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MASTER

DESIGN AND PERFORMANCE CONSIDERATIONS OF EVAPORATIVE-PAD, WASTE-HEAT GREENHOUSES*

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ABSTRACT

Rising fuel costs and limited fuel availability have forced greenhouse operators to seek alternative means of heating their greenhouses in an effort to reduce production costs and conserve energy. One such alternative uses power plant reject heat, which is contained in the condenser cooling water, and a bank of evaporative pads to provide winter heating.

This paper details the design technique used to size the evaporative pad system to meet both summer cooling and winter heating demands. Additionally a computational scheme that simulates the system performance is presented. This analytical model is used to determine the greenhouse operating conditions that maintain the vegetation in its thermal comfort zone.

The evaporative pad model uses the Merkel total heat approximation and an experimentally derived transfer coefficient. Energy balance considerations on the vegetation provide a means of viewing optimal vegetation growth in terms of greenhouse environmental factors.

In general, the results indicate that the vegetation can be maintained within its thermal comfort zone if sufficient warm water is available to the pads and the air stream flow is properly adjusted.

NOMENCLATURE

- a = air water contact area per unit volume (m^2/m^3)
- A = total area of house in contact with ambient environment (m^2)
- A_c = CELdek face area required for summer cooling (m^2)
- A_h = CELdek face area required for winter heating (m^2)
- A_L = area of house through which conduction loss occurs (m^2)
- A_p = installed face area of CELdek pad (m^2)
- A_s = area of house through which solar flux enters (m^2)
- C_H = humid specific heat of air-water vapor mixture (J/kg°C)
- C_L = water specific heat (J/kg°C)
- $[CO_2]_s$ = concentration of CO_2 in stomate (ppm)
- C_p = specific heat of dry air (J/kg°C)
- C_v = specific heat of water vapor (J/kg°C)
- D = leaf dimension parallel to wind velocity (m)
- G = air flow rate (kg/sec)
- H = latent heat of vaporization (J/kg)
- h = convection heat transfer coefficient for leaf surface ($W/m^2{}^\circ C$)
- h_1 = convection coefficient on inside of glass ($W/m^2{}^\circ C$)
- h_2 = convection coefficient on outside of glass ($W/m^2{}^\circ C$)
- h_G = heat transfer coefficient for the evaporative pad ($W/m^2{}^\circ C$)
- h_o = heat transfer rate for CELdek pads ($W/m^2{}^\circ C$)
- I = index for row in pad
- i = total heat of air at air temperature (J/kg)
- i_{vo} = water latent heat of vaporization at t_o (J/kg)
- i_{Tw} = total heat of saturated air at temperature T_w (J/kg)
- J = index to indicate pad column
- k = thermal conductivity of glass (W/m°C)
- K = overall pad mass transfer coefficient (kg/sec m^2)
- K_{SC} = m-m constant (kg/m^3)
- K_{SL} = m-m constant (W/m^2)
- K_{SW} = m-m constant
- k_1 = empirical coefficient specified by Gates for determining the convective heat transfer
- k_2 = empirical coefficient specified by Gates for determining the air boundary layer resistance to water vapor transport
- L = water flow rate (kg/sec)
- L_{PHAR} = light intensity in photosynthetic active region
- L_w = pad water flow rate (m^3/sec/linear meter)
- M = mean beam length (m)
- $\dot{m}_a$ = air mass flow rate in the greenhouse (kg/sec)
- $\dot{m}_{EVAP}$ = leaf transpiration rate (kg H_2O/sec)
- PD = air flow pressure drop in pads (kN/m^2)
- Q = total heat transfer to the greenhouse air (W)
- $\dot{Q}_a$ = greenhouse air flow rate (m^3/sec)
- Q_c = conduction heat transfer between house and ambient (W)
- Q_m = maximum greenhouse heat loss (J/sec)
- Q_r = heat gain of greenhouse air due to solar radiation (W)
- R_a = atmospheric thermal radiation (W/m^2)
- R_A = atmospheric boundary layer resistance (sec/m)
- R_g = thermal radiation from ground (W/m^2)
- R_s = stomate resistance (sec/m)
- ${}_cR_s$ = increase in stomate resistance due to CO_2 effect (sec/m)
- ${}_mR_s$ = minimum stomate resistance (sec/m)
- S_{dif} = diffuse solar flux (W/m^2)

S_{dir} = direct solar flux (W/m^2)
S_R = solar heat gain through glass (W/m^2)
T = air temperature (°C)
T_o = reference temperature (0°C)
T_H = dry bulb air temperatures in greenhouse (°C)
T_L = leaf temperature (°C)
T_w = water temperature (°C)
T_{w1} = water temperature entering the pads (°C)
T_1 = air temperature entering the pads (°C)
T_∞ = ambient air dry bulb temperature (°C)
U_0 = overall transfer coefficient for greenhouse glass (W/m^2°C)
V = evaporative pad volume (m^3)
V_a = air velocity in the greenhouse (m/sec)
V_H = volume of air in the house (m^3)
W = leaf dimension perpendicular to wind velocity (m)
α = air absorptivity
α_s = plant leaf absorptivity for shortwave radiation
α_t = plant leaf absorptivity for longwave radiation
ε_L = plant leaf emissivity
ρ_s = reflectivity of leaf canopy
ρ_{sat} = water vapor mass density of saturated air (kg/m^3)
σ = Stefan-Boltzman constant
τ = time
ϕ = relative humidity
ω = specific humidity (kg H_2O/kg air)

INTRODUCTION

Successful production of crops in greenhouse systems depends upon modification of the greenhouse environment to provide suitable growing conditions for the crop. This environmental modification generally requires cooling the greenhouse air in summer and heating it in winter. The summer cooling needs of the greenhouse are usually satisfied using an evaporative cooling system. This system operates by having cool water drip through a porous packing while air is circulated through the packing by fans located at the rear of the greenhouse. Since the packing is a direct contact device, the combined mechanisms of heat and mass transfer generally result in the air stream being cooled and humidified because of its contact with the water stream.

The winter heating requirements have traditionally been satisfied by burning natural gas, propane, or some other fossil fuel. Since the cost for these fuels, which can comprise as much as half the greenhouse operating cost during a winter growing season [1][1] has risen greatly in the past few years [2], growers have begun to seek alternate heating methods and less expensive fuels. One such alternate heating method utilizes the cooling system for heating by supplying warm water to the pads and using a series of louvers to recirculate the air in the greenhouse.

Analytical investigations [3,4,5,6,7] have indicated that power plant reject heat, contained in the condenser cooling water effluent, is a technically and economically feasible source for the warm water.

[1] Numbers in brackets designate References at end of paper.

Experimental studies [8,9], performed as a part of a cooperative Oak Ridge National Lab (ORNL) — Tennessee Valley Authority (TVA) waste heat utilization effort, have confirmed the technical feasibility of the system. A small (7.3 m x 30.5 m) greenhouse has been in operation for several years using the evaporative pad heating concept. The horticultural results from this greenhouse indicate that tomato and cucumber yields in the waste-heat greenhouse are comparable to or exceed typical greenhouse yields. Thus, it appears that power plant reject heat can replace fossil fuels for greenhouse heating, resulting in an economic saving to the operator and a fuel saving to the nation.

This paper summarizes the calculational techniques used to analyze the evaporative-pad greenhouse system. Design techniques, used to size the evaporative pad system to meet cooling and heating demands are presented. Operational factors, which are important in maintaining a suitable greenhouse temperature, are also analyzed using a computational scheme to simulate the system performance.

System Description

As shown in Fig. 1, the system heats the greenhouse by pumping heated water into the evaporative pad water distribution system and allowing it to drip through the packing. As the water flows through the packing, it interacts with the air flow being drawn across the pad by the fans located at the rear of the house. Sensible heat transfer from the water stream acts to heat the greenhouse air stream.

Aspen fiber pads have traditionally been used for greenhouse cooling systems. Recent ORNL investigations [10], however, have indicated that CELdek pads are superior when the pads are used in a bimodal (heating and cooling) system. CELdek is a packing made of special cellulose paper impregnated with antirot salts, rigidifying saturants, and wetting agents. It is manufactured by Munters Corp., of Fort Myers, Florida, in sections having a depth of 0.6 m and a face area of 0.36 m^2 (1.2 m high x 0.3 m wide).

The bimodal evaporative pad system satisfies the total greenhouse environmental modification requirements by operating the shutter system in the following manner. In summer operation the inlet and exhaust shutters are opened, and ambient air is drawn into the pad where it is cooled and humidified. It is then cycled through the growing section and exhausted to the atmosphere.

In winter operation the inlet and exhaust shutters are closed to conserve heat. After the greenhouse air has exited the growing section, it is recycled through the attic back to the pad where it is reheated. Since the air is continually recycled, the humidity in the growing section hovers near 100%.

Alternate configurations for the pads and fans have been developed and will be tested in the TVA/ORNL demonstration greenhouse being constructed at TVA's Browns Ferry Nuclear Station. These designs are discussed in Ref. 11. The demonstration greenhouse is scheduled to begin operation in the summer of 1978. Since the greenhouse configuration shown schematically in Fig. 1 has been demonstrated, it will be used as the basis for design calculations in this report.

ENVIRONMENTAL CONTROL SYSTEM DESIGN PROCEDURE

The design of the pad and fan systems is based on several variables. Included are (1) the temperature of the warm water supplied to the pads; (2) the available warm water flow rate; (3) site weather conditions such as number of heating degree days and minimum winter ambient dry bulb temperature; (4) the desired temperature within the greenhouse; and (5) the greenhouse shape and size.

Using this information the environmental control system is designed to meet both the summer cooling and winter heating demands. These environmental modification requirements are then used to determine the size of the evaporative pad and the associated air flow system.

Since the required CELdek pad area will be the greater of that required to meet the cooling load or heating demand, the maximum heating and cooling loads must first be calculated. The winter heating load can be determined using any of several reference sources (examples can be found in Ref. 7 or 12).

The amount of CELdek required for heating is calculated by dividing the design heating load by the amount of heat transferred per unit face area. As shown in Fig. 2 (from Ref. 10), the heat transfer rate per unit face area increases with increasing water flow rate. The warm water flow used for the design depends upon several factors. Available supplies of warm water and water transportation costs will play an important role in determining the flow rate that is feasible. Additionally, vendor literature [13] indicates that the optimal water flow rate for cooling is about 3.2 m^3/sec/linear meter of pad. Therefore, a water flow rate for winter heating of 2.1 to 6.4 x 10^{-4} m^3/sec/linear meter would be desirable to increase the annual use factor for the pump system. Once the flow rate has been chosen, the heat transfer coefficient is read from Fig. 2.

The CELdek heat transfer coefficient, h_o from Fig. 2, is used with the temperature difference between the entering air temperature and the temperature of the warm water entering the pad. Therefore, the required CELdek face area for heating is given by

$$A_h = Q_m/h_o\ (T_{w1} - T_1) \qquad (1)$$

where Q_m is the maximum heating demand.

The amount of CELdek required for cooling (A_c) is calculated from vendor literature [13] and is shown for various degree days[2] in Fig. 3. The cooling requirements were based on a 70% shade factor and a 3.3°C air temperature rise across the greenhouse. Similar information is available from the vendor for other design conditions.

As mentioned previously, the amount of CELdek to be installed (A_p) is the greater of A_H and A_c.

Once the pad size has been determined, the pumping and fan systems can be sized. Because the water flow rate (L_w) is given as a function of the linear size of the pad, the design water flow rate depends upon the CELdek area (A_p) and the assumed water flow rate. Since the packing has a face area of 1.2 m^2 per linear meter, the design flow rate (L) is given by:

$$L = A_p \times L_w/(1.2\ m^2/m) \ . \qquad (2)$$

The pump and piping system can then be sized to deliver this flow rate. The pumping head required in the system is basically that required for the piping system.

The recommended air movement rate for the CELdek pad is one greenhouse air exchange per minute, although this may be varied somewhat to suit individual design needs. The size of the greenhouse, therefore, fixes the air flow rate (Q_a). The air velocity (V_a) through the CELdek is then given by

$$V_a = \dot{Q}_a/A_p \ . \qquad (3)$$

The pressure drop through the CELdek for velocities of less than 1.53 m/sec is about 0.012 kN/m^2 (from Ref. 10). For velocities in excess of 1.53 m/sec, the pressure drop is given by

$$PD = (9.64 \times 10^{-7}) \times V_a - 0.038 \ . \qquad (4)$$

Given the air flow rate and the pressure drop, an appropriate fan system can be selected. Typically the air movement system consists of two or four fans to provide for greater system reliability.

Therefore, once the greenhouse design has been fixed, the heating and cooling system can be designed in a straightforward manner using site weather conditions and the technique embodied in Eqs. 1–4.

SYSTEM SIMULATION

The design equations previously discussed are useful for sizing the heating and cooling system to meet the maximum expected loads. However, they do not provide information concerning operating conditions (air flow, water flow, amount of ambient bleed air required, etc.) during periods other than at the design point. To predict the required changes in operating conditions, models of the greenhouse and vegetation, which have been experimentally verified [3,6], were used to simulate the system's performance. The vegetation model incorporates a leaf energy balance to determine the air temperature that results in optimal plant growth. The greenhouse model, which includes heat and mass transfer aspects of the evaporative pad and the air flow through the greenhouse, is used to determine the operating conditions that will produce the required greenhouse air temperature.

Greenhouse Model

To predict the air conditions in the greenhouse, it is necessary to model the evaporative and sensible energy transport occuring in the pad.

The heat and mass transfer processes are described by the Merkel total heat approximation. It states that the total energy transferred equals the energy lost by the water stream which in turn equals the energy gain of the air stream. If small changes in air and water flow rates, due to evaporation, are neglected, this can be written as

$$(h_G a/C_H)(i_{Tw} - i)dV \simeq LC_L\ dT_w \simeq G\ di \qquad (5)$$

where C_H, the air humid heat capacity, is given by

$$C_H = C_p + \omega C_v \ .$$

In addition to the Merkel equation, the transfer equation governing the sensible heat transfer between the water and air stream is as follows:

$$C_H\ dT = h_G a(T_w - T)dV \ . \qquad (6)$$

From Eq. 5 and 6 the differential change in water temperature, air total enthalpy, and air temperature can be found as follows:

$$dT_w = \frac{(i_{Tw} - i)}{C_L} \frac{h_G a}{C_H} dV \qquad (7)$$

[2] The annual degree days are defined as the sum of all days in the year when the daily mean temperature falls below 18.3°C times the difference between the daily mean temperature and 18.3°C.

$$di = \frac{\left(i_{Tw} - i\right)}{G} \frac{h_G a}{C_H} dV \quad (8)$$

$$dT = \frac{\left(T_w - T\right)}{G} \frac{h_G a}{C_H} dV \quad (9)$$

The evaporative pad is treated as a crossflow packing, and the air temperature, air total enthalpy, and the water temperature are found with the aid of Eq. 7, 8, and 9 and a numerical integration scheme (14). This is achieved by subdividing the pad into elemental areas as shown in Fig. 4. Equations 7, 8, and 9 applied for the element in row I, column J yield:

$$dT_w(I,J) = \frac{[i_{Tw}(I,J) - i(I,J)]}{L(J)C_L} \frac{h_G a}{C_H} dV(I,J) \quad (10)$$

$$di(I,J) = \frac{[i_{Tw}(I,J) - i(I,J)]}{G(I)} \frac{h_G a}{C_H} dV(I,J) \quad (11)$$

$$dT(I,J) = \frac{[T_w(I,J) - T(I,J)]}{G(I)} \frac{h_G a}{C_H} dV(I,J) \ . \quad (12)$$

The numerical integration scheme parallels the physical system by applying the differential changes in water temperature down the column, and the differential changes in air total heat and temperature across the row:

$$T_w(I + 1,J) = T_w(I,J) - dT_w \quad (13)$$

$$i(I,J + 1) = i(I,J) - di \quad (14)$$

$$T(I,J + 1) = T(I,J) + dT \quad (15)$$

The water and air flows are assumed to divide equally among the four columns and eight rows, respectively. The pad packing is assumed uniform. Therefore, the area of air-water contact, a dV(I,J), is equal for all elements. Further, the water temperature for all elements of the first row is equal to the inlet water temperature, and the air state for all elements of the first column is that of the entering air.

The numerical integration scheme begins at element (1,1) where all the parameters are known. The water temperature for each successive element in the column is computed using Eq. 10 and 13 until the temperature of the water exiting from the packing has been found. The calculation scheme then proceeds to the next column where the air state has been altered by its interchange with the water stream in the previous column. The new air states are known, however, since Eq. 11, 12, 14, and 15 were applied across the row while the water temperature calculation was being performed. Again the calculations begin at the top of the column and follow the water flow down the packing. After the water and air stream calculations are made for the final column, the mixed mean water temperature, air temperature, and air total enthalpy are determined.

For calculations relating to the plant leaf temperature, it is necessary to know the relative humidity, rather than the air total enthalpy. This is found using the defining equation for air total enthalpy:

$$i = C_p\ (T) + \omega[C_V(T - T_o) + i_{vo}] \quad (16)$$

and solving for the specific humidity, ω. The saturation specific humidity is then determined and the ratio of the actual to the saturated specific humidity gives the relative humidity. It should be noted that in Eq. 16 the value for the specific heat of moist air C_V is taken as 1.926 J/kg°C.

Evaporative pad performance is generally reported as a relationship between a dimensionless mass transfer coefficient (KaV/L) and a dimensionless flow ratio (L/G). For CELdek pads used in a crossflow configuration, the manufacturer's data [13] indicates this relationship is given by

$$KaV/L = 1.444\ (L/G)^{0.726}\ . \quad (17)$$

For the pad performance calculations the overall mass transfer coefficient from Eq. 17 was converted to a heat transfer coefficient using

$$h_G aV = C_H\ KaV\ . \quad (18)$$

It is apparent from the above discussion that the air and water states entering the pad must be known in order to begin the calculations. The water temperature entering the pad is essentially fixed by the warm water source. The air state entering the pad during summer operation is essentially that of the ambient. However, during winter operations, when the house air recycles, the air conditions entering the pad are a function of the air state leaving the pad and the net heat loss or gain of the air as it cycles through the house.

The air state variation is accounted for by subdividing the greenhouse into eight sections, four in the growing section and four in the attic section. The air in each subsection is assumed to be perfectly mixed and at a constant temperature. The air is warmed by the solar input and exchanges energy with the ambient air by conduction through the greenhouse glass. The solar input is a function of time [5] because of the time dependence of the solar flux, S_R, and the area through which the solar flux enters the house, A_S.

$$Q_R = \alpha S_R A_S\ . \quad (19)$$

The air absorptivity is obtained by computing the gas emissivity and using Kirkchoff's Law to equate absorptivity and emissivity. The mean beam length (M) used in the determination of emissivity is given by the ratio of volume of air in the house (V_H) over the house area in contact with ambient environment [15]; i.e., $M = 0.9\ (4V_H/A)$. Using the computed value of M, an average air temperature and relative humidity, and the partial pressure of water vapor, the air absorptivity can be computed. The solar heat gain through ordinary glass has been tabulated for various latitudes, time of year, and time of day.[3]

The transfer of energy due to conduction is found by equating the convective transfer on the inner and outer surfaces of the glass with the conduction transport through the glass as follows:

$$Q_c = h_1(T_H - T_1) = \frac{k}{Y}\ (T_1 - T_2) = h_2(T_2 - T_\infty) \quad (20)$$

Equation 20 is rearranged as follows:

[3] For example, see Carrier Air Conditioning Company. 1969. Carrier system design manual, Part 1-Load estimating. Syracuse, N.Y.

$$Q_c = \frac{1}{1/h_1 + 1/h_2 + 1/k/Y}(T_H - T_\infty) \tag{21}$$

$$= U_0(T_H - T_\infty) .$$

The overall transfer coefficient, U_0, for daylight hours is 6.48 $W/m^2°C$. During the night hours thermal radiation from the glass to the night sky is accounted for by augmenting h_2 by an effective coefficient for radiation on the order of 1.17 $W/m^2°C$, thus increasing U_0 to 7.65 $W/m^2°C$.

The total heat gain or loss by the air in the subsection is given by the sum of the two transfer modes as follows:

$$Q = Q_c + Q_R \tag{22}$$

$$= \alpha S_R(\tau)\ A_s(\tau) + U_0(\tau)[T_H(\tau) - T_\infty(\tau)]A_L \tag{23}$$

The temperature of the air changes as it moves from one subsection to another because of this net heat transfer. Since the air in the house is essentially saturated for winter operation, any net heat loss from the subsection results in an air temperature decrease and condensation of an appropriate amount of water vapor given by

$$Q = \dot{m}_a\ C_H(T_H - T_H') + \dot{m}_a\ H(\omega_{sat_{in}} - \omega_{sat_{out}}) . \tag{24}$$

Since ω_{sat} is a known function of temperature, Eq. 24 has T_H', the air temperature leaving the subsection, as the only unknown.

The calculations begin in section 1 where T_H is the pad air exit temperature. The heat loss from subsection 1 is calculated using Eq. 23. Then Eq. 24 is used to calculate T_H', the air temperature entering the next subsection. This procedure is used for subsequent sections until T_H' for the last section is computed. This air temperature, and corresponding specific humidity, are the air state entering the pad.

The air flow rate in the house for each time period is the flow, within the capability of the house fans, which will optimize the vegetation of leaf temperature.

Leaf Energy Budget

To determine the leaf temperature, the dominant modes of energy transport must be identified and described analytically. Radiation will be an important mode of energy transport because the leaf absorbs solar radiation to provide the necessary energy for the photosynthetic process. Because the air and the leaf will generally have different temperatures and an air flow is present, due to the greenhouse fans, convection heat transfer will also occur.

By studying the physiology of the leaf it becomes apparent that evaporative transport of energy also contributes significantly to the net exchange of energy between the leaf and its surroundings. Figure 5 indicates that the evaporation of water vapor from the leaf interior to the ambient takes place through small openings in the epidermis known as stomate openings. The guard cells of the stomate expand and contact to provide the opening necessary for transpiration to take place. This process can be represented by a resistance network, adapted from Lake [16], as indicated in Fig. 5. In this analogy the driving force is the concentration of water vapor and the resistances represent the boundary layer, R_A, the stomate opening, R_s, and the epidermal layer R_E.

An energy balance for the plant leaf equates the radiative gain with the radiative output, the convective contribution, and the energy transport due to transpiration [17] as follows:

$$q_{ABS} = q_{RAD} + q_{CONV} + q_{TRANS} . \tag{25}$$

The absorbed radiative energy is composed of shortwave solar and longwave thermal radiation. The average (per leaf surface) radiative gain is given by

$$q_{ABS} = 1/2\ [\alpha_s(1 + \rho_s)(S_{dir} + S_{dif}) \tag{26}$$

$$+ \alpha_t(R_a + R_g)] .$$

Reasonable values for α_s, ρ_s and α_t are, respectively, 0.70, 0.18 and 0.95.

Substituting the appropriate transfer expressions, Eq. 25 becomes

$$q_{ABS} = \varepsilon_L \sigma T_L^4 + h(T_L - T) + H\ \dot{m}_{EVAP} . \tag{27}$$

Because the leaf is in a controlled environment, it is reasonable to expect that sufficient irrigation is provided to insure the absence of water deficit in the plant. With no water deficit the air within the substomate cavity is saturated at the leaf temperature. Using the circuit analogy described in Fig. 5, the evaporation rate, $\dot{m}_{EVAP}$, is given by

$$\dot{m}_{EVAP} = \frac{\rho_{sat}(T_L) - \phi\rho_{sat}(T)}{R_A + R_S} \tag{28}$$

where the effects of the epidermal resistance, R_E, are assumed to be small and not included.

The air boundary layer resistance to water vapor transport, R_A, and the convection coefficient are given by

$$h = k_1(V_a/D)^{1/2} \tag{29}$$

$$R_A = k_2(W^{0.2}\ D^{0.35}/V_a^{\ 0.55}) \tag{30}$$

where the empirical coefficients, k_1 and k_2, are prescribed by Gates [17] from wind tunnel experiments with leaf models.

The effects of light intensity and carbon dioxide concentration on stomate resistance have been previously documented by Waggoner [18]. Incropra [19] has extended this model to include the thermal effects on stomate resistance so that the stomate resistance to water vapor diffusion is given by

$$R_s = 0.64 \left\{ {}_mR_s + {}_cR_s \frac{[CO_2]_s}{[CO_2]_s + K_{SC}} \right\} \tag{31}$$

$$\left\{ \frac{K_{SL} + L_{PHAR}}{L_{PHAR}} \right\} \left\{ \frac{K_{SW} + w(T_L)}{w(T_L)} \right\} .$$

Reasonable values for ${}_mR_s$, K_{SC}, and K_{SL} are, respectively, 50 sec/m, 100 sec/m, 300 ppm, and 34.8 W/m^2. The variation of stomate width with leaf temperature, w(T), is known for some species [20] and a reasonable value for K_{SW} is 3 μm [21].

When all of the environmental variables have been specified, Eq. 25 becomes a fourth order polynomial.

with the unknown being the leaf temperature. Since R_s is a function of leaf temperature, through $w(T_L)$, the solution involves an iterative procedure. Since the air flow rate in the house influences the air temperature (t), relative humidity (ϕ), and air velocity (V), varying the air flow rate can produce a leaf temperature that is as close as possible to its optimal value.

SAMPLE CALCULATIONS

To illustrate the use of the method described, a sample case was calculated for the Cleveland, Ohio area. This area was chosen because there is a relatively large greenhouse industry in this vicinity.

It was assumed that warm water at 29.4°C was available to heat a 2.5 ha greenhouse complex. A greenhouse temperature of 18.3°C and an ambient temperature of −17.8°C were used in computing the maximum heat demand. Using the technique outlined in Ref. 7, a value of 16 MW was calculated for the peak heat loss.

Since the air is saturated during winter operation, much of the heat loss results in condensation of water vapor. Using the maximum heat loss, the design greenhouse air flow, and accounting for water condensation yields a temperature drop of about 2.8°C for the air cycle through the greenhouse. This results in an air temperature of 15.5°C entering the pad. Using a water flow rate of 3.1×10^{-4} m^3/sec/m yields (from Fig. 2) a heat transfer coefficient of 937 w/m^2 − °C. Using these values in Eq. 1 yields a pad area of 1227 m^2 required for heating. Since the area has 6351 degree days, the pad area required for cooling (from Fig. 3) is 725 m^2. The installed pad area is thus governed by the greenhouse heating requirements.

To illustrate the operational analysis technique, Eqs. 5 through 31 were coded for the computer and the greenhouse air flow and vegetation leaf temperature computed for a typical March day. It was assumed that air flow rates between 1027 and 3090 kg/sec were possible, while the water flow was fixed at 0.312 m^2/sec. Figure 6 indicates the ambient dry bulb and solar flux variation throughout the day. Since the temperatures are above the winter design temperatures, it is possible that ambient bleed air would be required. For this example ambient bleed air flows of 20% of the total air flow are possible. The average leaf properties cited in the text were used for the computations. It was assumed that a cool season crop (such as cucumbers) was planted in the greenhouse and its optimal leaf temperature was 20°C. The air flow in the greenhouse was, therefore, varied in an effort to maintain a 20°C leaf temperature for the crop.

DISCUSSION OF RESULTS

As shown in Fig. 7, proper modification of the greenhouse air temperature results in leaf temperatures throughout the day that are within 2°C of the leaf optimal growth temperature. Examination of the temperature profiles in Fig. 7 and the solar flux profile in Fig. 6 indicates the relationship between the various modes of energy transport involved in the leaf energy budget. Since the air is nearly saturated, leaf transpiration is negligible. The major components of the energy budget, therefore, are solar heat gain and heat transfer from the air to the leaf via conduction. During the hours of darkness or low solar flux, an air temperature about 5 to 6°C above the leaf temperature is required to provide sufficient thermal input to the vegetation. However, when the solar flux is sufficient to make a significant thermal contribution, the difference between the air and leaf temperature decreases to about 2°C.

Since the greenhouse heating system was sized to maintain a greenhouse temperature of 18.3°C, it is not capable of maintaining greenhouse temperatures of 26°C during the cold early morning hours. This results in the leaf temperature falling below its optimal value for this time period. The leaf temperature, however, never drops more than 2°C below the optimal growth temperature. Since the growth-temperature curve for most vegetation has a 1 or 2°C plateau around the optimal temperature, this drop in leaf temperature does not significantly affect plant growth.

As seen in Fig. 7, the leaf temperature exceeds its optimum value during the hours of peak solar flux. Ambient bleed air, totaling 20% of the total greenhouse air flow, is used between the hours of 1000 and 1400 to reduce the air temperature and subsequently the leaf temperature. If larger ambient bleed air flows were included in the analysis, it would have been possible to reduce the greenhouse air temperature even further. This would have yielded a leaf temperature closer to the optimum. As previously discussed, however, a 1 or 2°C fluctuation from the optimal growth temperature does not significantly affect plant growth. Thus, the leaf temperature profile indicated in Fig. 7 should result in near optimum growth for the vegetation.

As stated previously, the greenhouse air temperature was modified by controlling the greenhouse air flow rate. The air temperatures plotted in Fig. 7 result from operating the greenhouse according to the air flow pattern illustrated in Fig. 8. During the cold night hours the air flow is maintained at its maximum level. Although the higher air flow results in a slight decrease in the overall pad transfer coefficient, the large volume of air circulating in the greenhouse results in a minimum drop in air temperature during the hours of peak heat loss. This effect dominates the loss in pad heat transfer ability and results in the slowest rate of temperature drop in the greenhouse.

Since lower air temperatures are required during hours of significant solar flux, the maximum air flow rate is again chosen to take advantage of the reduced heat transfer capability of the pad. As noted previously, the air temperature is further modified by venting some of the warm greenhouse air to the atmosphere and replacing it with cooler ambient air.

During the late afternoon hours the air flow is adjusted to provide a rapid increase in air temperature. This compensates for the decrease in solar flux input. Similarly, during the late morning hours, the air flow is adjusted to slow the rise in temperature to compensate for an increasing solar contribution.

The leaf temperature results indicate that, in addition to being useful for operational considerations, the leaf energy budget model could be a useful design tool. Once the crop has been selected, the leaf energy budget could be used to determine the required greenhouse air temperature. The greenhouse heating system could then be sized to maintain this temperature. Alternately, the leaf energy budget model could be coupled to a photosynthetic model to determine the effect lower leaf temperatures would have on plant growth. This model could be used in conjunction with the greenhouse model and appropriate economic data to determine the optimal greenhouse design and operating conditions.

CONCLUSIONS

Crops grown in an evaporative pad greenhouse can be maintained within their thermal comfort zone if the air stream flow is properly adjusted. In general, solar heat gain and conduction heat transport between the leaf and the air are the dominant modes of heat transfer. Air temperatures 5 to 6°C above the leaf temperature are required during darkness or low solar flux hours while lower air temperatures are required when the solar flux component is significant.

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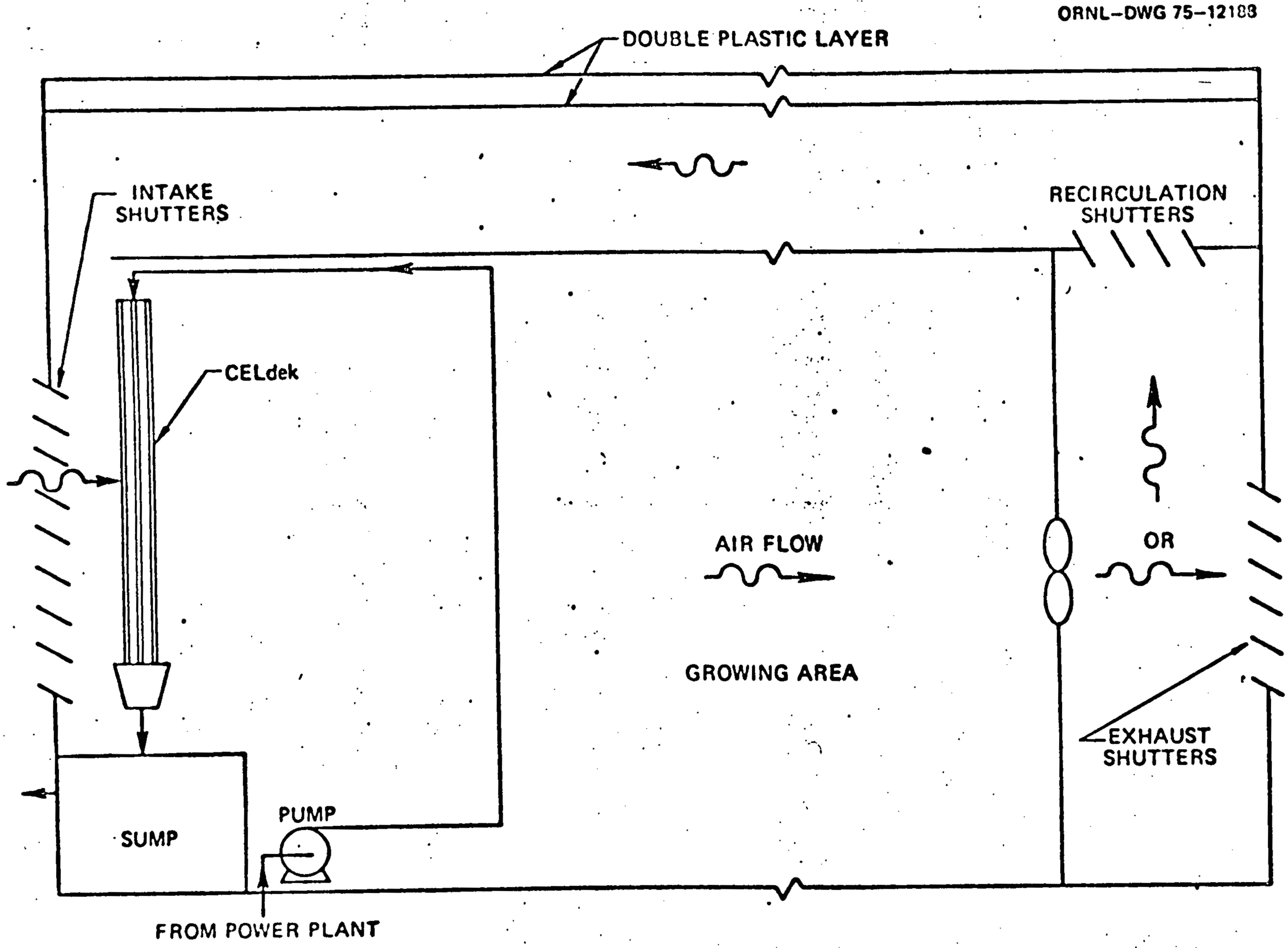

Fig. 1. Scematic of greenhouse using reject warm water heating system

June 15
4435-0625

Put units in metric

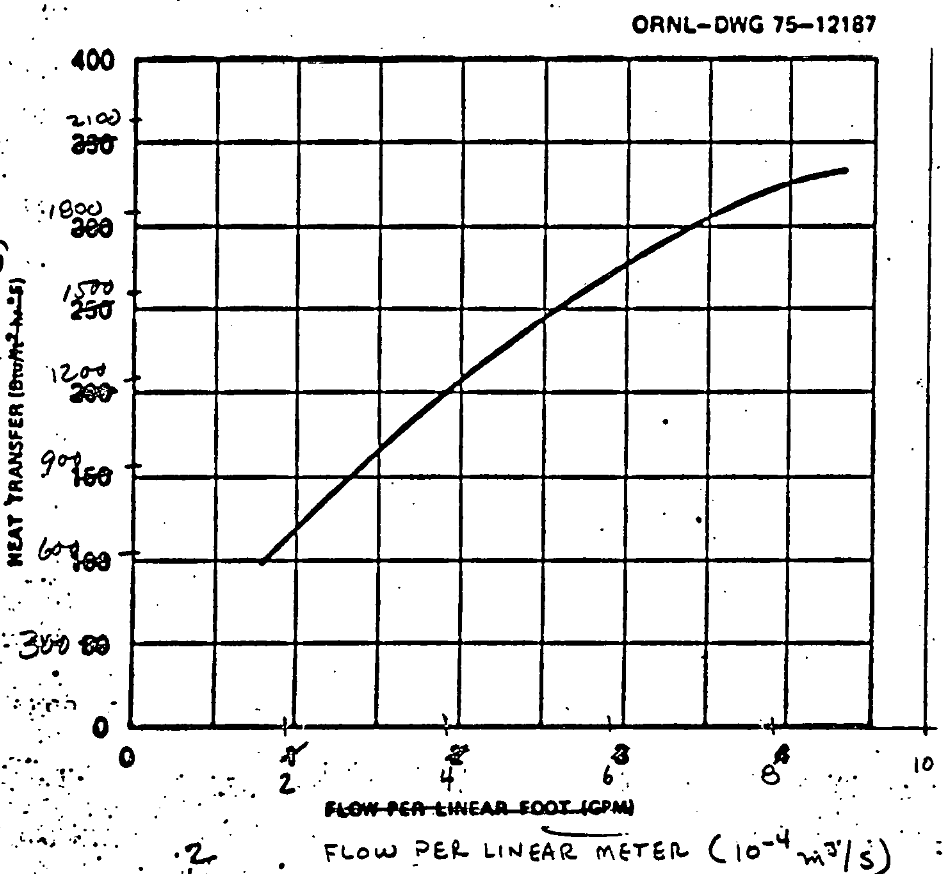

Fig. ~~8~~ 2. Heat transfer rate of CELdek.

Fig. 3 Required CELdek area for summer cooling for 2.5 ha greenhouse

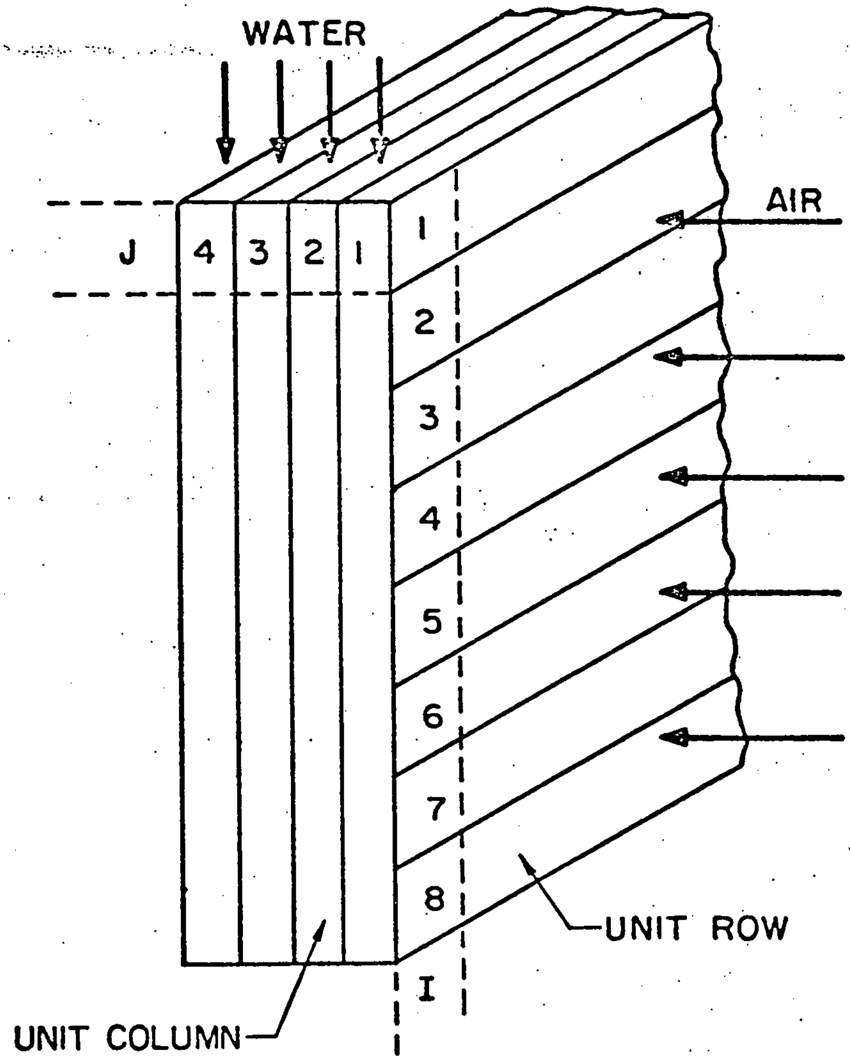

FIGURE 5 CROSSFLOW SCHEMATIC FOR EVAPORATIVE PAD

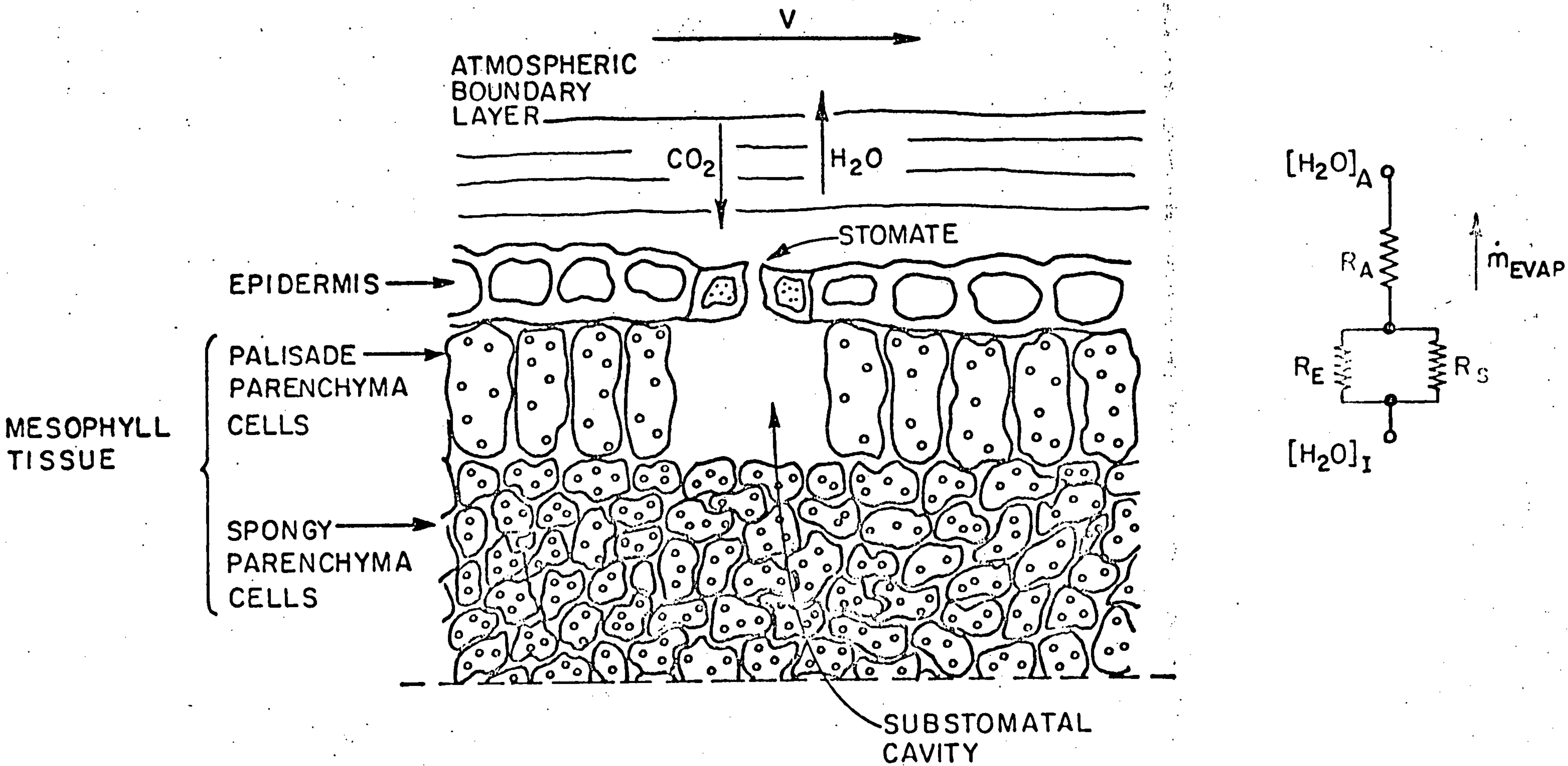

FIGURE 5 CUTAWAY VIEW OF TYPICAL LEAF WITH ELECTRIC ANALOG

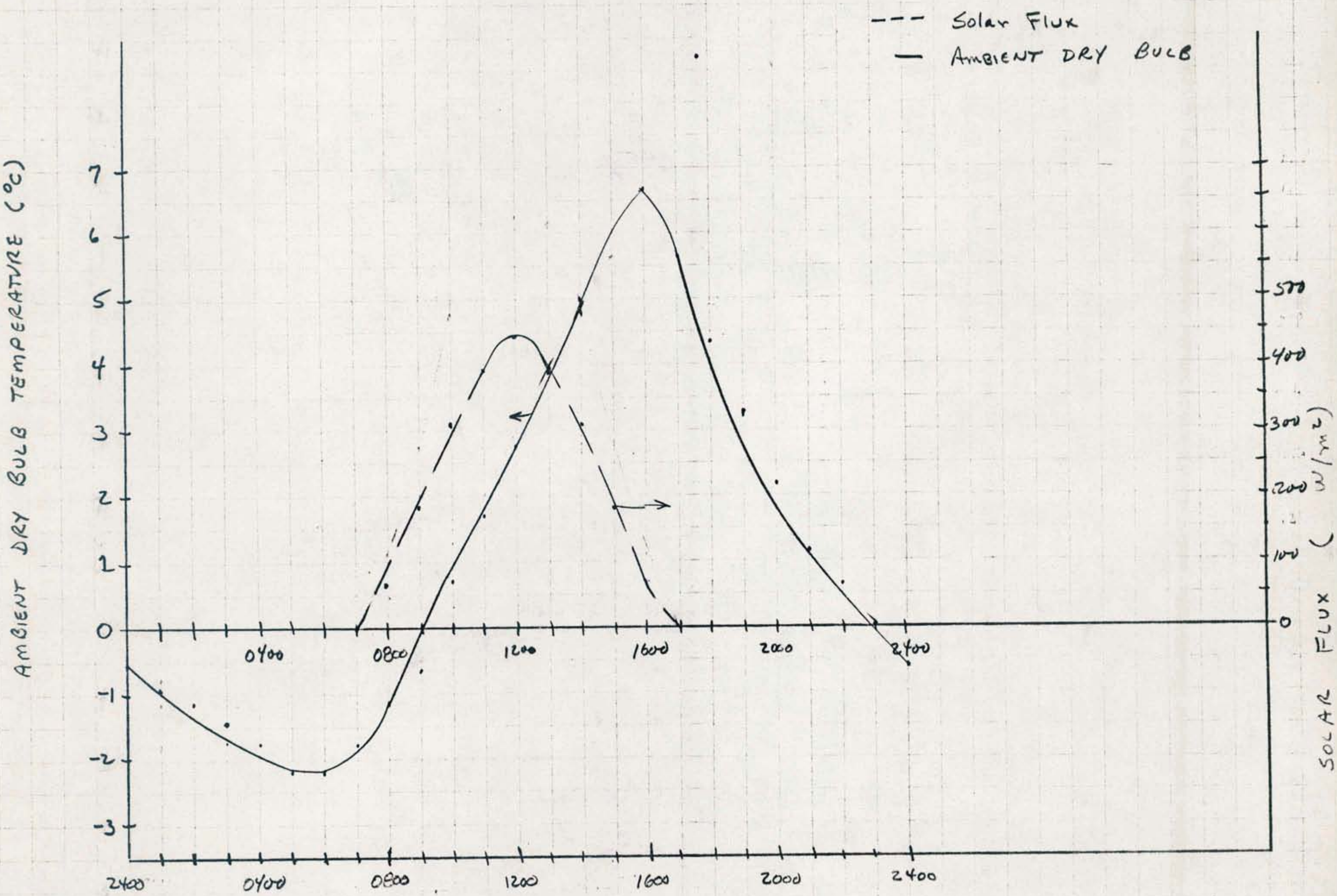

FIG 6. MARCH CLIMATIC DATA FOR Cleveland, Ohio

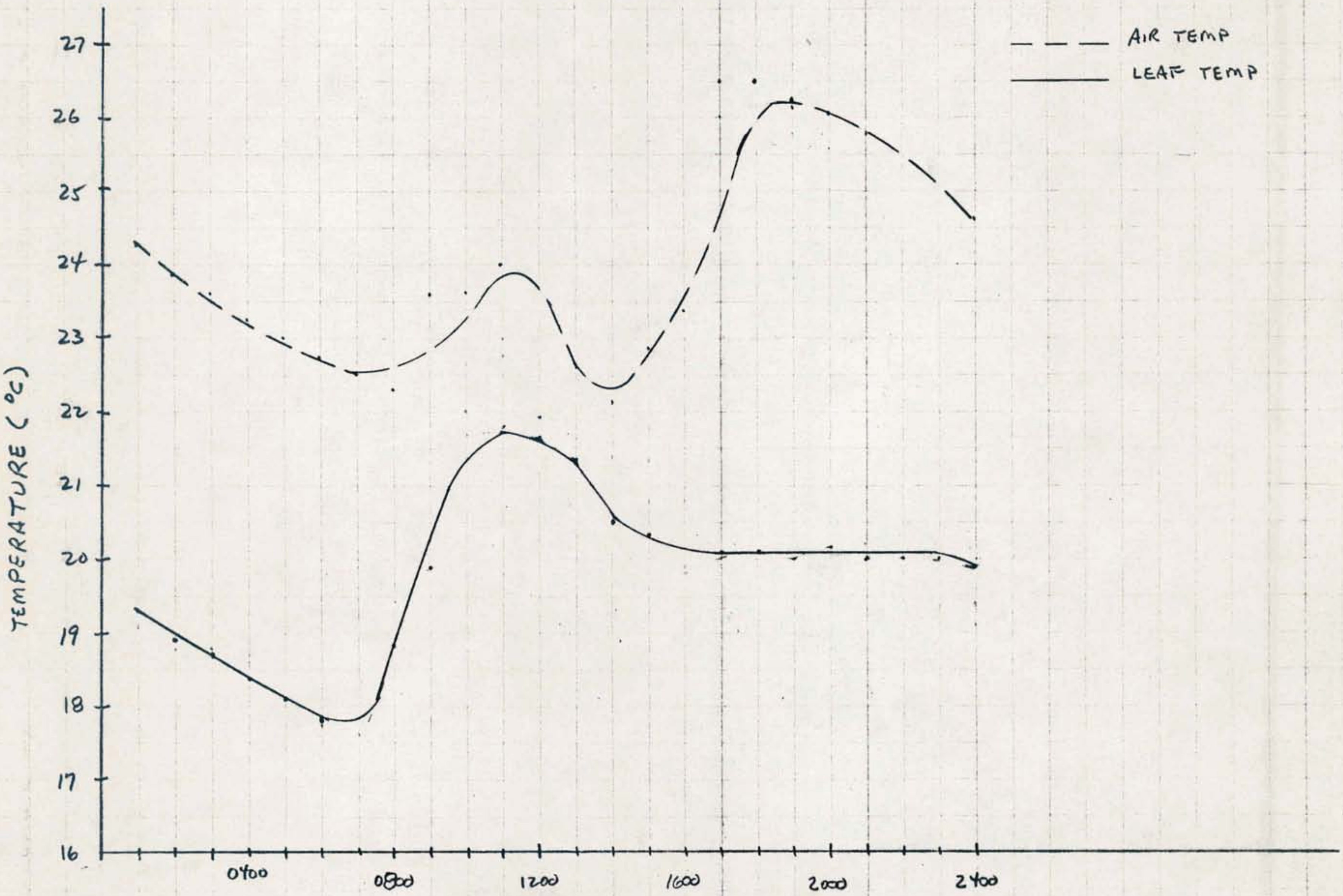

Fig 7. LEAF AND GREENHOUSE AIR TEMPERATURE PROFILES

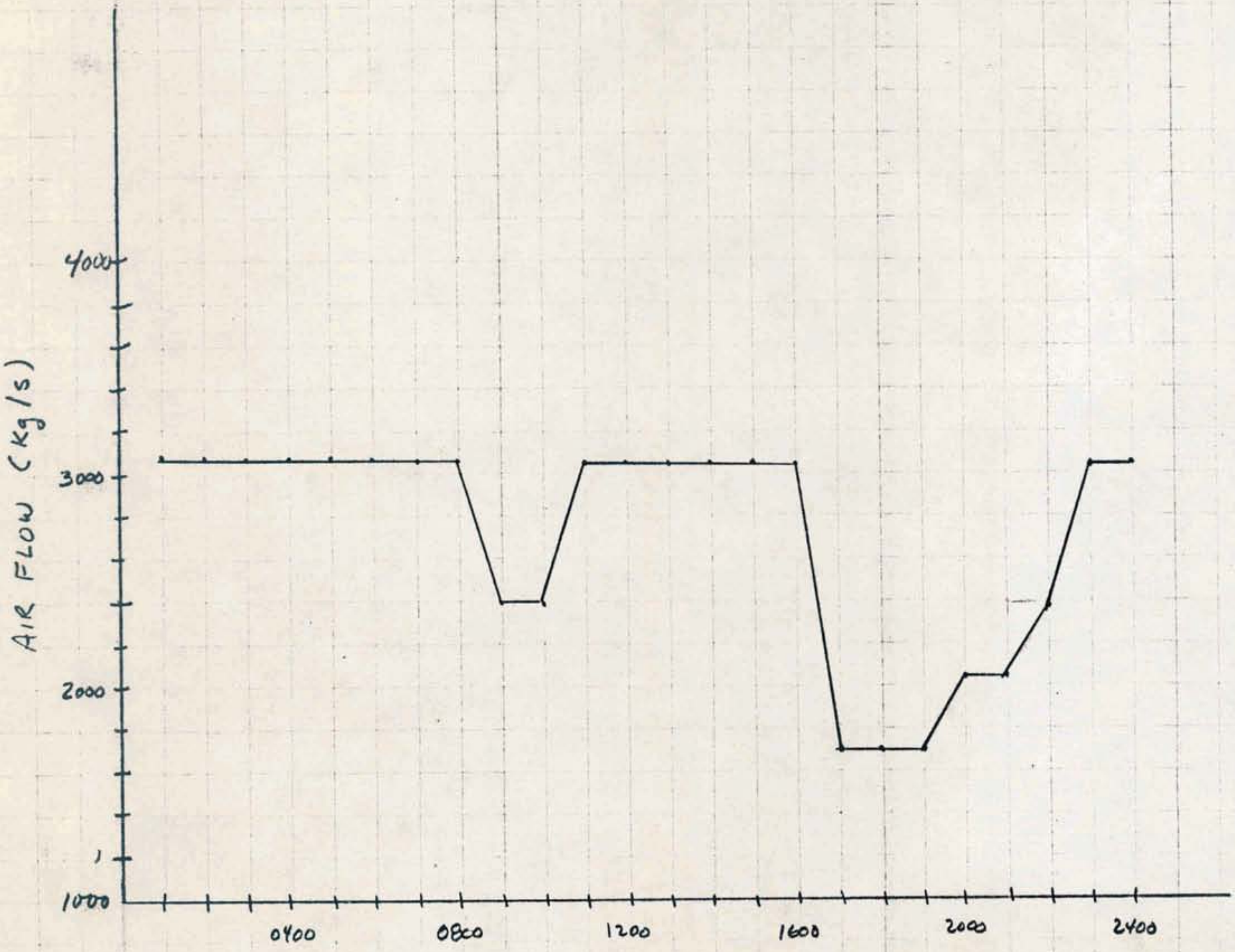

FIGURE 8. REQUIRED GREENHOUSE AIR FLOW PATTERN