 in the previous section were developed for certain assumed radiation conditions, their utility may be increased. Since the radiation is independent of the air temperature, its effect shows up as an additive constant for each curve drawn. If one wants to modify the radiation effect, all one needs to do is to draw another curve parallel to the original one, but shifted to the left or right depending upon how the radiation effect is to be changed.

There are, of course, many things the theoretical section did not indicate. The previous section only hinted at one possible effect of ice thickness. Nothing was said concerning the effect of spraying frequency or the effect of increased application rates.

The experimental work indicates there is a difference in the degree of protection afforded between twenty seconds frequency and two minutes frequency of water application. Data were taken at one minute but it can not be said from the curves of Figure 9 that there was any difference between one minute and two minutes. It is believed if there were any differences between one minute and two minutes, the experimental accuracy is such that these differences could not be readily detected. This can be understood because the differences found between twenty seconds and two minutes were only in the neighborhood of 2 - 3° F. This would lead one to believe the difference between one minute and two minutes is in the neighborhood of 1° F. To detect differences of this magnitude would take a very accurate experimental set-up and a very good, consistent procedure.

The Effect of Spraying Frequency

Theoretically there can be no difference between twenty second spraying frequency and two minute spraying frequency assuming 100 per cent efficiency of the water (i.e. all the water which is applied is available as sensible and latent heat) when the average temperature of the body is used as a criterion for frost protection. The same amount of heat is available in either case. Longer spraying intervals should not increase the heat load on the plant.

What then accounts for the difference which this investigator as well as others have found in favor of shorter spraying intervals? Perhaps it is the effect of greater temperature fluctuations at the longer frequencies which have an effect. Actually in the opinion of the author, the effect which is showing up is a decrease in the efficiency of water applied at the longer frequencies. When the water is applied it does not freeze immediately upon striking the surface but remains as a water film until sufficient heat has been removed to freeze it. Although the instantaneous application rate is the same for various frequencies of sprinkling in the field and was the same in the experimental freezing chamber, the volume of water applied per sprinkler pass is greater at longer spraying frequencies. This is simply because the spraying time is longer. This might have an effect even on a horizontal leaf because the internal forces (surface tension) of the water are only able to support a water film of a certain thickness before run-off occurs. Just how thick a water film can be supported depends on the shape of the surface and in the case of a leaf, depends upon its angle with the horizontal. This would indicate that applying water in thin films is the most efficient method. Whether there is a definite breaking point between twenty seconds and two minutes or whether the efficiency relationship is linear between the two frequencies is not known. This phase of the problem needs more study. Perhaps the efficiency could better be approached by evaluating the ice build-up for constant conditions of air temperature, radiation and application with varying application frequencies.

It is felt the condition in the freezing chamber where a horizontal leaf was sprayed with water would tend to minimize any difference in efficiency between two minutes and twenty seconds spraying frequencies. The author would expect greater differences if a bud or blossom were used as the plant medium.

Effect of Increasing the Application Rate

An interesting effect can be noticed from the experimental curves of Figure 9. At the higher application rates the curve tends to flatten out. This effect is the same as was noticed by Rogers (14) in his laboratory work. If the efficiency of sprinkling was the same at the higher application rates, one would expect the curves to steepen at the lower temperatures due to the decrease in wind speed. This is not the case. This again indicates a decrease in efficiency at the higher application rates. It is again a case of more water being applied than can be retained in the form of a film on the surface.

When the application rate is increased in the field, normally the instantaneous rate is increased. This was not the case in the freezing chamber where the application rate was increased by lengthening the "on period" of the spray water. This would indicate that in the field the effect of decreasing the efficiency as the application rate increased would be more pronounced than it was in the freezing chamber.

At twenty seconds frequency, the slope of the experimental curve was fairly steep until an application rate of about .09 in/hr was reached. After this point the curve began to level off. This might indicate a rapid decrease in efficiency after this point was reached.

Effect of Increasing Ice Thickness

This effect was not studied experimentally in this investigation although it could be responsible for some of the variability noticed. In the previous section, the effect of increasing the ice thickness for a flat plate during one dimensional steady state heat transfer is shown. This alone, however, can not be used to judge the effect of ice thickness on leaves in the field. The effect was considered for a constant film coefficient and air temperature. In the field, perhaps the increased ice load as it forms over the edge of the leaves gives a sheltering effect which might decrease the film coefficient on the bottom side. Also there is undoubtedly some effect of the increased mass of ice tending to decrease the temperature fluctuations when all the water freezes some time before the next application is made. In the authors opinion the effect of increasing the mass of ice whether it be on a bud or leaf is beneficial except for the increase in weight. However, the steady state effect discussed before is important to consider especially in an experimental investigation. It must be remembered that this steady state effect is important only in a flat plate analogy.

One possible effect of increasing the ice thickness on a bud or blossom would be to decrease the film coefficient as shown in the previous section.

Nieman (10) shows cooling curves for various shapes with different masses which offer part of the answer to the question of the effect of an increased mass of ice. He showed for any of the shapes, the larger the mass of ice, the less the temperature decrease for a given film coefficient and time. He further shows that for a constant mass of ice, spheres cool more quickly than the other shapes.

The Effect of Wind Speed

From the experimental results represented by Figure 9, no effect of the wind speed can be shown because the wind was fluctuating and depended to a certain degree upon the required air temperature. However, the experiments which were run with a steady wind of 1.25 miles per hour parallel to the leaf, showed striking differences between the edge and center temperatures especially during the early phases of a test. Consequently, the leaves showed damage along the entire leading edge. The tests run at lower wind speeds did not show this obvious damage along the leading edge of the leaf.

According to the theory of convective heat transfer from a flat plate, the film coefficient at the leading edge is infinite. The film coefficient then rapidly decreases as one proceeds from the edge. This would certainly help explain the obvious occurrence of the leading edge damage of the leaves. In a spherical or cylindrical body this effect does not occur. One can add to this the effect of two dimensional heat transfer from an actual leaf as it effects the heat loss from the edge.

The above effects seem to show up in the field during moderate frosts as leaf fringe damage is much more prevalent than complete damage or center damage.

The most obvious effect of wind is the increasing of the film coefficient as was pointed out in the discussion of the previous section.

Method of Procedure

A great deal more information is needed concerning minimum allowable plant temperatures, the allowable duration of these minimums, the effect of supercooling and whether some particular parts of a culture are more susceptible to frost damage than are others.

Because there is a high variability in the susceptibility of different plants of even the same species, the appearance of damage by itself is not a good criterion of frost protection from a laboratory experimental standpoint.

It was attempted in this study to determine a minimum safe plant temperature and then base the degree of frost protection obtainable on this minimum temperature. Although the appearance of frost damage was not used as the criterion for frost damage, it is considered important to use plant tissue rather than something artificial because of the effect of the roughness on the underside of the leaf on the film coefficient. Because of this roughness, a smooth flat plate analogy does not strictly apply to a leaf.

During tests using the first method of procedure where the plant temperature was not allowed to go below 31^{o} F for protection, there was very little if any frost damage noted on any of the leaves after a critical test. By a critical test it is meant one which was run at the minimum air temperature for frost protection.

Although it is reassuring to conduct tests by means of the first method of procedure where frost damage as well as temperature can be used as indication of minimum allowable air temperature, the author feels the third method of procedure is the most satisfactory from an experimental viewpoint. The third method obviates the difficulty of the large time lag due to the excess precipitation which accumulates before the air temperature reaches its critical value. The third method often damages the plant if the initial air temperature is too low as it frequently is. However, the response appears to be faster and the tests can be run more quickly. This also allows the data to be taken before the increase in ice build-up can have much effect.

Although the second method might not give a realistic answer to the minimum air temperature for frost protection under specified conditions, it is perhaps the best method there is of detecting differences between spraying frequencies, shapes, sizes, application rates, and wind speeds. The nice part about this procedure is that it is extremely simple, requiring perhaps the least elaborate control and instrumentation. Also the tests require the least amount of time.

If there really is a difference in the efficiency of the water applied, it should show up using this method.

An absolute figure for the degree of frost protection can not be arrived at by this method, but the differences between the various conditions studied can be very nicely brought out.

It became clear as the tests were run that the mass of ice had some effect on the average plant temperature. Whether this was due to the increased surface area provided or the decreasing of the film coefficient on the underside of the leaf is not known. It does point out, however, that this variable should as nearly as possible be eliminated in future laboratory tests.

CONCLUSIONS

1. In the range of precipitation of six to eleven hundreths of an inch per hour, twenty second spraying frequency gives frost protection to an air temperature about two degrees Fahrenheit lower than does two minutes spraying frequency.

2. At the twenty second spraying frequency, the increase in protection between six hundreths of an inch per hour and one tenth of an inch per hour was greater than the increase in protection between one tenth and fourteen hundreths of an inch per hour.

3. Theoretically the larger the shape (sphere, flat plate, cylinder), the lower the amount of water needed for frost protection at a given air temperature and wind speed.

4. The higher the wind speed, the more water required for frost protection, other conditions being constant.

5. Spheres require higher application rate for frost protection than flat plates and cylinders of the same length or diameter when the wind direction is parallel to the flat plate or cylinder.

6. With wind direction perpendicular to a cylinder about the same application rate for protection is required as for spheres of similar diameter.

7. A higher application rate is theoretically required for flat plates for frost protection when the wind direction is perpendicular to the plate than when it is parallel to

the plate.

8. When the wind is blowing parallel to a leaf, fringe damage occurs before center damage because of the higher film coefficient and greater exposed area at the edges.

SUMMARY

An experimental apparatus for simulating radiation frost conditions was constructed. Bean leaves four to six weeks old were used as the plant tissue for the tests. From a series of tests of the leaves, a leaf temperature of 31° F was chosen as the minimum temperature for frost protection. Thermocouples were located on the upper side of the leaf for the measurement of the leaf temperature. The air temperature in the apparatus could be well controlled. The airflow was unsteady varying from one third to seven eights of a mile per hour. The average airflow used depended upon the minimum temperature desired.

The third procedure attempted was found to be the most satisfactory. The air temperature was quickly lowered to a level below that expected as a safe level for frost protection. The air temperature was then raised in increments until the plant temperature reached a steady value of 31° F. This air temperature was then taken as the minimum level for frost protection for the precipitation rate and spraying frequency being studied. Data was taken at twenty seconds, one minute, and two minutes spraying frequencies.

A theoretical study was made of the application rate necessary to maintain various shapes (flat plate, sphere, cylinder) at a temperature of 31.5° F. Curves were drawn of the theoretical application rate versus the air temperature

for various wind speeds, sizes of the shapes, and various directions of the wind with respect to the shape.

SUGGESTIONS FOR FURTHER STUDY

1. An evaluation of the efficiency of sprinkling by actually measuring the ice build-up for various combinations of application rates and repeat frequencies could be made.

2. An evaluation of the wind patterns and velocities in various cultures during a radiation frost could be made. This would also include the effect of typical wind changes during the frost.

3. A study of the effect of the increased mass of ice on the temperature fluctuations and the film coefficient.

4. More knowledge is needed about minimum allowable temperatures for various cultures at different stages of growth. This investigation should include the freezing point of the cell sap as well as the ability of the culture to supercool under various conditions.

5. Further study on the heat loss by evaporation, especially at the first wetting of the leaves or blossoms is needed.

6. More complete local negative radiation data are needed.

7. A controlled experimental study on the temperature of buds and blossoms under various conditions of wind, temperature, precipitation rate and frequency and radiation could be carried out.

8. A study of the precipitation and drop patterns from frost protection sprinklers is needed. It is especially

important to determine where the areas of low precipitation occur and what this precipitation is in relation to the average precipitation of the sprinklers.

9. There is a need for the development of a technique for determining the local precipitation rate in the field. This is of particular importance when field research is carried out on tree crops where the local precipitation rate can be far different from the average precipitation rate as normally determined.

10. Actual field studies of frost control where the local precipitation rate, wind speed, air temperature in and away from the sprinkled area, plant temperature, and negative radiation measurements are made.

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APPENDIX A

Freezing Chamber Design Calculations

For the dimensions of the freezing chamber, the shape factor from floor to ceiling assuming non-conducting but reradiating walls is from Geidt (4):

$$F_{1R2} = 0.5$$

Since the floor and ceiling are not black body radiators, this shape factor must be modified by the following formula from Geidt (4):

$$F_{(1R2)Gr} = \frac{1}{\frac{1}{F_{1R2}} + \left(\frac{1}{e_1} - 1\right) + \frac{A_1}{A_2}\left(\frac{1}{e_2} - 1\right)}$$

where:

$F_{(1R2)Gr}$ = shape factor from floor to ceiling assuming grey body conditions and non-conducting but reradiating walls.

F_{1R2} = shape factor from floor to ceiling assuming black body conditions and non-conducting but reradiating walls = 0.50

e_1 = emissivity of floor surface (wet, dark soil, e = 0.85)

e_2 = emissivity of ceiling surface (frost, e = 0.90)

A_1 = area of floor

A_2 = area of ceiling.

Substituting:

$$F_{(1R2)Gr} = 0.44$$

For a net heat transfer of 25 BTU/hr ft^2 from the soil at

32^{o} F to the refrigerated ceiling, the ceiling temperature is computed as follows:

$$25 = (0.44)(0.174)\left[\left(\frac{492}{100}\right)^4 - \left(\frac{T_c}{100}\right)^4\right]$$

where:

T_c = ceiling surface temperature (^{o} Rankine)

solving:

$$T_c = 400\,^{\circ}R = -60\,^{\circ}F$$