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THE USE OF WARM WATER FOR HEATING AND COOLING PLANT AND ANIMAL ENCLOSURES

S. E. Beall and G. Samuels

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THE USE OF WARM WATER FOR HEATING AND COOLING PLANT AND ANIMAL ENCLOSURES

S. E. Beall and G. Samuels
Reactor Division

JUNE 1971

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FOREWORD

The work reported herein is an accumulation of several small related efforts over the past two years, some in AEC programs, some supported by the Department of Housing and Urban Development but mostly the result of the personal efforts of the authors in preparing an entry for the 1970 contest of the Swedish Association of Engineers and Architects "Energy at a Bargain Price."

THE USE OF WARM WATER FOR HEATING AND COOLING PLANT AND ANIMAL ENCLOSURES

S. E. Beall and G. Samuels

ABSTRACT

A method of utilizing the warm water from the condenser of a power plant to heat and cool greenhouses and animal enclosures is described. Most of the heat is exchanged by direct contact between air and the condenser cooling water in fibrous pads with the remainder being transferred to the air from finned tubes for humidity control. As this method was also used to cool the enclosures in the summer, the system can be used as the power plant heat rejection system.

Using the Fort St. Vrain Nuclear Generation Station (which uses cooling towers to reject 500-Mw heat) of the Colorado Public Service Company of Colorado as a site for study, it was found that the cost of the equipment required to utilize the warm water was in the range of the cost of heating systems for conventional greenhouses. Since the cost of heating greenhouses in the Denver area is over $5,000 per acre per year, the potential value of the heat being wasted from the plant is greater than $1,000,000 per year.

Keywords: Greenhouses
Animal Enclosures
Waste Heat
Controlled Environment

INTRODUCTION

The growing concern over thermal pollution has attracted attention to the importance of making some use of the heat in water being discarded by industrial processes, especially steam-electric generating stations. Since 1968, the Oak Ridge National Laboratory has studied[1] several possible uses (including municipal, industrial, agricultural and aquacultural) for warm waters. This document reports the results of the agricultural applications for controlled environment structures.

It is a happy coincidence that the temperatures of many industrial effluent waters are in the range which most plants and animals find comfortable, that is, 60 to 90°F. In exploring ways to take advantage of this coincidence, we have developed a conceptual design of a heating-cooling system which makes use of warm water to control temperatures in greenhouses and animal enclosures. In this design the greenhouses, etc., perform as cooling towers to discard some or all of the heat from the power station or other heat producer, while maintaining the desired temperature within the house. Although this study examines the feasibility of using all of the reject heat from a medium-sized power station, we recognize that few prospective users could consider large complexes. However, whatever fraction of the heat can be used in such a fashion conserves total energy resources, offers an attractive saving to the grower, and could be profitable to the utility.

The practice of growing fruits and vegetables in heated greenhouses has been practiced since Roman times. Pliny[2] recorded the cultivation of cucumbers for Emperor Tiberius. Commercial greenhouse flower culture long has been common throughout the world, but only in recent years have vegetables been cultured indoors on a commercial scale. Undoubtedly, the rising cost of labor and transportation and the growing scarcity of suitable land have contributed to the rising prices of vegetables and the improved profit outlook for indoor culture. Commercial ventures of considerable size have been successful in Sweden, Holland, Rumania and Iceland. In the United States the largest area under covered cultivation is in the vicinity of Cleveland, Ohio, where approximately 700 acres are in use for flower and vegetable production.

The production of broilers in enclosures is a very large industry in the United States, but is nearly confined to the mild sections of the country where heating costs are minimum. Egg production is better distributed and, although not as sensitive to climatic conditions, requires sizeable quantities of heat for maximum production efficiency. Few larger animals are grown in enclosures although swine are provided with shelter in cold areas.

Enclosed production offers several opportunities for higher yields, better use of land, automation to reduce labor, and animal waste pollution abatement. Figure 1 shows how the growth of several animals can be improved by better environmental control, and Table 1 indicates how careful attention to temperature can improve the conversion of feed to flesh. Plants also produce best when optimum temperatures are maintained, as indicated in Fig. 2.

The cost of maintaining the desired temperatures in winter and summer is a principal element in the product food cost. Because of this expense, many warm-weather crops are not grown in the winter and many cool-weather crops cannot be grown profitably during the hot months. A one-acre glasshouse installation maintained at 65°F minimum temperature for a winter, with heat costing $1.50/million Btu, might have a total heating cost as high as $8,000 to $10,000; poultry heating costs of more than 1¢/bird are not unusual for cold climates. Presumably, an operation which broke even financially with this expense could show a profit at lower heat cost. Much lower heat costs and more profitable operations are possible if the thermal discharge of power plants can be made available to the large food growers. Figure 3 shows our estimate of the incremental cost of low-temperature heat from modern power stations — nuclear and coal. The calculation[12] assumes that the heat in the normal condenser outlet water is available at no cost. At any higher temperature, the cost is based on the reduction in plant efficiency, using a 14% fixed charge rate for the increased capital cost required to maintain a fixed electrical output and an additional fuel charge of 16¢/10^6 Btu for nuclear fuel or 30¢/10^6 Btu for coal.

We have explored a conceptual design of a large controlled environment installation which might replace the heat disposal equipment normally installed in a medium-sized power station. The plant chosen for study is the

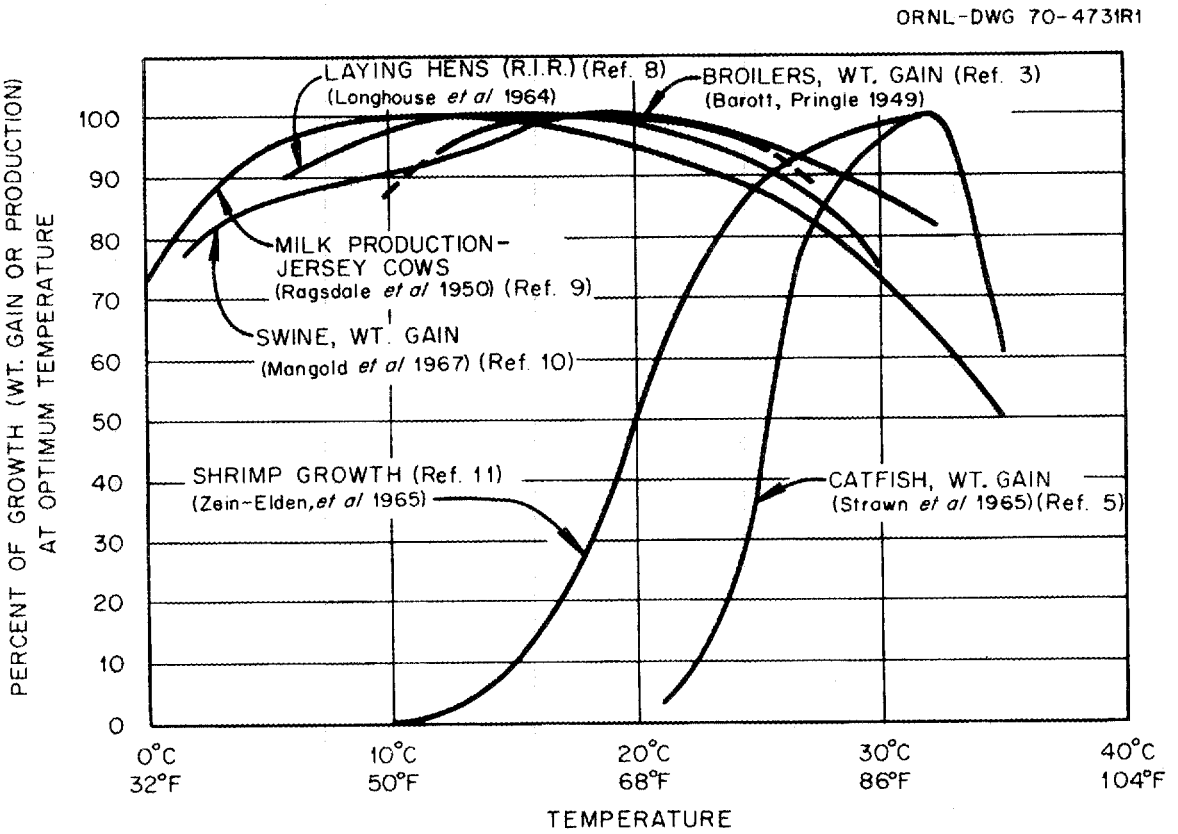

Fig. 1. Effect of Temperature on Growth or Production of Food Animals.

Table 1. Effect of Temperature on Feed Conversion

		Temp. (°C)	Lb Feed/Lb Weight Gain	Feed Efficiency (%)
Chickens[3, 4]		24	2.7	39
		30	2.1	48
		34	1.9	52
Catfish[5]		20	4.0	25
		29	1.3	75
		32	1.5	65
Hogs[6]				
	Individual Hog	3	5.9	17
	Group	3	4.3	23
	Group	8	3.8	27
	Group	15,23	3.4	29
Milk Cows[7]	(Jersey)	-13	2.1	48
		10	0.65	128
		38	0.55	180

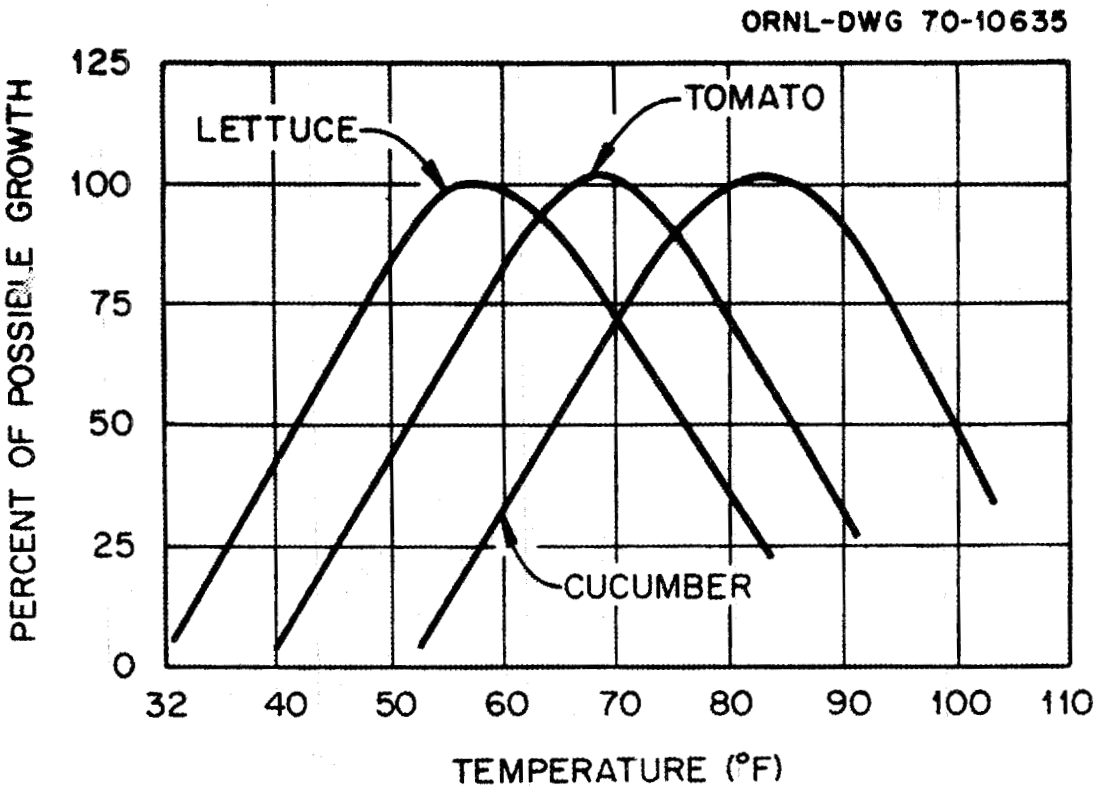

Fig. 2. Idealized Growth Curves for Several Crops (Ontario Department of Agriculture and Food - Publication 526).

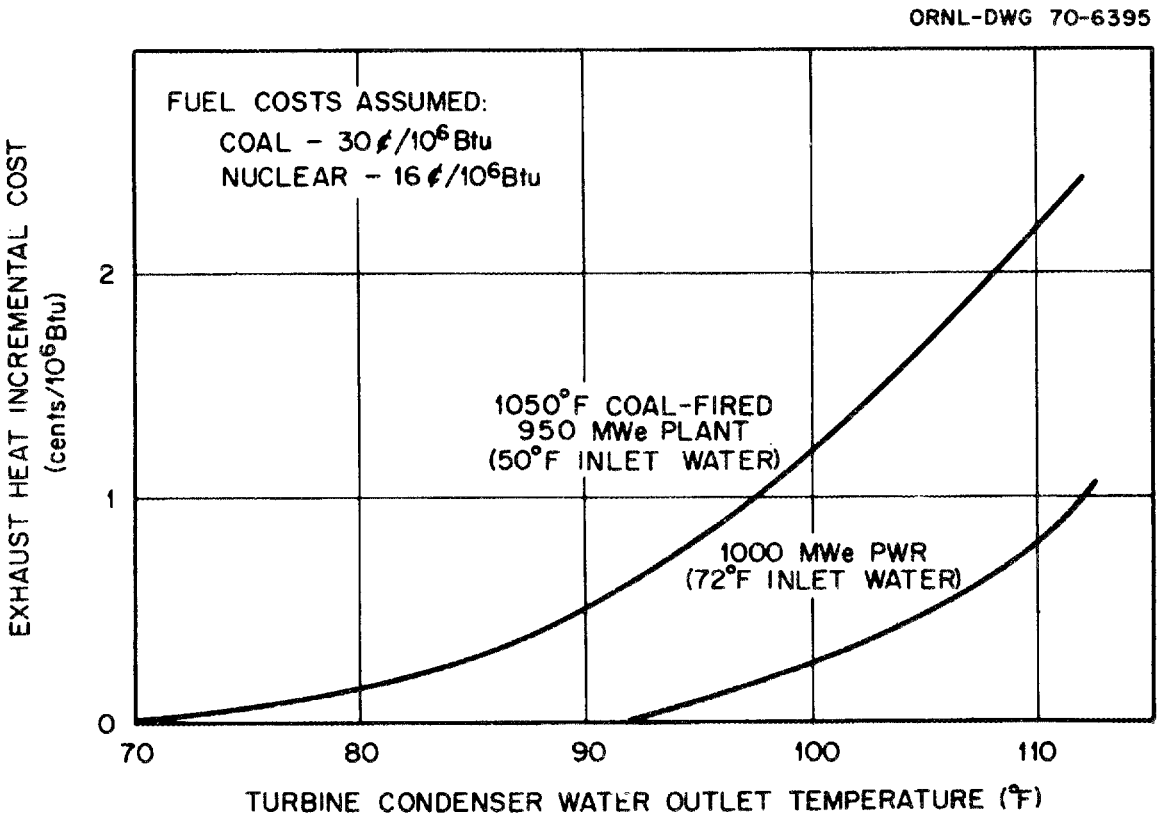

Fig. 3. Incremental Cost of Warm Water Heat.

330-Mwe gas-cooled nuclear station being constructed at Fort St. Vrain, Colorado, to be operated by the Public Service Company of Colorado. Admittedly, we chose this plant for study because of several favorable factors, but we believe there are many similar sites which would be equally attractive. Fort St. Vrain is located about 35 miles north of Denver, Colorado. The site of approximately 1200 acres is in a fertile agricultural area at which water is in short supply. As at most reactor plant sites, the land is needed for an exclusion area and would not normally be used except on a lease arrangement for grazing or open-field agriculture. The climate is cold enough to limit the outdoor cultivation of most flowers and vegetables to the summer months. Very few broilers are grown in Colorado but egg production is practiced extensively. The air is dry, with a low wet-bulb temperature, making it possible to cool 100°F water to below 75°F by evaporation in the hot months. Winter temperatures in the Denver area are low, and poorly insulated structures such as greenhouses or poultry houses require large quantities of heat.

The population of Denver is about 1.1 million people and their fresh vegetable and poultry requirements are large. The U.S.D.A. records for vegetable shipments to Denver in 1968 are shown in Table 2. The large quantities of fresh vegetables needed by the Denver area, if they could be supplied from local greenhouses, would require at least 600 acres operated throughout the year. The broiler and egg needs of the city are also large: based on the average consumption in the United States, the Denver population requires 22 million broilers and 400 million eggs per year. The poultry house area for this level of production would be about 100 acres.

The cost of shipping this produce from western and southern states is several cents per pound. This cost, of course, puts local producers in an advantageous position. Even at a 2¢/lb freight charge, a crop which yielded 200,000 lb/acre would give the grower a $4,000/acre advantage. Broiler production rates are greater than 1/2 million pounds per acre, making a saving of $10,000 or more per acre appear possible in shipping costs alone.

Table 2. Denver, Colorado (Metro Population, ~1,100,000)

Fresh Vegetable Unloadings - 1968*

Vegetable	Unit Wt. (lb)	No. Units Per Car	Total Wt. Per Car (lb)	No. Cars	Total Wt. (lb)	Assum. Yield Acre-Year	Estimated Greenhouse Area Req'd. (Acres)
Cucumbers	50	650	32,500	196	6.5×10^6	518,000	12
Eggplant	32	775	24,800	48	1.2×10^6	67,500	18
Lettuce (leaf)	26	850	22,000	2,250	50×10^6	756,000 hd. (190,000 lb)	260
Green Onions	60	350	21,000	273	5.7×10^6	400,000 lb	14
Peppers	30	775	23,250	188	4.4×10^6	75,000 lb	58
Radishes	11-1/4	2,800	31,500 (84,000 bu.)	329	10.4×10^6 (27.5×10^3 bu.)	2.1×10^6 lb (5.7×10^6 bu.)	5
Squash	45	750	33,750	216	7.3×10^6	80,000	90
Tomatoes	50	720	36,000	707	25.5×10^6	230,000	110
Strawberries	10	1,400	14,000	225	3.2×10^6	80,000	40
				4.432	113.7×10^6		607

*U. S. Department of Agriculture, "The Yearbook of Agriculture, 1968."

THE FORT ST. VRAIN POWER PLANT

The high-temperature gas-cooled reactor (HTGR) plant at Fort St. Vrain was designed for the Public Service Company of Colorado by the Gulf General Atomic Company.[13] It is to operate at 39.5% thermal efficiency and is equipped to dispose of 1.7×10^8 Btu/hr by means of a forced-draft cooling tower. Under maximum power conditions, the tower receives 102°F water from the turbine condenser at a rate of 155,000 gpm, cools it, and returns it to the condenser at 80°F. The contract cost for the tower is about $1,500,000 and operating and maintenance costs are estimated to be $50,000 per year.

In essence, we propose that this cooling tower could be replaced at roughly similar costs by installing many smaller coolers in a very large greenhouse (and/or poultry house) range built close to the power plant on land leased from the utility. (There are no plans to build such an installation.) We will show that the greenhouses could be cooled and heated with the hot condenser outlet water and that the water would be returned at temperatures about 10°F lower than the proposed cooling tower design temperatures. Our design is based on a humid heat exchange system but provides for the addition of some dry heat to reduce humidities, if necessary, to protect against diseases of fungal growths and other problems associated with high humidities. Although the discussion henceforth relates to greenhouses, we believe the concept is feasible for other controlled environment applications in agriculture.

COOLING AND HEATING DESIGN REQUIREMENTS OF GLASSHOUSES

The type of greenhouse we have chosen for the heat-transfer study is the rigid, glass-covered, "Clear Span" type manufactured by the National Greenhouse Company of Pana, Illinois. Each house is approximately 50 feet wide by 100 feet long, arranged with common sides so that 10 units form one structure 500 feet wide. Each 50 x 100 ft unit is equipped with evaporative cooling pads (to be described later) across one end (50-ft width) and with exhaust fans at the opposite end. We allow for sufficient finned tube surface for the addition of dry heat to reduce relative humidities to the 75 to 80% level. The arrangement of houses, roads, and piping is shown in

Fig. 4. Note that the cooling-heating pad ends are located "back-to-back" to provide an economical piping layout, and the fans exhaust into the 25-ft wide service road area.

The complete installation of greenhouse, poultry houses, and food-processing plant is laid out on the site plot in Fig. 5. Space is provided for a future fish culture area of 50 acres that would be fed by blowdown water from the cooling system. With several different, large-scale food-producing and processing units on the same site, there are several opportunities for synergism. Both animal wastes and processing wastes could be incorporated into fertilizer and feed to be recycled. A common management of the complex would be an obviously desirable choice. These and other features of the food complex should be the subject of further study.

Cooling and Heating Design Considerations

The properties of glass make greenhouses very efficient solar radiation heat traps. Glass transmits most of the incoming solar radiation (0.3 to 2 microns), but because the wave length of the radiation from the inside of the house is in the 3 to 50 micron range where glass is a good absorber, very little of the energy is reradiated from the house. Thin polyethylene and mylar plastic sheet and special grades of rigid Fiberglas have transmission characteristics similar to glass and are widely used as greenhouse covering materials.

The total energy absorbed by a glass surface depends on the latitude and altitude of the location, the time of year (solar declination), the time of day, the atmospheric conditions, and the orientation of the surface. For the Fort St. Vrain site (approximately 40 degrees north latitude), the maximum solar radiation occurs at noon (suntime) on June 21, and amounts to an estimated 295 $Btu/hr \cdot ft^2$. On December 21, when the solar altitude is a minimum, the heat load on a clear day will be about 140 $Btu/hr \cdot ft^2$. Only part of the incoming solar energy results in a sensible heat load on the cooling system. Approximately 50% of the energy evaporates water from the plants and ground and represents a latent heat load.[14] For cooling design purposes the maximum heat load on the greenhouses for the Fort St. Vrain

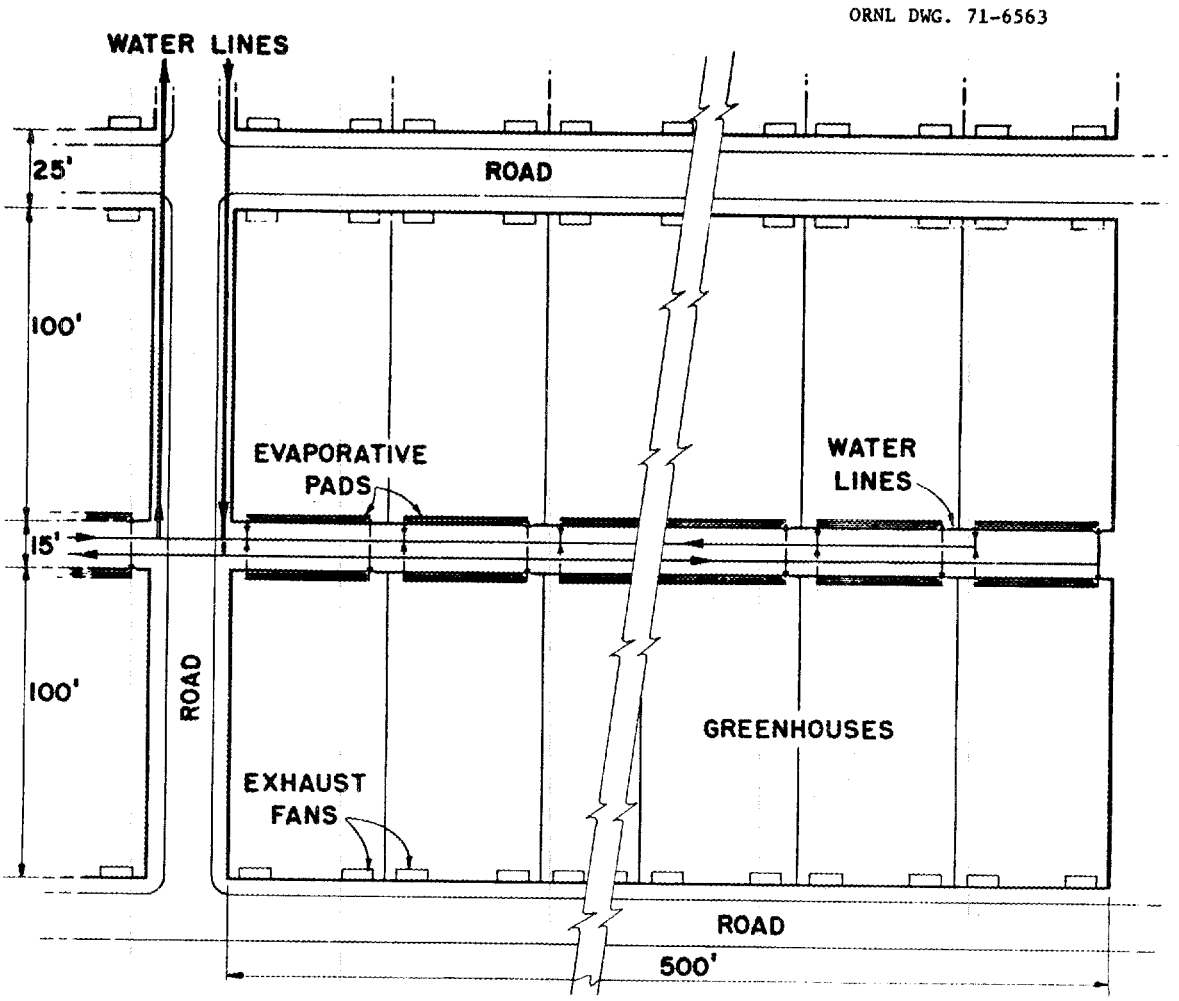

Fig. 4. Road and Piping Arrangement.

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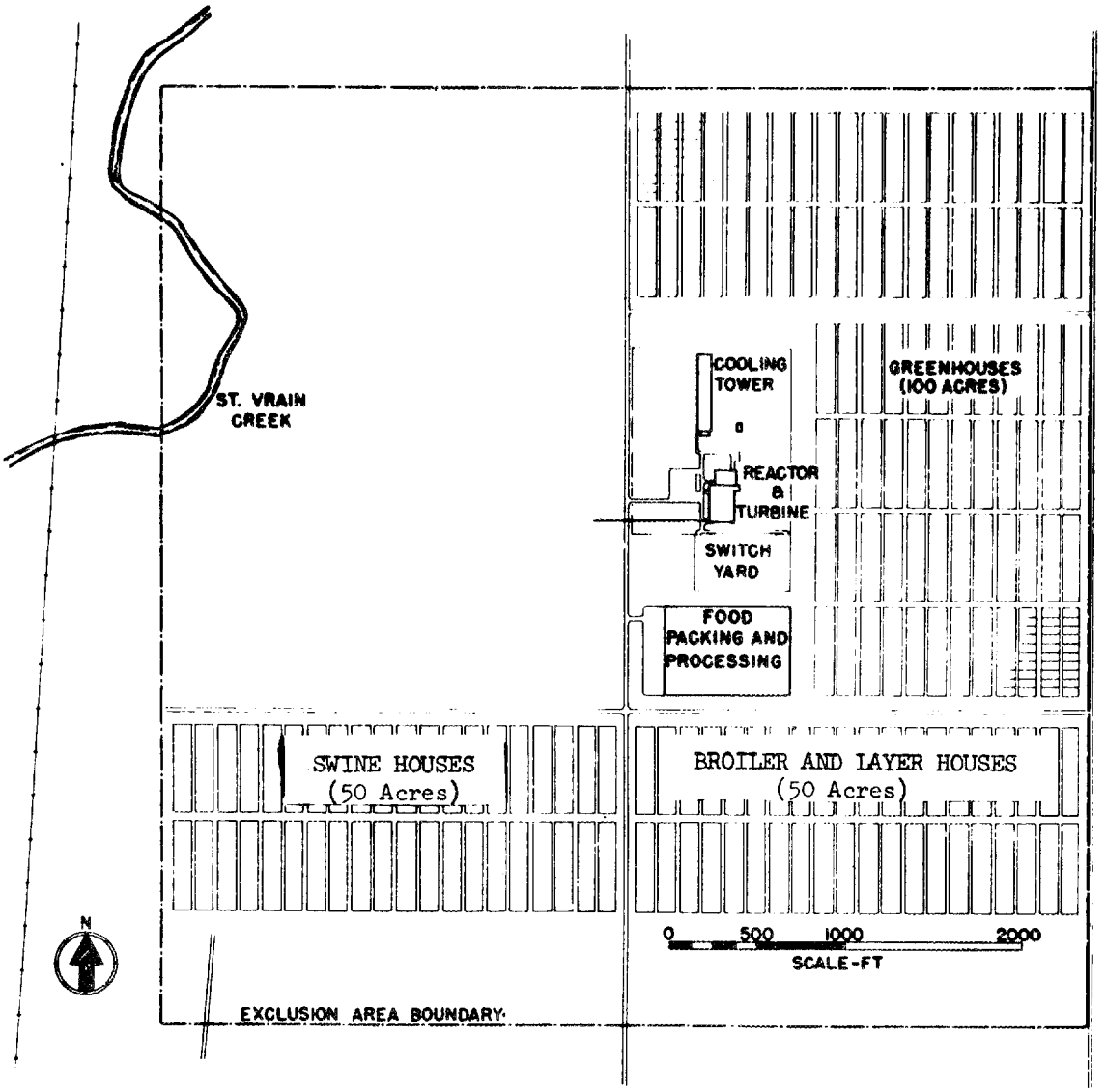

Fig. 5. Possible Layout of Food Production Complex at the Fort St. Vrain Site.

site has been taken as 150 Btu/hr·ft^2 sensible heat and 145 Btu/hr·ft^2 latent heat. The ambient air conditions were taken as 95°F dry bulb and 64°F wet bulb. It should be noted that combining the worst air design conditions with the peak solar load is rather conservative because the peak air temperature normally occurs after the sun has passed its maximum altitude.

The maximum heating requirements are determined by the outside air temperature, the wind velocity, and to some extent by the design of the greenhouse and the method of heating. The air and wind conditions assumed for design purposes are 0°F and 50% relative humidity and a wind velocity of 15 mph. It is also assumed that these conditions occur on a clear night with an effective sky temperature of -100°F. The greenhouses are assumed to be built in rows of ten 50-ft x 100-ft units; the heat loss through the side walls will be small compared to the roof losses. The heat loss from the roof as a function of the outside air temperature and the roof temperature is shown in Fig. 6. The heat-transfer coefficient for an air velocity of 15 mph is about 4 and the ratio of roof area to the horizontal area is assumed to be 1.08. Heat losses are also shown for the same conditions except that the air temperature has been reduced to -30°F. The total heat loss then depends on a heat balance between the radiation, convection, and condensation heat input to the roof and loss from the roof. This balance depends on the method of heating and the internal air recirculation with the houses and will be discussed later.

The Evaporative Pad System

Evaporative pad cooling systems are commonly used in greenhouses to reduce the temperature in the houses on hot, clear days. Under normal operating conditions in a dry climate, these systems reduce the temperatures by 10 to 20°F. The following discussion is concerned with the use of evaporative pads as both a cooling system for the house and a heat dump or cooling tower for the power plant. The evaporative system chosen for study is one manufactured (for greenhouses) by Acme Engineering and Manufacturing Company of Muskogee, Oklahoma. The Acme cooling pads are 2-in.-

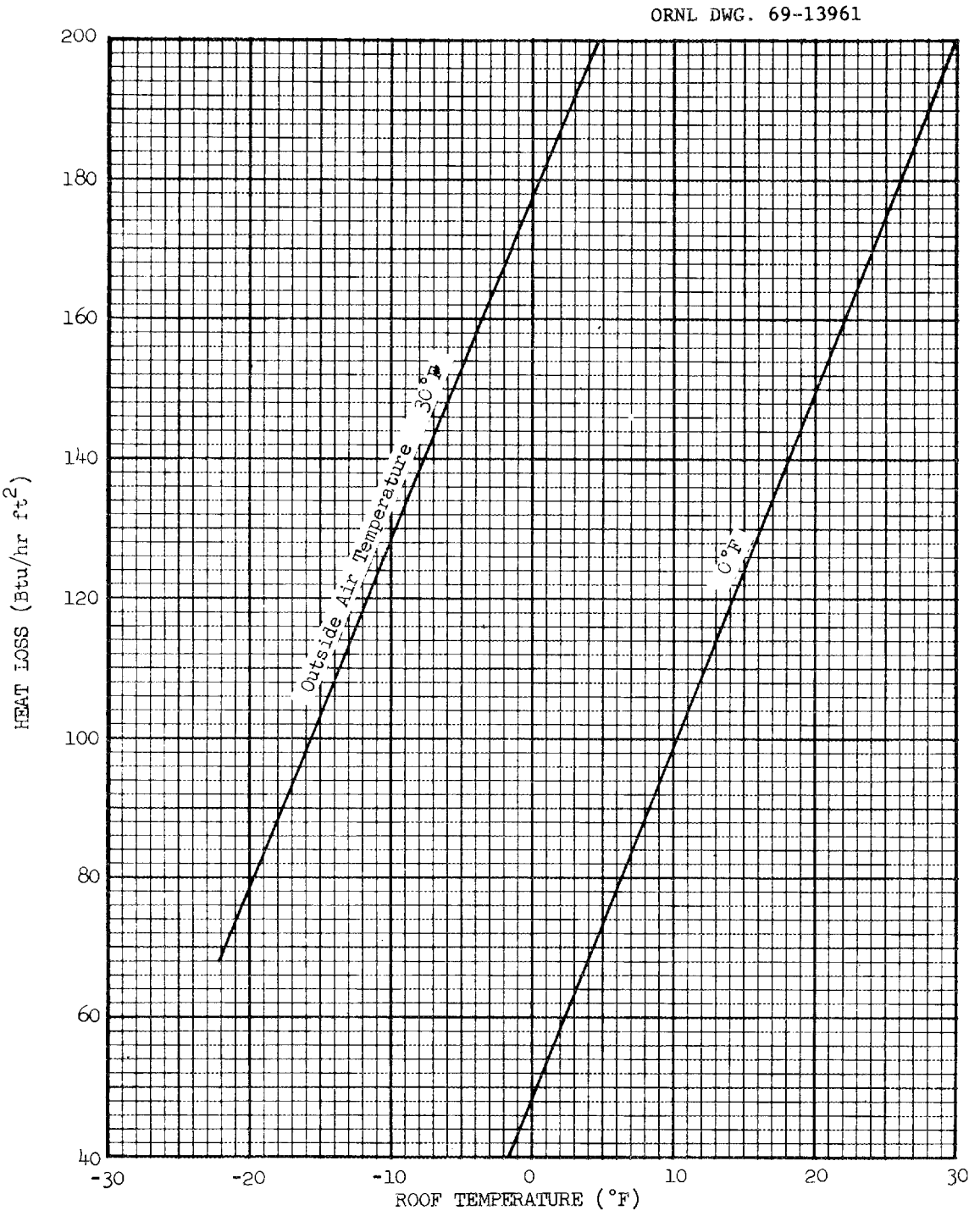

Fig. 6. Night Heat Loss per Ft^2 of Horizontal Area vs Roof Temperature.

thick wire frames filled with a semi-permanent fibrous material (see Fig. 7). Water flows onto the pad from a trough at the top and maintains a water film on the fibers throughout the pad. Air flows horizontally through the vertical pads and is cooled (or heated) depending on how much evaporation occurs and how much sensible heat transfer takes place. Our system is designed so that the water collected at the base of the pad always will be approximately 22°F cooler (for the Denver area) than the entering water, and the air leaving the pads will be regulated to the temperature required by the crop--generally in the 60 to 80°F range.

As noted earlier the peak solar heat input to the houses is about 150 Btu/hr·ft^2·sensible and 145 Btu/hr·ft^2·latent in the summer. For a basic house size of 50 ft x 100 ft, the total sensible heat load is 750,000 Btu/hr. Assuming a maximum temperature rise of 10°F through the house, the air flow rate is calculated to be about 73,000 cfm. The fans normally used in these systems supply a head of about 0.1 in. of water and will deliver 150 to 200 cfm/ft^2 of area of the evaporative pads. The required face area of the pads is then about 400 ft^2 and would be arranged along the 50-ft length and be 8-ft high. The air flow rate is 182 cfm/ft^2.

Although the pad manufacturer recommends a water rate only 1/3 to 1 gpm/ft for normal cooling installations, in our application the water rate is as high as 3.5 gpm in order to provide an adequate supply of cooled water for the power station condenser. We have tested the pads at flow rates as high as 5 gpm/linear ft. The power station operating condition also sets the water temperature requirements* 102°F to the pads and 80°F from the pads — a drop of 22°F.

In designing a greenhouse cooling-heating system, it is necessary to calculate the humidity and temperature of the conditioned air, given outside air characteristics and the desired water temperatures. Cooling tower calculational methods are concerned primarily with the water conditions and

*These requirements are set by the performance of the existing cooling tower but as will be seen further in this study, the evaporative pads are capable of cooling the same volume of water to a 70°F-90°F range, and offer a more efficient turbine summer operation to compensate for the higher inlet temperature and, therefore, somewhat lower turbine efficiency in the winter months.

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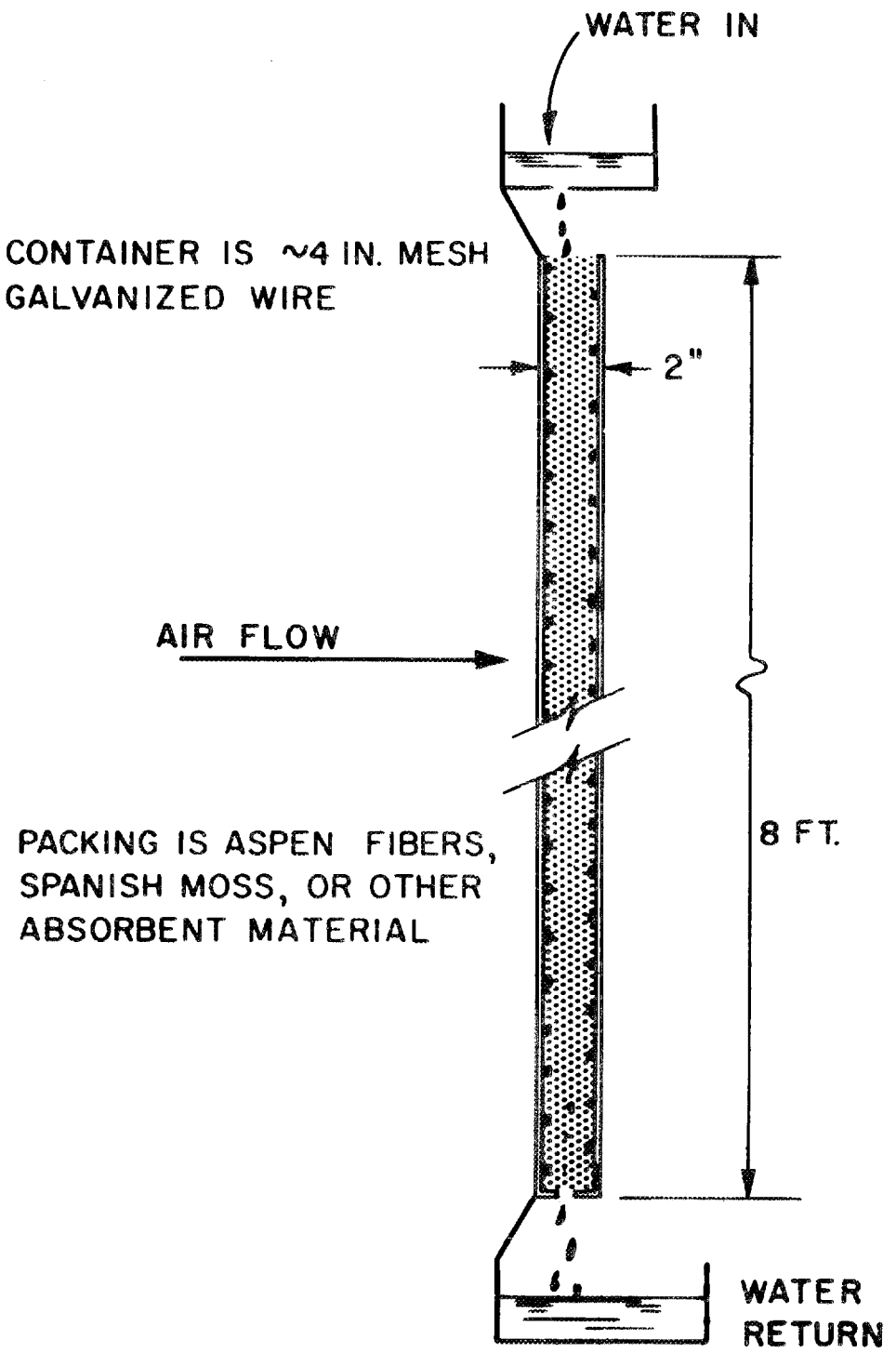

Fig. 7. Pad Assembly

not the condition of the air. Thus it was necessary to perform an analysis of the sensible and latent heat transfer with the pads and to develop calculation procedures for evaporative pad performance. The details of the analysis and the method of calculation are presented in the Appendix, and the results are discussed later in this report. But first it will be informative to describe how the system works.

A typical arrangement of heating-cooling pads and recirculation within a house is shown in Fig. 8. Note that a clear plastic sheet separates the attic from the growing area, and permits the attic to be used as a return air duct to recycle the circulating air. The percentage recycle depends on whether heating or cooling is required; recycle is accomplished by closing the vent louvers (see Fig. 8).

Warm weather conditions generally require maximum cooling at midday. For these conditions the vent louvers are fully open to permit discharging the maximum amount of heat to the outside. Air is pulled through the pads at one end of the house, flows over the plants, and is discharged at the opposite end.

A significant additional advantage of the direct water-air contacting is the cleaning effect on the air. Air contamination with dust, SO_2, and hydrocarbons is already a serious environmental problem and there have been many reports of plant damage resulting from high SO_2 and smog concentrations in the highly industrialized areas of the world. One of the best methods of removing these contaminants is by water washing, which is accomplished very efficiently in the proposed evaporative pad system.

Winter Operating Conditions

As the temperature decreases, the control system partially closes the vent louvers, effecting recirculation through the attic, and back to the pads to be mixed with the inflow of external air. At some low outside temperature, about -30°F, the vent louvers must be closed completely if the air temperature over the plants is to be kept at about 72°F. In this operating mode, all of the air is returned through the attic to the inlet end to be reheated in the pads before being directed back to the plants.

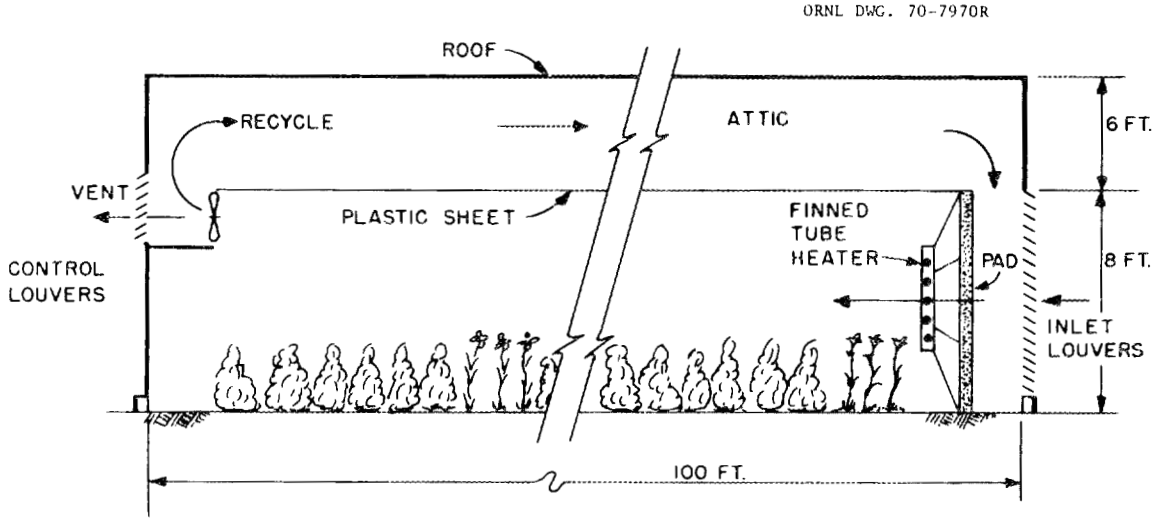

Fig. 8. Typical Greenhouse and Air Flow System.

The atmospheric conditions existing in the greenhouse for several winter operating conditions are shown in Table 3. The total heat discharged by the power plant is sufficient to heat 200 to 250 acres of greenhouses with ambient temperatures as low as -30°F. For 200 acres the water flow rate per greenhouse is 44,100 lb/hr. The first five cases shown in Table 3 are for 44,100 lb/hr and a total air flow rate of 153,000 lb/hr, which is one-half the maximum capacity of the fans. The recycle rate is the return air flow rate through the attic and the vent rate is the amount discharged at the fans (neglecting system leakage, this is also equal to the outside air brought into the pads). The last two cases in Table 3 are for the emergency condition in which the power plant is shut down and heat is being supplied by an auxiliary oil-fired boiler. In these cases the water rate would be reduced to 60% of the normal flow. With a 150-Mw auxiliary boiler, the house temperature could be held at 56°F with a 0°F outside temperature. As indicated by the other temperatures in the "air temperature over plants" column, a rather wide range of temperatures is available for various crop needs.

Table 3. Greenhouse Conditions for Winter Operation (200 Acres).

Outside Air Temp (°F)	Water Flow Rate (lb/hr)	Air Flow Rate		Air Temperature		Range of Water Temp (°F)	Mean Roof Temp (°F)
		Recycle (lb/hr)	Vent (lb/hr)	Over Plants (°F)	Through Attic (°F)		
-30	44,100	153,000	0	72	72-65	66-88	1
-15	44,100	153,000	0	76	76-69	71-93	15
0	44,100	153,000	0	80	80-74	75-97	26.5
0	44,100	148,400	4,600	72	72-65	66-88	21
0	44,100	141,400	11,600	63	63-56	56-78	15
0	26,500	153,000	0	56*	56-50	51-73	12
0	26,500	148,400	4,600	51*	51-44	45-67	8.5

*Emergency Power Cases.

We recognize that the very high nighttime humidities resulting from the recirculating mode of operation in winter or on cool summer days may

be objectionable because of the difficulty of controlling bacterial and fungal growth on plants. Although some plant physiologists argue that modern bactericides and fungicides are adequate protection for plants, others argue for lower humidities. The University of Arizona and the University of Sonora, Mexico, are cooperating in an experiment[15] to demonstrate that vegetables can be grown in a 100% humidity greenhouse environment. Their heat exchange system is similar to the one proposed here but makes use of a falling film of warm seawater on asbestos surfaces instead of the fiber pads and fresh water assumed in our study. Poultry researchers also recommend lower humidities to avoid bird stress and to maintain dryer droppings. For these reasons our design includes a bank of plate-coil dry heaters (10,300 ft^2 of heating surface) downstream of the evaporative pads so that the temperature of the saturated air from the pads can be increased a few degrees with a resultant reduction in relative humidity to the 75 to 80% range. In this mode of operation, the water from the power plant would first flow through the plate coils before emptying into the trough above the evaporative pads. One-fourth to one-half of the total heat would be transferred from the coils and the remainder from the warm water on the evaporative pads.

We find that the Fort St. Vrain reactor discharges sufficient heat for a 200-acre operation during -30°F weather. For cases in which less than 200 acres are heated, or for ambient temperature conditions greater than 0°F, the excess heat supplied to the greenhouses would be dumped by increasing the vent rate. This excess heat could be used to increase the house temperatures above those in Table 3 but would result in high return water temperatures to the power plant, which is not desirable. In any case, the water temperature returned to the power plant condenser will be about 5 to 15°F less than the temperature of the air passing over the plants.

All of the winter conditions listed in Table 3 are for clear nights and rather severe wind conditions (15 mph). For an overcast night sky, the radiation heat losses are lower because the effective sky temperature is closer to the ambient air temperature than the -100°F typical of a clear night. During the day the sky ceases to be a heat sink and becomes

a heat source and the conditions in the houses are much different. Even in late December, when the solar heat load is a minimum, the heat input varies from about 45 Btu/hr-ft^2 at 9:00 a.m. to 140 Btu/hr-ft^2 at noon and will require a large vent flow rate. The air temperature leaving the pads will be 70 to 75°F--5 to 7°F above the return water--and still near saturated if no dry heat is added through the finned-tube heater. However, as the air flows through the house, the solar heat input will increase the dry bulb temperature and reduce the relative humidity. For the noon heat load the temperature of the air will increase to 80 to 85°F and the relative humidity will drop to 70 to 80% without heat input from the finned tubes.

Summer Operating Conditions

The greenhouse and power plant condenser water temperatures available during summer operation are summarized in Table 4. Two ambient air conditions are shown. The 95°F dry bulb and 16% relative humidity (64°F wet-bulb, inlet air) case corresponds to summer design conditions for the Denver area. The second set of conditions, 50°F dry bulb and 73% relative humidity has the same humidity ratio (0.0056) as the first case but with the dry-bulb temperature reduced from 95°F to 50°F. Two water flow rates per greenhouse are also shown in Table 4. The 88,200 lb/hr rate corresponds to rejecting all the plant waste heat (~500 Mw) to 100 acres of greenhouse while the 44,100 lb/hr rate is equivalent to distributing it to 200 acres. Note that even for the 100-acre case, 89°F cooling water will maintain the house temperature below 86°F.

The maximum fan capacity is 182 cfm per ft^2 of pad area of 306,000 lb/hr per greenhouse section. For the full fan capacity, a water flow rate of 88,100 lb/hr and the summer design conditions, the average temperature of the air leaving the pads is about 76°F, the relative humidity is 80%, and the return water to the condenser is 67°F. As the air passes through the greenhouse, its temperature increases to 86°F and its relative humidity decreases to 67%. When the air temperature drops during the night, the energy exchange process in the pads will change and for cool nights will heat the air. Assuming that the humidity ratio remains constant at 0.0056 lb of water per lb of air and the air temperature drops to 50°F, the temperature

Table 4. Summary of Greenhouse Operating Conditions During Summer.

Ambient Conditions		Air Flow Rate	Water Flow Rate	Range of Conditions in Greenhouse		Range of Water Temp
Dry Bulb Temp (°F)	Relative Humidity (%)	(lb/hr)	(lb/hr)	Temp (%)	Relative Humidity (%)	(°F)
95	16	306,000	88,200	76-86	80-67	67-89
50	73	306,000	88,200	~58	~95*	51-73
50	73	153,000	88,200	~67	~100*	57-79
95	16	306,000	44,100	71-81	85-71	64-86
50	73	306,000	44,100	~53	~90*	47-69
50	73	153,000	44,100	~57	~100*	49-71

*These RH's can be reduced by adding dry heat as explained on page 21.

in the greenhouse will be about 58°F, and the relative humidity will rise to 95% unless warm water is fed to the finned-tube space heater. The third row of Table 4 shows the effect of reducing the air flow rate to one-half, and the last three rows show the conditions for one-half the flow rate to each greenhouse. The water and air temperatures and the relative humidity leaving the pads for the summer night case with a water rate of 44,100 lb/hr and fans at one-half full flow are shown in Fig. 9.

All of the cases discussed above are with only the evaporative pads operating. In order to reduce the relative humidity, either dry heat must be added or in some cases air must be bypassed around the pads. Calculations were made to demonstrate the manner in which the dry heat exchanger, which is in series with the pads, can be used to control the relative humidity. The heat exchanger was assumed to be the full width of the pad (50 ft) and 40 in. high. The heater is a finned-tube type unit with five 3-in. diam tubes expanded into 40-in.-long plate fins (72 fins per in.). The fins are 6-in. wide and .024 in. thick and mounted in the vertical position (see Fig. 8). The total outside surface of each unit is about 10,300 ft^2. For most cases of interest the fin efficiency is between 40 and 45% and the effective area is about 4500 ft^2.

Figures 10 and 11 show the effect of the heater on the conditions of the air passing over the plants. Both of these cases are for the summer

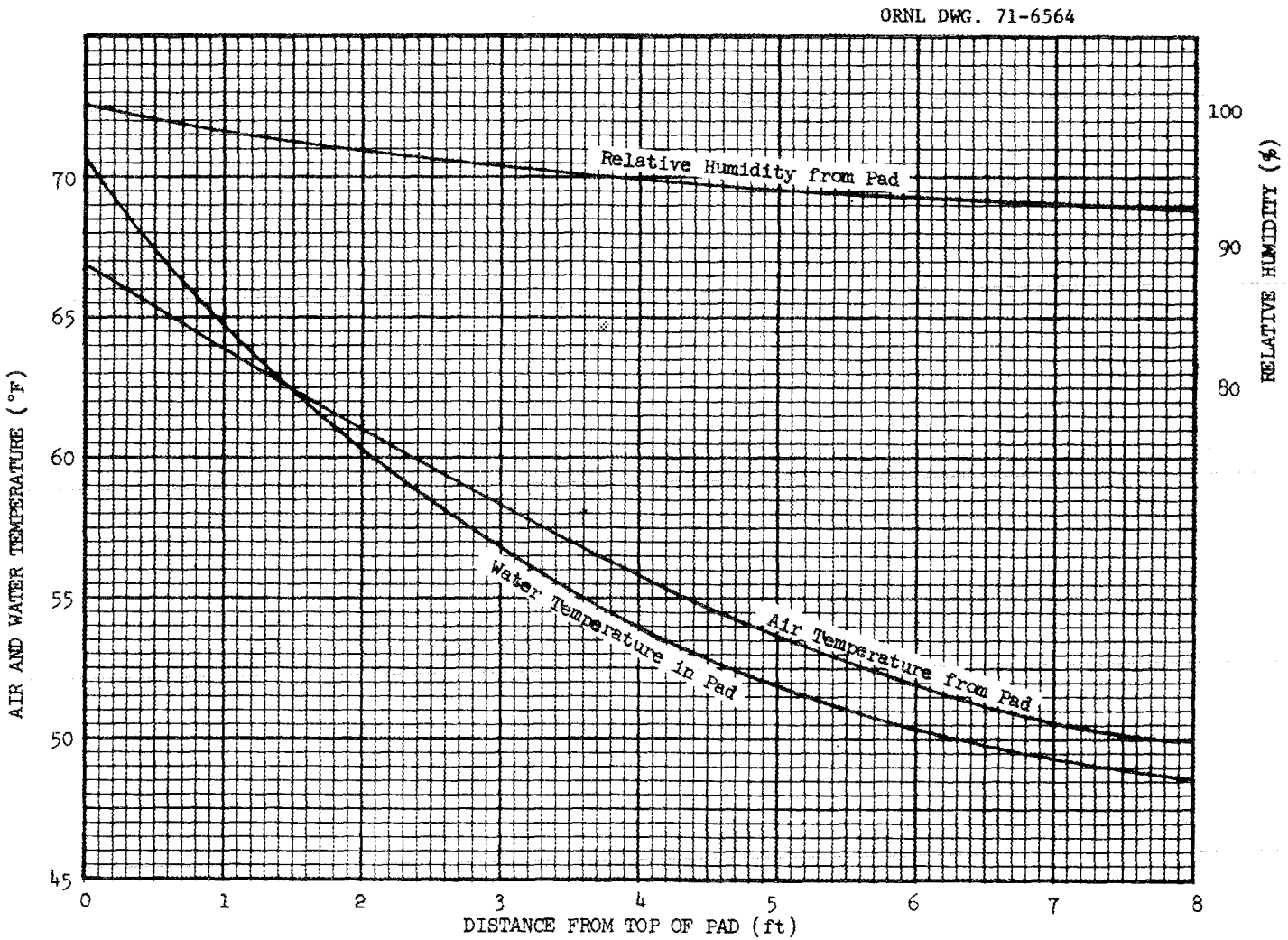

Fig. 9. Air and Water Conditions for Summer Night Operation with Evaporative Pads.

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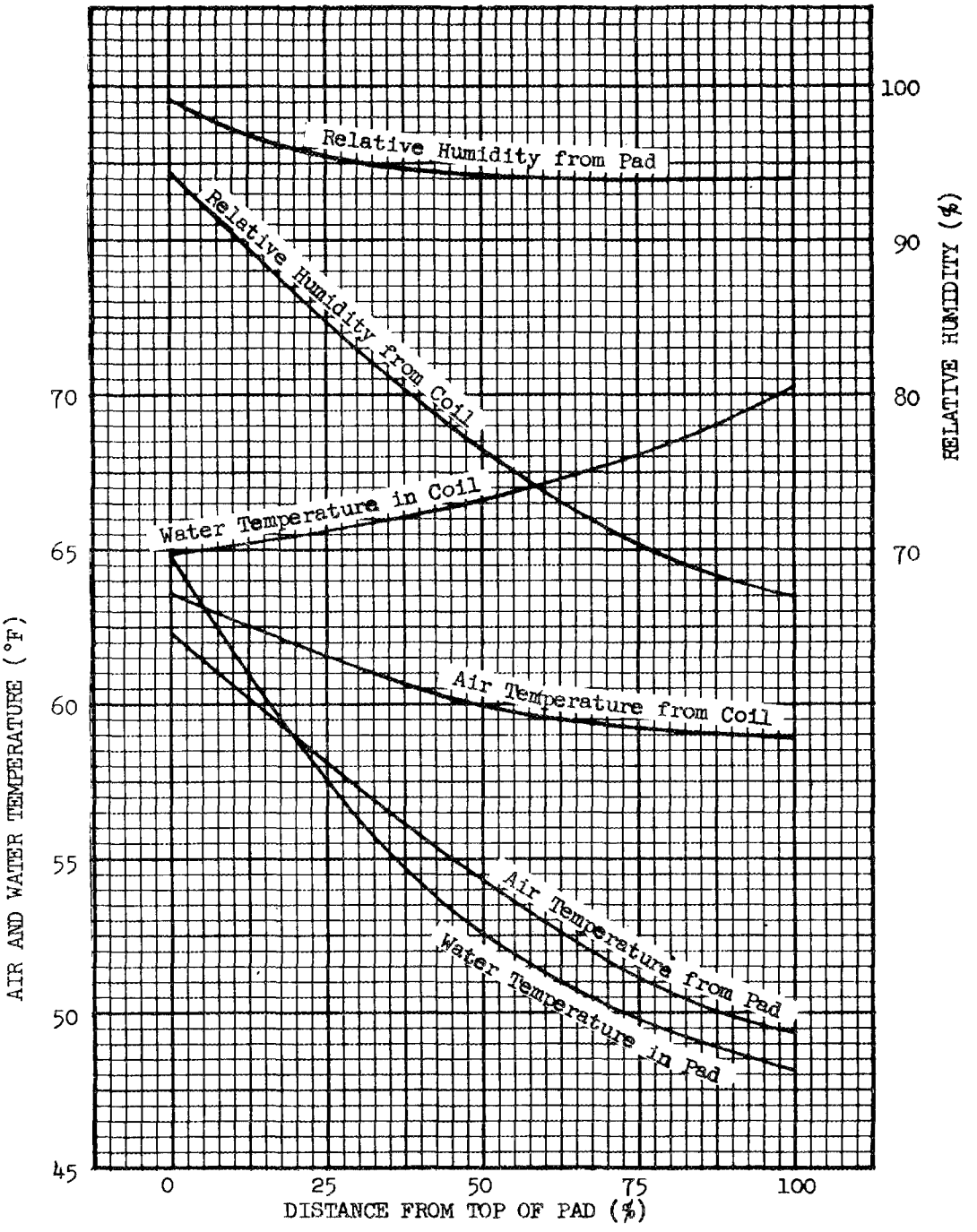

Fig. 10. Air and Water Conditions for Summer Night Operation with Evaporative Pads and Finned-Tube Heater with Upward Flow in Heater.

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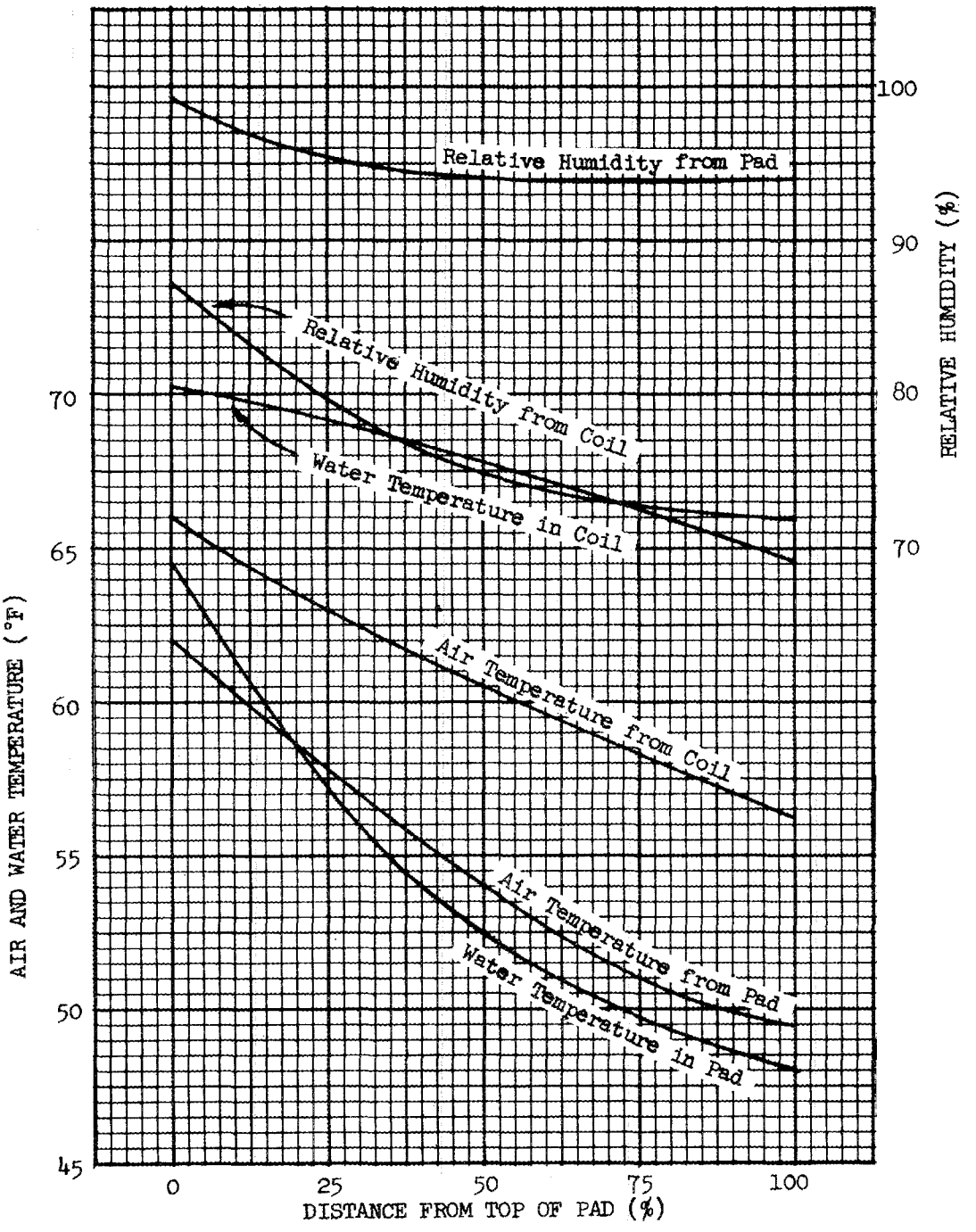

Fig. 11. Air and Water Conditions for Summer Night Operation with Evaporative Pads and Finned-Tube Heater with Downward Water Flow in Heater.

night conditions with a water flow rate of 44,100 lb/hr and the fans at one-half full flow. In Fig. 10 the water is assumed to flow upward through the heater and for Fig. 11 the flow is downward. About 25% of the heat is transferred from the heater and the remainder from the pad. The average relative humidity leaving the heater is between 75 and 80%.

CONTROL SYSTEM

It is difficult to describe the conditions of both air and water, in summer and winter, without leaving the impression that our system is complicated. Actually, it is very simple and easy to control because the heating and cooling rates are low and there is no worry about sudden temperature changes.

The control system is also simple. The temperature of the atmosphere inside the houses is the control point and would be measured by four or five shaded thermocouples. The average of this signal would cause small electric motors to open the vent louvers when the signal becomes higher (warmer) than the desired temperature (the setpoint), and to close them when the air temperature is lower than desired. The action of the louvers would therefore determine whether warm air were blown to the outside or sent to the attic space to be recirculated back to the heat exchangers.

The inlet air louvers at the heat exchangers have no control because they are the gravity-type, opening on the small pressure difference when air is vented from the opposite end of the house and closed by the force of gravity when the venting ceases.

Humidity is the second variable which we wish to regulate. Several humidistats will measure the water content of the air and the average signal used to operate a regulating valve in the waterline to the finned-pipe (dry) heater. By opening the valve, warm water would flow to the finned heater, thereby warming the air and reducing its humidity.

Both the temperature and humidity control are standard equipment for a modern greenhouse and although they normally cause different gadgets to operate, the cost and difficulty would not be greater for the proposed system.

POULTRY AND SWINE HOUSES

The application of the heating-cooling system described for the greenhouse installation to poultry and swine houses is simplified by the fact that the solar heat input is greatly reduced and the heat losses are smaller. The heating and cooling needs are therefore greatly reduced--to less than

one-half--but the same basic heat exchange system can provide the desired environment in the poultry and swine houses. The temperature range for optimum growth and feed efficiency for broilers and laying hens is 70 to 80°F, whereas for pigs it is 60 to 70°F. A humidity range of 75 to 85% is satisfactory.

We have assumed poultry houses and swine houses such as are in common use in the United States, modified only to accommodate the heat exchangers and inlet louvers in one end, fans in the other end, and a sheet of plastic to separate the attic from the growing area, as shown for the greenhouse in Fig. 8.

As a typical broiler house, we chose for study the design developed by the U.S.D.A. and several agricultural experiment stations. The houses are single wooden structures 24 ft by 124 ft for a total area of 3000 ft^2. Each house produces 3200 birds 2 1/2 to 3 lb in weight in nine weeks, making the annual production 16,000 birds per year. We assume 350 such houses occupying a space of about 26 acres in our hypothetical food-growing complex.

The laying houses chosen were the completely automated caged-bird houses used by several large United States egg producers. Each 40-ft by 125-ft unit houses 12,800 hens which produce about 8,000 eggs per day. We have included 70 laying houses with a total production of 560,000 eggs per day in our proposed installation. An incubating house for both broilers and layers is provided to supply the necessary young chicks on site.

The swine production units are of the type developed by Hazen _et al._, at Iowa State University. The farrowing and nursery building is 29 ft by 132 ft, with 3400 ft^3 of space for 1750 pigs a year. The growing-finishing houses are 50 by 120 ft and accommodate 700 growing pigs at a time. The floors and pens are concrete and the structures are sheet metal. We suggest that an area of 50 acres be devoted to 100 farrowing houses and 75 growing houses to permit a total production of about 30 million lb of pork per year.

PROBLEMS RELATED TO USE OF HEAT FROM POWER STATIONS

Although the results of this study apply equally to fossil power plants, there are implementation problems for both nuclear and fossil plants. Probably the most important problem relates to the financing and managing of a very large food production operation. The total cost of a 200-acre installation such as described in this report would range from 15 to 30 million dollars. Even a 100-acre installation (which could dispose of the entire 500 Mw waste heat) would be a large investment, by food-growing standards, although by steam plant standards (75 to 100 million dollars for a 330-Mwe station), it is not too large.

Perhaps more difficult would be the organizational and managerial problems associated with a combined vegetable—poultry—pork operation. Each requires technology and expertise which would be difficult to find in an existing organization, although some integrated feed companies come close. It would seem prudent for any organization which might undertake such a venture to establish one sector at a time.

Another problem, which would have to be explored with the utility, is how the foodplex would manage operations to guarantee sufficient cooling capacity and to insure that the return cooling water would not damage the turbine-condenser cooling surface. In turn, the foodplex operator would need to be informed in decisions to vary power (and cooling) load.

For nuclear power stations, the question of radioactive contamination in the cooling water will arise. The normally installed monitors should provide sufficient protection, especially if the food areas are at least a few hundred feet from the station, because the water transit time would be long enough (many seconds) to permit shutting off the flow. Another possibility is contamination as a result of airborne activity from an accident but this is judged to be a small enough chance as not to be a problem. Although not an exactly parallel situation, land around nuclear stations is already being used for normal farming operations without difficulty.

SITE LAYOUT

Figure 5 is a layout showing 880 50-ft by 100-ft greenhouses (100 acres) and 50 acres each of poultry and swine houses at the Fort St. Vrain reactor site. The arrangement shows rows of ten houses having common sides with the ridge line running east and west. Figure 4 shows the roads and piping arrangement. Each row of ten houses would be located "back-to-back" with an adjacent row and all piping located in a 15-ft alley. The front, or fan exhaust side, of each row faces a road. Roads would also be located at the side of each unit of ten houses.

The optimum pumping and piping arrangement will depend somewhat on the power plant heat dump system. If the present condenser cooling water pumps have sufficient head, they could be used to pump the water directly from the condenser to the greenhouses. The return water would then be pumped back to a cold water inventory basin which is the condenser supply water. An alternate arrangement which separates the greenhouse system from the power plant is shown in Fig. 12. In this system the condenser water is dumped to a hot water basin which is then the supply for the greenhouses. Again, the return water is pumped back to a cold water basin. This latter method is more expensive as it requires additional pumps; however, some credit should be taken for reducing the head required for the present pumps.

Our cost estimates are based on greenhouse requirements. The equipment and heating requirements for the poultry and swine houses would have lower costs but we have not estimated them. For the greenhouses we calculated a 100-acre case and a 200-acre case. For the 100-acre case, the piping system to the greenhouses was assumed to consist of four main 36-in.-diam lines from the condenser or the holding basin. The size of the mains was reduced as water was supplied to laterals in the greenhouse area. Rather than attempt to optimize the system, the following assumptions were made; the pipe size was selected so that the water velocity would not exceed 15 ft/sec and after the velocity dropped below 10 ft/sec, the next smaller pipe size was used (again with the 15 ft/sec limitation). The same assumptions were used to size the lateral with the exception that when the

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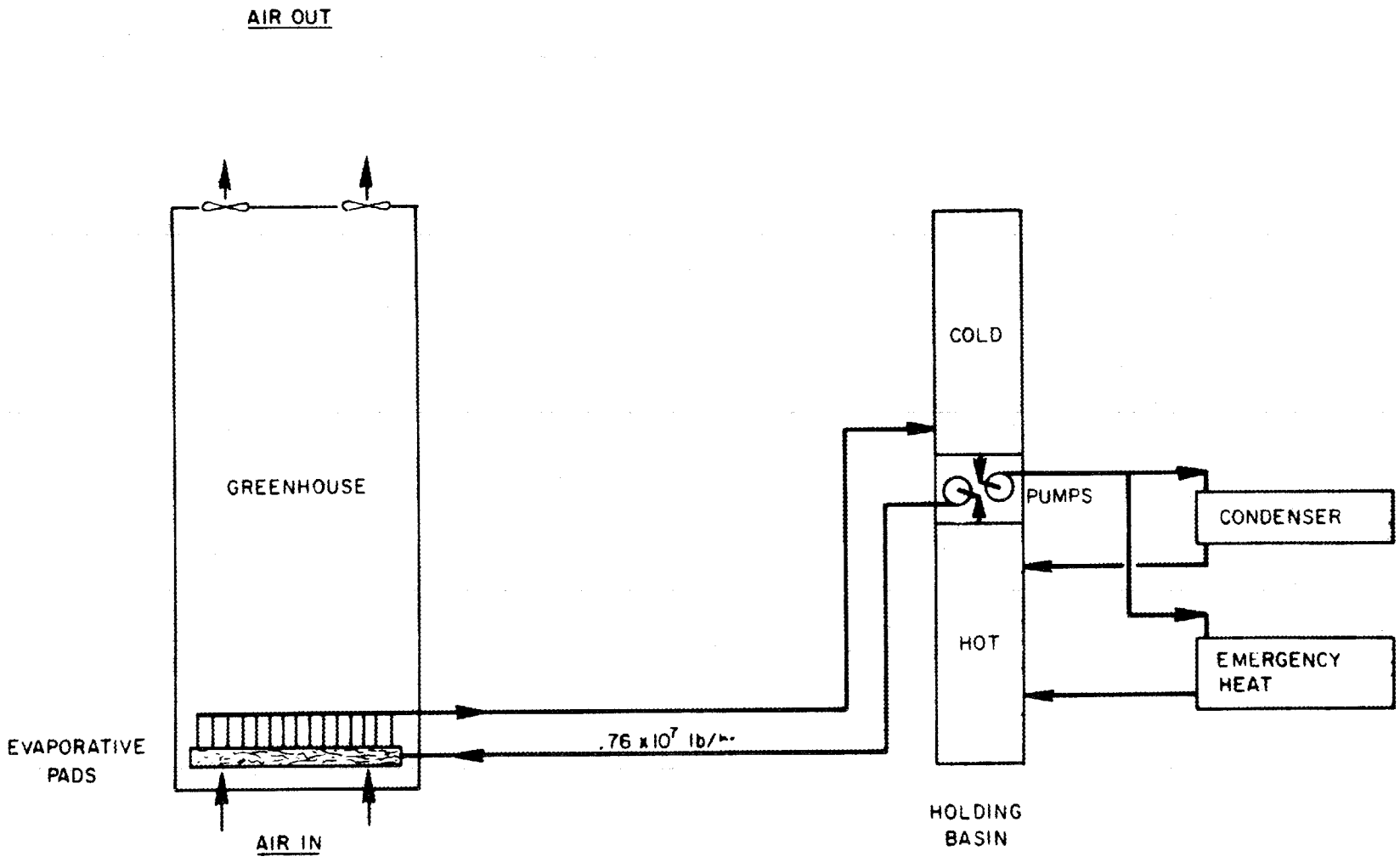

Fig. 12. Main Pumping Station - Alternate Arrangement.

size dropped to 6 in., no further reductions were made. A summary of the piping sizes and lengths are shown in Table 5.

Table 5. Piping Sizes and Lengths for 100 Acres.

Size (in.)	Total Length (ft)
Main Piping Runs	
36	12,700
30	3,300
24	2,400
16	2,500
Laterals	
10	11,000
8	5,500
6	5,500
Connections to Greenhouses	
3	14,000

The return water from two units of ten houses each was assumed to drain by gravity to a common sump where it was then pumped back to the cold basin. The return system requires the same size mains as the supply system but does not require the laterals that were used to supply the individual houses. There are a total of 44 return pumps required. The head required for the pump farthest from the basin is 40 to 45 ft and for the nearest pump is 25 to 30 ft. The flow rate for each return pump is 3500 gpm. The horsepower required for these pumps varies between 30 and 45. No allowance is made for elevation differences between the pumps and the return basin.

The maximum head loss in the worst supply line is 70 to 75 ft and the total flow rate for all four mains is 155,000 gpm. The total horsepower required, if supplied by a single pump, would be about 3400.

For the 200-acre case, the number of mains was increased from four to eight and the diameter of the piping reduced. In general, the total length of lines required was doubled and the diameter of the piping reduced to about 71% of that for the 100-acre case.

Costs

An estimate of the direct costs for a heating and cooling system is shown in Table 6. Only the costs of those items required to use the power plant waste heat are presented here. These costs include the pumps, piping, valves, trenches, pits, emergency heat facility, wire, and plastic to form the attic and the dry heat exchangers. It is assumed that the houses would be equipped with the pad-fan system for summer operation regardless of the type of heat used, so the costs of the pad-fan system is not repeated in our costs. Also, no costs were included for the hot or cold water basins because the present cooling tower system already has a basin capacity of about 10^6 gal. The Fort St. Vrain Plant also has four cooling water pumps; two rated at 43% of full flow and two at 7%. All pumps have a discharge head of 70 ft. If the two-basin system shown in Fig. 11 is used, then the head and power required for the present pumps could be reduced. If the water were to be pumped directly from the condenser to the greenhouses, then the head and power of the pumps would have to be increased. The costs for the main pumping station shown in Table 6 represents an incremental cost that is somewhere between the two methods.

Table 6. Estimate of Direct Costs of Heating and Cooling Equipment For 100 Acres and 200 Acres of Greenhouses

Items	Costs	
	100 Acres	200 Acres
Supply and return mains	818,000	1,162,000
Supply laterals and valves	287,000	407,000
Return trenches	132,000	242,000
Return pits	119,000	195,000
Return pumps	120,000	220,000
Main pumping station	300,000	300,000
Emergency heat facility	650,000	975,000
Plastic attic liners	440,000	880,000
Finned-tube heaters	1,320,000	2,640,000
Total	4,186,000	7,021,000

It should be emphasized that the pumping and piping system is sized to use the entire 500 Mw discharged from the power plant. As the area of greenhouses increases, the cost of the pumping and piping system per acre decreases. The cost of the main pumping station would be approximately the same. The number of return pumps would be doubled; however, the power and flow rate would be halved. The size of all of the piping could be reduced to about 70% of their present diameter. The cost of the piping and pumps for 200 acres would be about 1.40 times that for 100 acres and the 200 acres would use the plant waste heat much more efficiently.

No costs were included for temperature or humidity controls. With the system chosen, the costs should be comparable to those required for conventional greenhouses.

SUMMARY AND CONCLUSIONS

The evaporative pad cooling system will adequately cool the ambient air even when supplied with excessive amounts of warm water. The only question concerning their operation is the maximum amount of water that can be supplied to the pads before they become flooded. This should not be a problem for the 200-acre case and will probably be no problem for the 100 acres. With regard to the power plant operation, the pads are more effective than the cooling towers specified (water temperature range of 80 to 102°F). The water from the pads for a 64°F wet-bulb design temperature would be about 67°F.

For winter operation, the pads are very effective for heating the greenhouses. The temperature of the air passing over the plants is about 5 to 7°F above the return water temperature. The major problem with using only the pads for heating is that the relative humidity of the air leaving the pads will be in the range of 90 to 100%. By using a finned-tube heat exchanger in series with the pads, the temperature of the air passing over the plants will be about 10°F above the return water temperature and the relative humidity of the air reduced to 75 to 80%.

The economics of the proposed system depends on such factors as the number of acres used, whether the finned-tube heaters are required, and whether one takes credit for eliminating the cooling towers. A summary of

the various options is given in Table 7. In examining this table, one should note that the cost of conventional heating systems for greenhouses is between 20,000 and $25,000 per acre and that the value of the heat to the greenhouse operator is between 5,000 and $10,000 per year per acre. For the 200-acre case the total value of the heat at conventional prices is between 1,000,000 and $2,000,000 or 8 to 16% of the value of the electrical output of the plant.

Thus, it appears that the waste heat from many existing and future power plants (fossil and nuclear) can be used profitably in several food-growing operations, large and small. The grower's profit from heat cost savings could be 5,000 to $10,000 per acre, depending on the plant location.

The profit to the utility, even if it gives away the heat and takes credit only for cooling tower capital and operating cost savings, can be $250,000 or more per year for a station such as Public Service of Colorado's Fort St. Vrain.

Table 7. Summary of Cost Per Acre

	Number of Acres	
	100	200
Without taking credit for replacing cooling towers		
Cost/acre with finned-tube heaters	$41,860	$35,110
Cost/acre without finned-tube heaters	28,660	21,910
Taking $1,500,000 credit for replacing cooling towers		
Cost/acre with finned-tube heaters	26,860	27,610
Cost/acre without finned-tube heaters	13,660	14,410

Several important problems must be studied before such a proposal could be considered feasible. The concept of heat transfer and environmental control with warm water must be demonstrated. The problem of implementing the scheme should be examined by agriculturists, financiers, and economists, as well as by utility executives. The organizational and managerial problems require technology and expertise which might not be available except in some of the integrated feed-food companies.

Questions of protection against radioactive contamination of the food product, in case nuclear plants are used as the heat supply, and questions related to the effects of the agricultural operations on the power plant must be answered to the satisfaction of the entrepreneur.

None of these problems seems insurmountable, and in view of the possible benefits of energy conservation, pollution reduction, and overall economic gain, it is suggested that a demonstration facility incorporating these ideas would be a worthwhile undertaking.

Finally, although it appears conceivable to use a large fraction of the heat presently being discarded from power stations, we recognize that smaller-scale complexes using a proportionately smaller fraction of the heat are the more likely possibility.

APPENDIX

Method of Analysis

In conventional greenhouse evaporative cooling systems a relatively small quantity of water (a minimum of 1/3 gpm/ft of pad length) is recirculated through the pads. For these conditions the water temperature will be approximately the same as the air wet-bulb temperature and will remain at a constant temperature throughout the pad. The effectiveness, η, of the pads, defined as

$$\eta = \frac{t_{a1} - t_{a2}}{t_{a1} - t_{wB}} \quad , \tag{1}$$

where

t_{a1} = inlet air dry-bulb temperature, ^{o}F,

t_{a2} = outlet air dry-bulb temperature, ^{o}F,

t_{wB} = inlet air wet-bulb temperature, ^{o}F,

is between 80 and 85% for an air flow of 150 cfm/ft^2. For these operating conditions, the enthalpy of the air remains constant while passing through the pad and one can determine the product of the heat transfer coefficient and the surface area. Starting with the basic equation,

$$q = hA\ \Delta tm \quad , \tag{2}$$

where

q = heat transfer rate, Btu/hr,

h = heat transfer coefficient, Btu/hr·ft^2·^{o}F,

A = area, ft^2

Δtm = mean effective temperature difference, ^{o}F,

for the case under consideration,

$$\Delta tm = \frac{(t_{a1} - t_w) - (t_{a2} - t_w)}{\ln\left(\frac{t_{a1} - t_w}{t_{a2} - t_w}\right)} , \qquad (3)$$

where

t_w = water temperature, ^oF.

From Eq. (1)

$$t_{a2} = t_{a1} - \eta(t_{a1} - t_w) . \qquad (4)$$

Substituting Eq. (4) in Eq. (3) one gets

$$\Delta tm = \frac{(t_{a1} - t_{a2})}{\ln\left[\frac{(t_{a1} - t_w)}{(t_{a1} - \eta(t_{a1} - t_w) - tw)}\right]} ,$$

$$= \frac{t_{a1} - t_{a2}}{\ln \frac{(t_{a1} - t_w)}{(1 - \eta)\ (t_{a1} - t_w)}} ,$$

$$= \frac{t_{a1} - t_{a2}}{\ln\left(\frac{1}{1 - \eta}\right)} , \qquad (5)$$

and then,

$$q = hA \frac{t_{a1} - t_{a2}}{\ln\left(\frac{1}{1 - \eta}\right)} . \tag{6}$$

The heat transfer rate can also be written as:

$$q = G\ C_p\ (t_{a1} - t_{a2}) \quad , \tag{7}$$

where

G = weight flow of air, lb/hr,

C_p = specific heat of air, Btu/lb·°F.

Equating Eqs. (6) and (7)

$$G\ C_p\ (t_{a1} - t_{a2}) = hA \frac{(t_{a1} - t_{a2})}{\ln\left(\frac{1}{1 - \eta}\right)} ,$$

$$hA = G\ C_p\ \ln\left(\frac{1}{1 - \eta}\right) . \tag{8}$$

For an effectiveness, η, of 80% for an air flow rate of 150 cfm/ft^2, hA is found to be 250 Btu/hr·°F for each ft^2 of face area of the pads.

The value of the product hA was also calculated by using data from packed beds both to check the above value and to determine the effect of

varying air flow rate. During operation, the pads are about 2-in. thick and contain between 4 and 5 oz of aspen wood fibers/ft^2 of face area. The fibers are splinters 4- to 6-in. long and "about the diameter of a pencil lead." Assuming the fibers to be cylindrical rods 0.040-in. diam, the surface area/ft^2 of face area is 6.12 ft^2/ft^2 of face area. The heat transfer coefficient correlation was taken from Jakob[16] and was

$$j_a = \frac{h}{\rho\ C_p\ V} (P_r)^{2/3} \quad , \tag{9}$$

where

$j_a = 1.625\ (re)^{-.507}$,

ρ = air density, lb/ft^3 ,

V = air face velocity, ft/hr ,

P_r = Prantl's No.

Re = Reynolds No.

For the normal flow condition of 150 cfm/ft^2 of face area, the heat transfer coefficient was found by Eq. (9) to be 45 $Btu/hr \cdot ft^2 \cdot {}^oF$. The product of hA is then 275 $Btu/hr \cdot {}^oF$. The rather close agreement with that calculated above is probably fortuitous, considering the lack of detailed surface to volume measurements on the fibers. Using the 250 $Btu/hr \cdot {}^oF$ value as a base, the hA vs air flow rate is shown in Fig. A1.

The heat and mass transfer analysis required for the pads is identical to that for cooling towers. The present case is somewhat more complicated than the usual cooling tower because the units use cross-flow rather than counter-flow paths. For most conditions that will exist in the pads, both sensible and latent heat exchange is taking place. The sensible heat transfer equation is

$$G\ C_p\ dt_a = h\ (t_w - t_a)\ dA \quad , \tag{10}$$

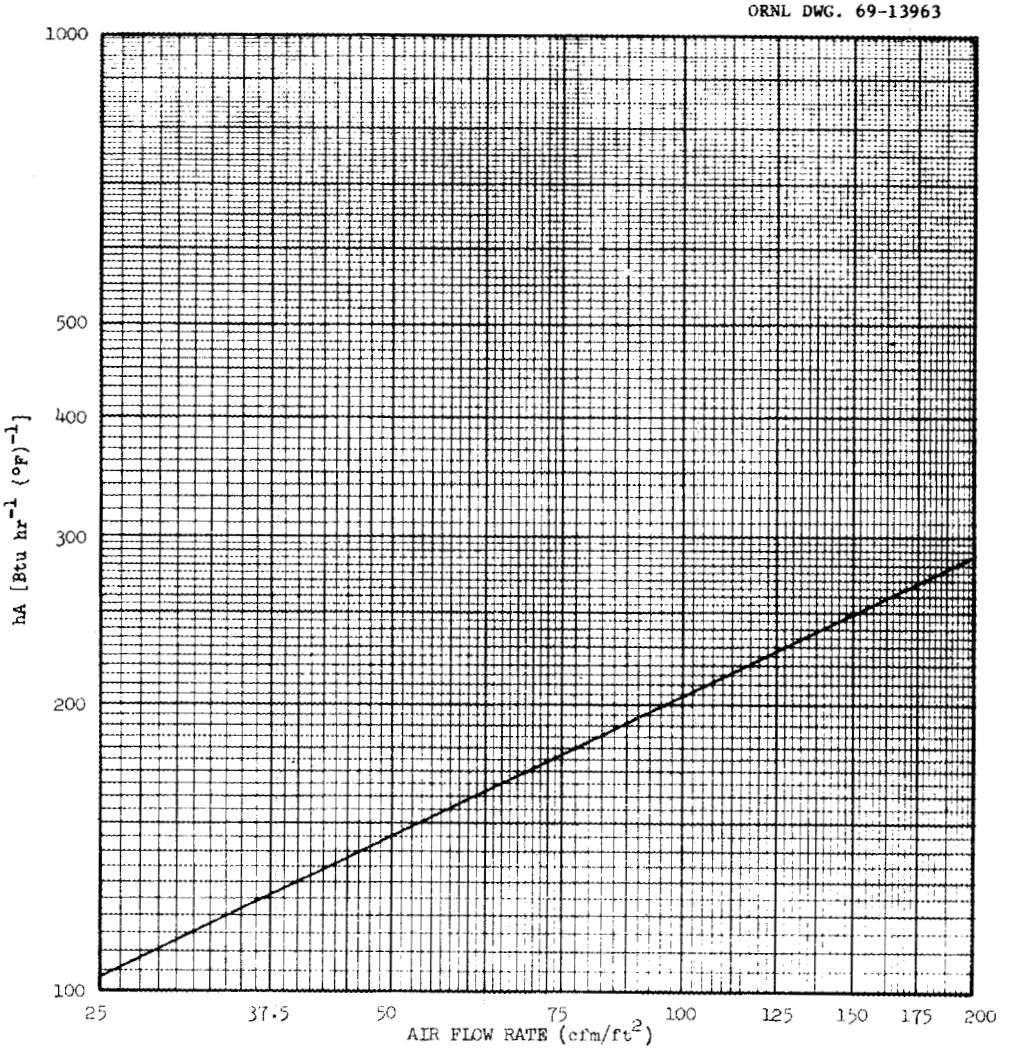

Fig. A1. hA per Ft^2 of Face Area vs Air Flow Rate.

using the relation that the mass transfer coefficient, K, is equal to

$$K = \frac{h}{C_p} \quad , \tag{11}$$

Eq. (10) becomes

$$G\ C_p\ dt_a = K\ C_p\ (t_w - t_a)\ dA \quad . \tag{12}$$

The latent heat transfer equation is

$$G\alpha\ d\ H_a = K\alpha\ (H_s - H_a)\ dA \quad , \tag{13}$$

where

H_a = humidity ratio (lb of water/lb of dry air) for air,

H_s = humidity ratio (lb of water/lb of dry air) for saturated air at the water temperature,

α = heat of vaporization, Btu/lb.

Combining Eqs. (12) and (13) one gets

$$G\ C_p\ dta + G\alpha\ dH_a = K\ C_p\ (t_w - t_a)\ dA + K\alpha\ (H_s - H_a)\ dA \quad , \tag{14}$$

$$= K \left[(C_p t_w + \alpha\ H_s) - (C_p t_a + \alpha\ H_a) \right] dA$$

$$G\ C_p\ dta + G\alpha\ dH_a = K\ (\lambda_s - \lambda_a)\ dA \quad . \tag{15}$$

where

λ_s = enthalpy of saturated air at the water temperature, Btu/lb,

λ_a = enthalpy of air, Btu/lb.

Also

$$G\ C_p\ dta + G\alpha\ dH_a = G\ d\lambda_a \quad . \tag{16}$$

Then

$$G\ d\lambda_a = K\ (\lambda_s - \lambda_a)\ dA \quad , \tag{17}$$

which is the basic equation for cooling towers.

The total heat transferred is usually taken as

$$q = L\ (t_{w1} - t_{w2}) \quad , \tag{18}$$

where

L = water flow rate, lb/hr.

This equation neglects the amount of water evaporated and under usual conditions introduces only a small error. The actual evaporation rate varies from about 1% for most cases to 3% for very hot days. The total heat

transferred by the water is given to the air so that

$$q = L(t_{w1} - t_{w2}) = G\,(\lambda_{a1} - \lambda_{a2}) \quad . \tag{19}$$

At this point let us introduce a fictitious flow rate, w, defined as

$$q = w\,(\lambda_{s1} - \lambda_{s2}) \quad . \tag{20}$$

Referring to Fig. A2, it can be seen that the flow rate, w, corresponds to a fluid having the vapor pressure of water and the enthalpy, λ, properties of saturated air. It is interesting to note that at 75 to 80°F, the slope of the λ_s vs t_w curve is such that $d\lambda_s/dt_w = 1$; that is, the effective specific heat is 1, the same as water and for tower operating about this point, the value of w will be the same as L, the water flow rate. However, for high or lower water temperatures, the value of w is considerably different from that of L.

Equation (17) can now be rewritten as

$$w\,d\lambda_s = K\,(\lambda_s - \lambda_a)\,dA \quad , \tag{21}$$

or

$$\int \frac{d\lambda_s}{\lambda_s - \lambda_a} = \frac{KA}{w} \quad , \tag{22}$$

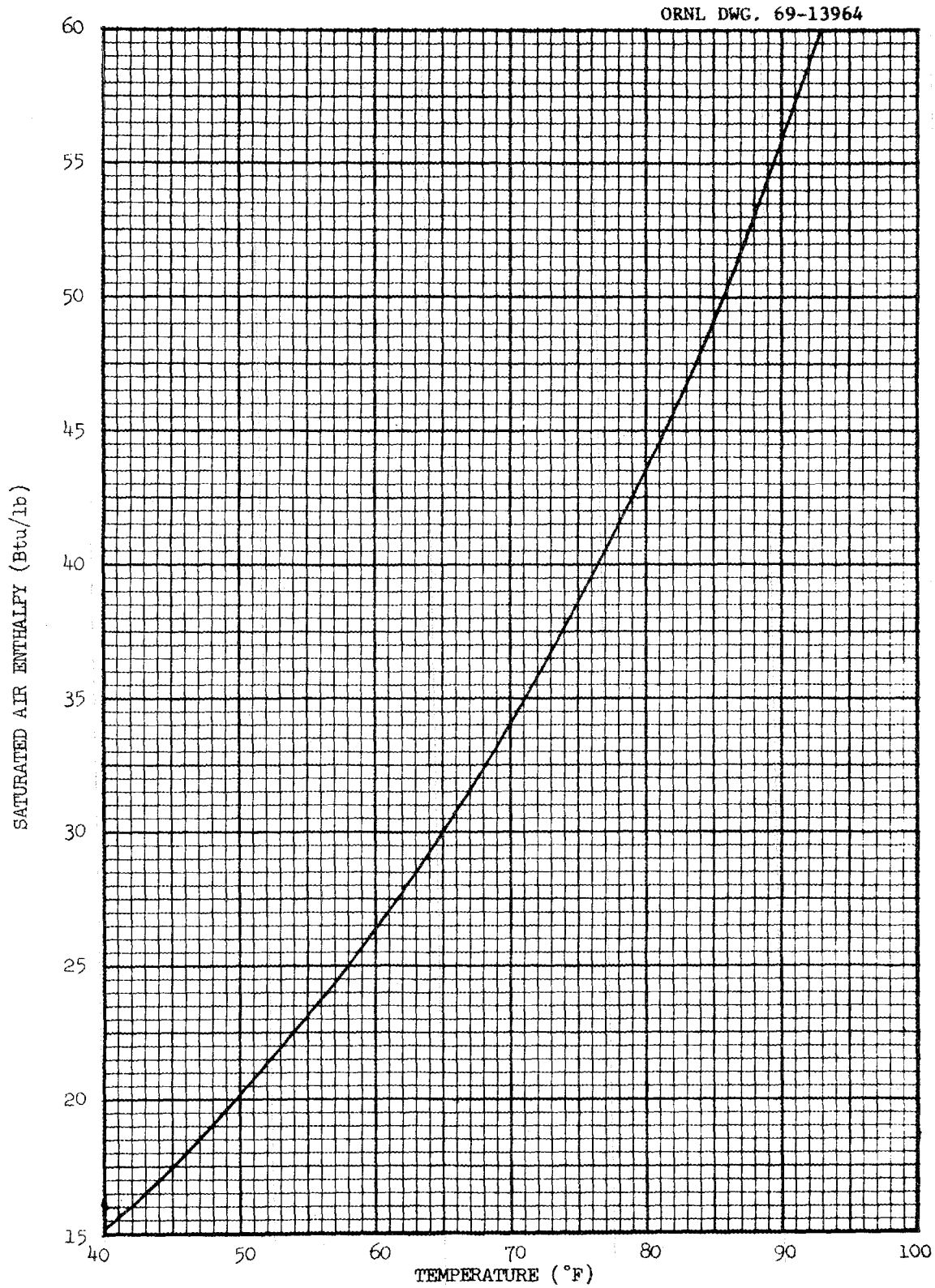

Fig. A2. Enthalpy of Saturated Air.

where $\frac{KA}{w}$ replaces the normal "tower characteristic" term $\frac{KA}{L}$. The integral term of Eq. (22) is the enthalpy change of the saturated air divided by the mean effective enthalpy difference between saturated air and the air in the pads. A second equation similar to Eq. (22) can also be written

$$\int \frac{d\lambda_a}{\lambda_s - \lambda_a} = \frac{KA}{G} \quad , \tag{23}$$

where the term $\frac{KA}{G}$ is called "the number of air transfer units."

The evaluation of the integral term in both Eqs. (22) and (23) is complicated by the curvatures of the saturated air line and the fact that the pads are designed for crossflow. In all cases considered here, the temperature drop in the water will be 22°F and for this range the error in neglecting the curvature of the saturated air line is less than 1°F in the water temperature. This error will be neglected at this time.

The procedure derived by Nusselt and reported by Jakob[17] to determine the mean effective temperature difference for crossflow between unmixed streams is used to determine the performance of the pads. In this procedure, two dimensionless terms are defined as,

$$a' = \left(\frac{hA}{w\ C_p}\right)_1 \quad , \tag{24}$$

$$b' = \left(\frac{hA}{w\ C_p}\right)_2 \quad , \tag{25}$$

where a' refers to stream 1 and b' refers to stream 2. Both a' and b' are equivalent to $\Delta t/\Delta t_m$ where Δt is the temperature change in the stream and

Δt_m is the mean effective temperature difference. To apply this procedure to the present case, the dimensionless terms will be defined as

$$a = \frac{KA}{G} \quad , \tag{26}$$

$$b = \frac{KA}{w} \quad . \tag{27}$$

Nusselt evaluated a term, Z, shown in Fig. A3 which depends only on a and b.

$$Z = \frac{\lambda_{a2} - \lambda_{s1}}{\lambda_{a1} - \lambda_{s1}} \quad , \tag{28}$$

where λ_{a2} in this case is the mixed mean exit enthalpy of the air. From Eq. (28) it can also be shown that

$$q = -G\ (1 - Z)\ \ (\lambda_{a1} - \lambda_{s1}) \quad . \tag{29}$$

For the present study in which the water temperature difference is always 22°F and the inlet air conditions are known, the procedure for using Eq. (29) and Fig. A3 requires one to first assume a $\Delta\lambda_s$ to determine w. Then determine $\frac{KA}{w}$ and $\frac{KA}{G}$ to get the value of Z from Fig. A3. Use Z in Eq. (29) to determine λ_{s1}. Then with λ_{s1} and knowing the temperature difference of the water is 22°F, one can determine λ_{s2} and check the initial assumption. Normally two, or at most three, trials will give a

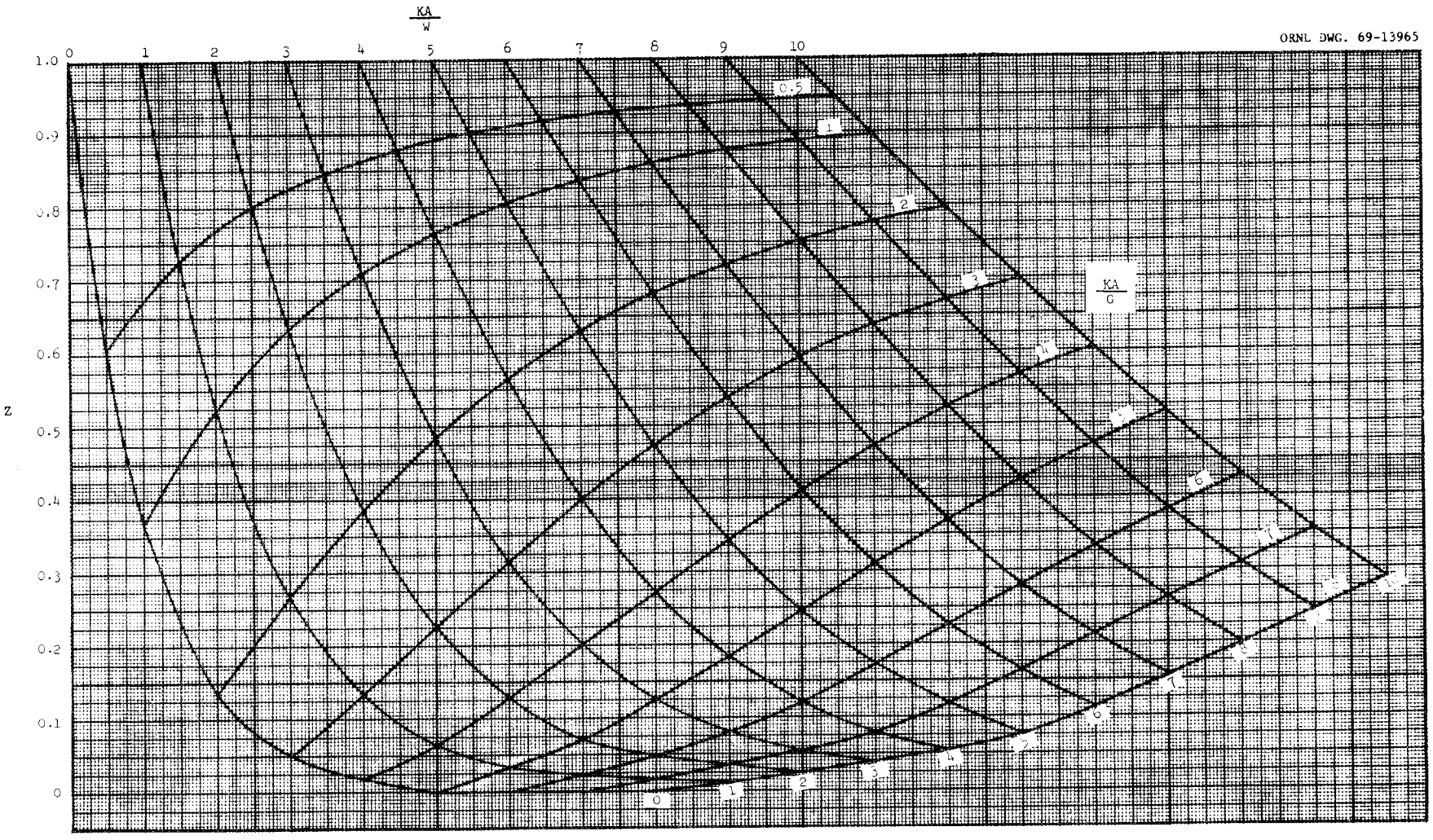

Fig. A3. Z Factor for Cross Flow Heat Exchangers.

satisfactory solution. Once the value of w is established, one can then determine the water temperature and air humidity at any place or section with the pad. For example, if one wants the mean water temperature through the pad, the value of $\frac{KA}{G}$ remains the same and incremental values of $\frac{KA}{w}$ are used. If the temperature of the water leaving 10 sections is wanted, then the temperature leaving the first section is found by using $0.10 \frac{KA}{w}$, the temperature leaving the second section by using $0.20 \frac{KA}{w}$, etc. This procedure also gives the mixed mean air humidity for each section. To divide the pads in sections in the direction of air flow, the procedure is the same except the incremental values of $\frac{KA}{G}$ must be used.

Figure A4 shows an example of the pad performance using the method outlined above. The water flow rate per linear foot of pad is 1765 lb/hr, the air flow rate is 182 cfm/ft^2, and the inlet air conditions are 95°F dry bulb and 64°F wet bulb for which the inlet air enthalpy is 29. An overall energy balance for the pad shows that for a 22°F drop in the water temperature, the inlet and exit temperatures are 89 and 67°F. For the results shown in Fig. A4a, the analysis was done for ten vertical sections and the water temperature along the pad and the exit air enthalpy for each section is given. In Fig. A4b the analysis is done on the top vertical section only but with the pad divided into four parts in the direction of air flow. The lower graphs then give the water temperature leaving Section 1 at different horizontal positions and also the air enthalpy leaving these sections.

The above procedures give detailed knowledge of the water temperature and the air enthalpy. However, for use in greenhouses, one also needs to know the relative humidity of the air. To determine the relative humidity one needs to know either the air temperature or humidity ratio in addition to the enthalpy. Once the temperature distribution of the water is known, then Eq. (10) can be used to calculate the air temperature in the pad. For the case shown in Fig. A4, the air temperature leaving Section 1 is about 88 to 89°F and the relative humidity about 75%. The air leaving Section 10 is about 74°F with a relative humidity about 58%. The mixed mean outlet air temperature is 76 to 77°F, and the relative humidity about 80%.

It should be noted that the pad effectiveness used in these calculations is that reported for low water flow rates while the actual value will

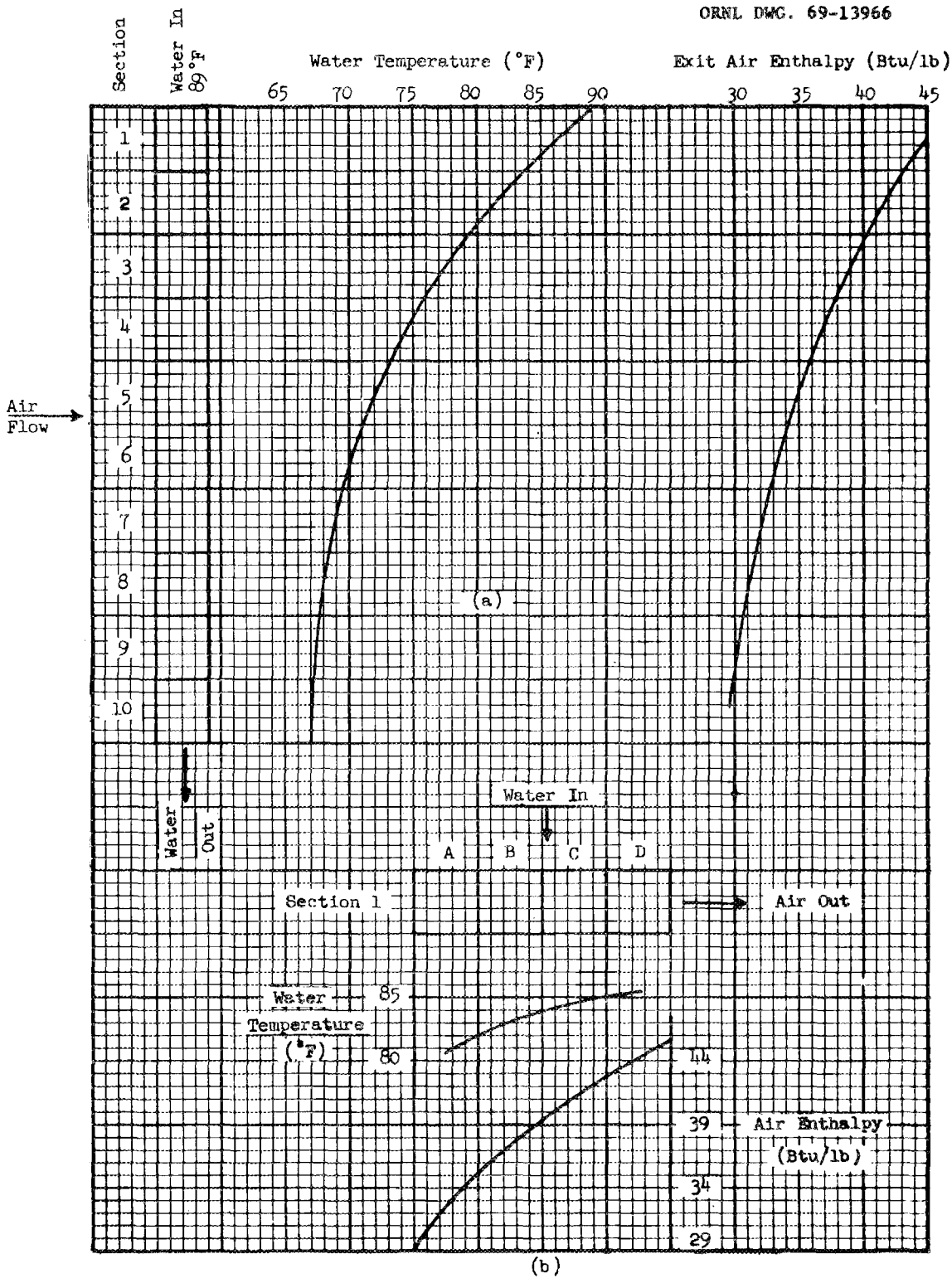

Fig. A4. Typical Air and Water Conditions in a Cross Flow Evaporative Cooling Pad.

depend on the flow rate, especially at high flows. The problem here is that as the flow increases, the water will bridge between the fibers and block the flow of air from these areas. Also, at high flow rates, these bridged areas may extend from the top to the bottom of the pad, thus forming rivulets which will reach the bottom of the pad at temperatures above that predicted by the preceding analysis. The effect of the water flow rate needs to be determined experimentally.

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