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AN EVAPORATIVE PAD GREENHOUSE MODEL FOR WASTE HEAT UTILIZATION ASSESSMENT*

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ABSTRACT

This is the third in a series of articles on the development of a methodology for assessing waste heat utilization technologies and optimizing the mix of technologies used on a site-specific basis. As part of this effort a model of an evaporative pad heating greenhouse has been developed. This model uses readily obtainable data regarding plant kinetics and climate conditions. The outputs include the time to harvest, crop yield, and mass flow rates of heated water in and out of the greenhouse required to maintain the desired temperature. This model has been used in simulations in order to assess the economic feasibility of this method of utilizing waste heat.

Greenhouses represent one of the potential technologies available for waste heat utilization. As part of a continuing effort to develop a procedure to enable decision makers to select the optimum mix of waste heat utilization technologies, a model has been developed to describe the thermal behavior of greenhouses [1-3]. The objective is to predict the flow of heated water required to maintain the optimum growth of selected plant species. In this article, we present both a

* This is the third in a series of articles on the development of a methodology for assessing waste heat utilization technologies and optimizing the mix of technologies used on a site-specific basis. The first article provided an overview of the methodology, and the second dealt with the aquaculture model [*JES, 19*(2), pp. 95-115 and pp. 115-134]. Later contributions will describe models for simulating the surface heated greenhouse, crop drying, and wastewater treatment components of an integrated waste heat utilization complex.

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http://baywood.com

brief review of the pertinent literature and the details of the model development for the case of evaporative pad greenhouses.

LITERATURE REVIEW

The 10-18.3°C minimum night time temperatures required for growth in greenhouses can be maintained by most condenser effluents, and prospects are particularly attractive for closed-cycle plants [4]. One low-cost system distributes warm water at 30-35°C through flat plastic sleeves lying between greenhouse plant rows, maintaining interior air temperature above 9°C even during exterior air temperature lows of –11°C. The system, which is located in France, successfully produces vegetables and flowers in 9 ha of greenhouses [5].

A 7.3 by 30.5 m glass-glazed greenhouse using an evaporative pad and forced air circulation system for heat transfer was tested as part of an ambitious demonstration of waste heat utilization by the Tennessee Valley Authority. Warm water flowed through the 7.3 by 1.2 m pad of a patented material called CELdek, which was 30.5 cm thick. Both heating and cooling modes were possible. By recirculating saturated air through an attic and back to the pad, heat could be transferred to the air with no evaporative cooling (Figure 1). This air would be heated further by a finned-tube heat exchanger which would also lower the humidity of the air as it entered the growing area. By opening a vent at either end and forcing nonsaturated exterior air through the wet CELdek pad on its way

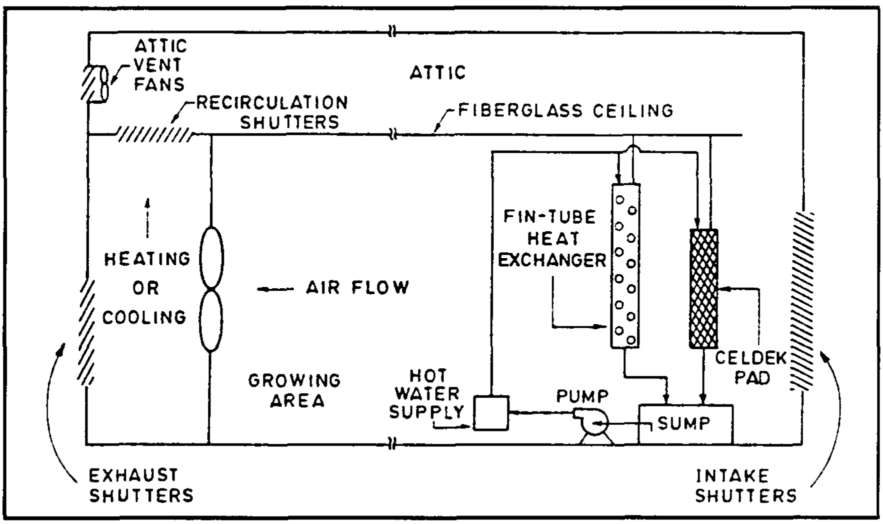

Figure 1. Schematic drawing of waste heat research greenhouse, Muscle Shoals, Alabama [6].

to the growing area, a net cooling due to evaporation could be sustained. The higher capital costs of the air circulation and heat transfer equipment needed for a waste heat greenhouse would be recovered in 6.5 years by avoidance of the fuel costs necessary in a conventional greenhouse [6].

A low-cost greenhouse developed at Rutgers University is heated by warm water which flows through the porous rock floor after passing through an aquaculture raceway. With floor and air heat exchangers, as well as a black polyethylene night time cover, interior temperatures of 14.4°C were maintained even when exterior temperatures fell below –17.8°C. Unlike the TVA system, no fans are required; thus, much less electricity would be used. Preliminary economic analysis indicates a payback period of under one year, although waste heat delivery costs were not included [7].

At least three commercial greenhouse operators receive waste heat from Northern States Power Company's Sherburne County Plant. Both flowers and vegetables are grown in arch-roofed plastic greenhouses. Dry finned-tube heat exchangers and conventional air handling units are able to maintain 13-16°C greenhouse temperatures even when outside temperatures fall to –42°C. Pipelines connecting both generating units to the greenhouses provide 97 percent service reliability [8].

A variety of greenhouse designs have been discussed in the literature. The two basic types which have the most attention are the evaporative pad and the surface heated greenhouses. Evaporative pad-heated greenhouses use direct-contact heat exchangers. Air is forced through spongy pads which are constantly supplied with warm water. Surface-heated greenhouses use a film of warm water trickled over the outside of the greenhouse, to heat the interior. The purpose of the research described here has been to develop quantitative models to thermally describe the greenhouse. The general approach for the evaporative pad and surface heated greenhouses has been the same. In each case, a materials balance and a heat balance are utilized. However, the details are sufficiently different to justify segregation of the two models. Therefore, the surface heated model is described elsewhere [1, 9] and the evaporative pad-heated greenhouse model is presented in the following paragraphs.

EVAPORATIVE PAD MATERIALS BALANCE

Iverson, Puttagunta, Meek, and Chisholm give dimensions of a standard greenhouse with approximately an acre of growing area [10]. This provides a pattern for the greenhouse floor plan which we will use as a single module (Figure 2). Any greenhouse complex will consist of groups of these greenhouse modules. The greenhouse module consists of twenty bays of dimensions 24 ft x 94 ft (7.3 m x 28.65 m), connected by a main access aisle measuring 12 ft x 240 ft (3.7 m x 73 m).

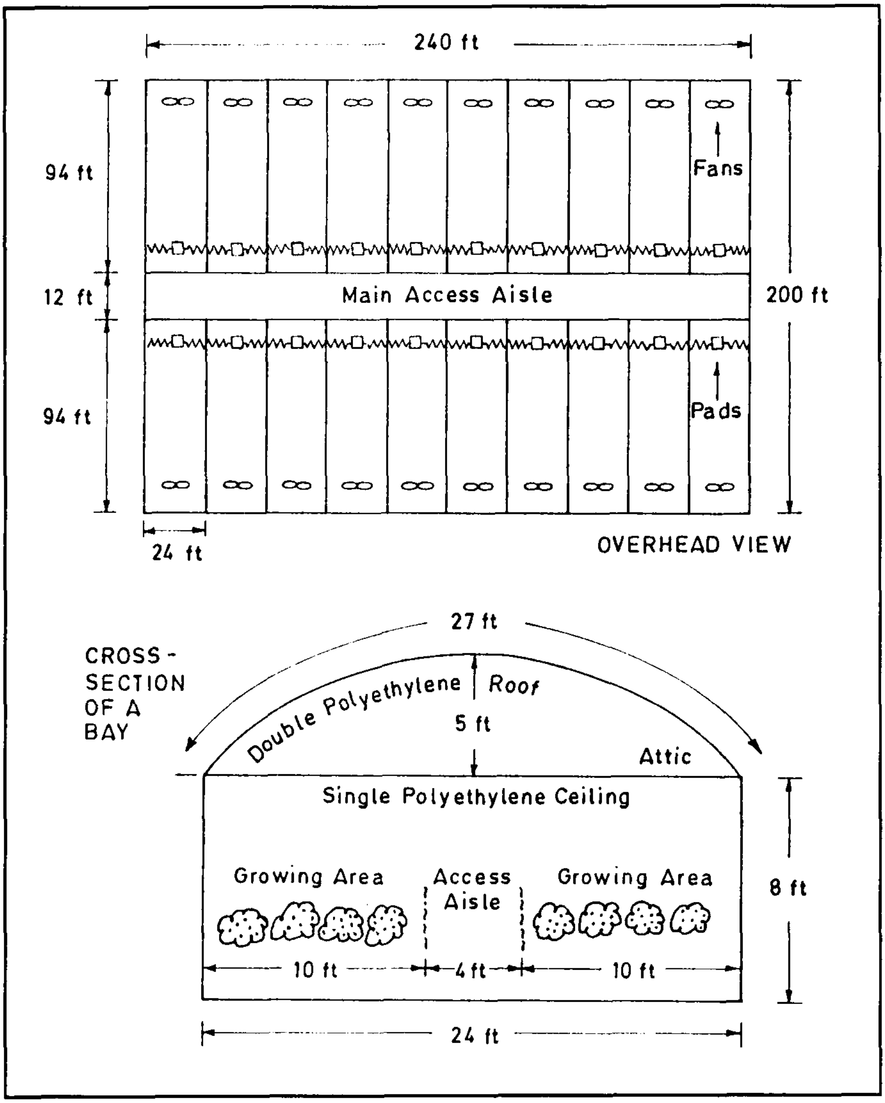

Figure 2. Single module of evaporative pad greenhouse.

We are mainly concerned with predicting the output of produce (flowers or vegetables). While there are inputs such as water, seedlings, and fertilizer, these are generally applied on a fixed schedule which makes their prediction trivial.

The growth of the vegetables and flowers will be represented by the logistic equation:

$$\frac{dX}{dt} = rX\left(1 - \frac{X}{K}\right) \tag{1}$$

where:

X = quantity of plant biomass, lb
t = time, hr
k = carrying capacity, lb
r = intrinsic rate of natural increase, hr^{-1}

The factor $\left(1 - \frac{X}{X}\right)$ represents crowding. In the absence of crowding, growth would proceed at a rate rX, which would be exponential growth.

Equation (1) may be integrated to obtain an expression for X in terms of t, or vice versa. The limits of integration are from time = t_o to time = t and from X = X_o to X = X_t. By separation of variables:

$$\int_{X_o}^{X_1} \frac{dX}{X\left(1 - \frac{X}{K}\right)} = r\int_{t_o}^{t} dt = rt \tag{2}$$

We use partial fractions on the left-hand side to integrate:

$$\int_{X_o}^{X_1} \left[\frac{1}{X} + \frac{\frac{1}{K}}{\left(1 - \frac{X}{K}\right)}\right] dX = rt \tag{3}$$

and obtain

$$\ln\left[\frac{X_t}{\left(1 - \frac{X_t}{K}\right)}\right] - \ln\left[\frac{X_o}{\left(1 - \frac{X_o}{K}\right)}\right] = rt \tag{4}$$

Let

$$a = \ln\left[\frac{X_o}{\left(1 - \frac{X_o}{K}\right)}\right] \tag{5}$$

and note that

$$\ln\left[\frac{X_t}{\left(1-\frac{X_t}{K}\right)}\right] = \ln\left[\frac{KX_t}{K-X_t}\right] \tag{6}$$

Therefore, equation (4) can be expressed as:

$$\ln\left[\frac{KX_t}{K-X_t}\right] = rt + a \tag{7}$$

or,

$$\frac{KX_t}{K-X_t} = e^{(rt + a)} \tag{8}$$

Solving for X_t, we obtain

$$X_t = \frac{K}{1 + Ke^{-(rt + a)}} \tag{9}$$

Defining $C = e^{-a}$, equation (9) may be re-expressed as

$$X_t = \frac{K}{1 + CKe^{-rt}} \tag{10}$$

Note that the limit of X_t as t increases is K and so X_t approaches the carrying capacity, K, as it should. Re-arranging equation (7), we have an expression for t as a function of X_t:

$$t = \frac{\ln\left[\frac{KX_t}{K-X}\right] - a}{r} \tag{11}$$

If we use X_o as the planting weight, and define a harvest weight as X_t:

$$X_{harv} < K \tag{12}$$

then we can calculate the time-to-harvest:

$$t_{harv} = \frac{\ln\left[\frac{KX_{harv}}{K-X_{harv}}\right] - \ln\left[\frac{KX_o}{K-X_o}\right]}{r} \tag{13}$$

where

t_{harv} = time of harvest, hr
X_{harv} = weight at time of harvest, lb
X_o = weight at time of planting, lb

This can be simplified as.

$$t_{harv} = \frac{1}{r}\ln\left[\frac{X_{harv}(K - X_o)}{X_o(K - X_{harv})}\right] \tag{14}$$

The dependence of the growth rate on temperature is discussed by Yarosh, et al. [11]. The growth curves are approximated by:

$$r = r_{max}\sin\left[\pi\left(\frac{T - T_L}{T_H - T_L}\right)\right] \tag{15}$$

where

r_{max} = maximum growth rate, hr^{-1}
T = temperature at which growth occurs, °F
T_H = upper zero-growth temperature, °F
T_L = lower zero-growth temperature, °F

Note that π should be replaced by 180 if the sine is taken using degrees instead of radians.

The maximum growth rate (r_{max}) is dependent upon the amount of sunlight available. Above some saturation intensity, most of the additional light is wasted, and so growth at moderate and high intensities is about the same. The effective light available to a plant is [12]:

$$I_{eff} = I_s\left[\ln\left(\frac{I_i}{I_s}\right) + 1\right] \tag{16}$$

where

I_{eff} = effective light intensity, foot-candles
I_s = saturation light intensity, foot-candles
I_i = incident light intensity, foot-candles

We can convert total insolation to mean light intensity using [12]:

$$\bar{I} = (2.71)S \tag{17}$$

where

S = total insolation, Btu-ft^{-2}-d^{-1}
$\bar{I}$ = mean light intensity, foot-candles.

The yield data upon which the growth calculations are based assume "normal conditions." For the United States, we can assume 400 Langleys/d [13]. This is equivalent to 1476 Btu-ft^{-2}-d^{-1}, or a mean light intensity of 4000 foot-candles. Saturation is reached at about 10 percent of full sunlight, so we can take 400 foot-candles as the saturation light intensity [14]. This is consistent with values reported by Oswald and Gotaas [12].

We assume that the intrinsic growth rate is proportional to the effective light intensity. We calibrate this so that "normal growth" occurs at the typical mean light intensity of 4000 foot-candles:

$$r_i = 0.3\left[\ln\left(\frac{I_i}{400}\right) + 1\right] r_o \tag{18}$$

where

r_o = growth rate under normal light conditions, hr^{-1}
r_i = growth rate under I_i light conditions, hr^{-1}

Finally, we recall that there is glazing (glass or polyethylene) covering the growing area, which cuts down on the actual amount of light received by the plants. Combining equations (17) and (18), we obtain:

$$r_s = 0.325 r_o \left[\ln\left(\frac{2.71\tau S}{400}\right) + 1\right] \tag{19}$$

where

r_s = growth rate under τS light conditions, hr^{-1}
τ = solar radiation transmittance, decimal.

From Yanda and Fisher [15], $\tau = 0.73$ for triple polyethylene.

In practice, we take the value obtained for r in equation (15) and use this as our value for r_o in equation (19) to obtain a corrected value of r_s to use as the intrinsic growth rate in the model described by equations (1) through (14). In Table 1, the data and parameters needed to apply this growth model to tomatoes and hybrid tea roses are summarized. Other input requirements include site-specific weather data (S), the known constant, τ, and the design parameter, T.

HEAT BALANCE FOR EVAPORATIVE PAD GREENHOUSE

The frame of the evaporative pad greenhouse is covered by a double layer of polyethylene; a single layer of polyethylene forms an attic. The air is circulated through the attic by ventilation systems, and so that attic air is at approximately the same temperature as the growing area.

J. N. Walker gives the following heat balance [20]:

$$H_f + H_s + H_e + H_r = H_c + H_t + H_{PH} + H_g + H_V \tag{20}$$

Table 1. Plant Growth Data and Parameters

Property	*Hybrid Tea Roses* *[16, 17]*	*Tomatoes* *[16, 18, 19]*
T_L, °F	44	40
T_H, °F	90	100
K	115000 blooms/acre	120000 lb/acre
r_{max}, hr^{-1}	0.11984	0.0444
t_{harv}, hr	1200	3504
X_{harv}	89507 blooms/acre	92000 lb/acre
X_o	5595 blooms/acre	600 lb/acre

NOTE: These values were obtained by comparing simulation results with published data.

where

H_f = heat output of the evaporative pad, Btu/hr
H_s = solar heat gain, Btu/hr
H_e = heat released by equipment, Btu/hr
H_r = heat from plant respiration, Btu/hr
H_c = heat lost by conduction, Btu/hr
H_t = thermal radiation heat loss, Btu/hr
H_{PH} = solar energy used for photosynthesis, Btu/hr
H_g = heat lost to the ground, Btu/hr
H_V = heat lost or gained in the ventilating air, Btu/hr

Note that H_r, H_{PH}, and H_e are negligible and will be ignored.

Heat Loss by Conduction

The conductive heat loss is found using:

$$H_c = UA(T_i - T_a) \tag{21}$$

where

U = heat transfer coefficient, $Btu\text{-}hr^{-1}\text{-}ft^{-2}\text{-}°F^{-1}$
T_i = temperature inside of greenhouse, °F
T_a = temperature of the ambient air, °F
A = area exposed to the outside, ft^2.

Here, U = 0.56 for double plastic. We can compute an overall UA for the structure through these area calculations:

$$\begin{aligned} \text{Area of roof} &= 20 \text{ bays x } 27 \text{ ft x } 100 \text{ ft} = 54{,}000 \text{ ft}^2 \\ \text{Area of sides} &= 2 \text{ sides x } 8 \text{ ft x } 200 \text{ ft} = 3{,}200 \text{ ft}^2 \\ \text{Area of ends} &= 20 \text{ ends x } 252 \text{ ft}^2 = 5{,}040 \text{ ft}^2 \\ \text{Total area} &= 62{,}240 \text{ ft}^2 \\ \text{UA total} &= 62{,}240 \text{ x } 0.56 = 34{,}854.4. \end{aligned}$$

Therefore,

$$H_c = 34854.4\,(T_i - T_a) \tag{22}$$

Heat Loss to the Ground

The heat lost to the ground, H_g, is relatively small but will be included because of the large ground to wall area ratio:

$$H_g = 0.1\,A_g\,(T_i - T_{gw}) \tag{23}$$

where

A_g = area of the greenhouse floor, ft^2
T_{gw} = ground water temperature, °F
0.1 = heat transfer coefficient of the ground, But-hr^{-1}-ft^{-2}-$°F^{-1}$

The upper layer of soil will be in equilibrium with the greenhouse temperature T_i. Although the soil surface will experience fluctuations due to evaporation and radiative heat transfer, the thermal conductivity of the soil is low enough to damp these fluctuations considerably. The ground temperature is that of the ground water which is approximated by the mean annual temperature [3].

Here,

$$A_g = 240 \text{ ft wide x } 200 \text{ ft long} = 48{,}000 \text{ ft}^2.$$

Thus,

$$H_g = 4800\,(T_i - T_{gw}) \tag{24}$$

Heat Loss by Radiation

The thermal radiation heat loss is the difference between the thermal radiation from the surface and the thermal radiation from the atmosphere. This is the reasoning used by Walker [20].

$$H_t = \sigma A_t \tau_t\,(\varepsilon_s T_s^4 - \varepsilon_a T_{aa}^4), \tag{25}$$

where

σ = Stefan-Boltzman constant = 1.714×10^{-9} Btu-hr^{-1}-ft^{-2}-°F^{-4}
τ_t = thermal radiation transmittance, decimal
A_t = area of the surface radiating thermally, ft^2
ε_s = emissivity of the surface, decimal
T_s = absolute temperature of the surface, °R
ε_a = apparent emissivity of the atmosphere, decimal
T_{aa} = absolute atmospheric temperature near the ground, °R

The area of the surface radiating thermally, A_t, is essentially the area of the ground, unless there is considerable foliage. The thermal transmittance of polyethylene is 0.698, but since the radiation must pass through three layers, the overall transmittance is $(0.698)^3 = 0.34$. The emissivity of the surface is 0.90, and that of the atmosphere is 0.82. The temperature of the surface is close to the average temperature inside of the greenhouse, T_i. We must add 460 to the F temperature to convert them to °R. Therefore,

$$H_t = 2.79725 \times 10^{-5} [(0.9(T_i + 460)^4 - 0.82 (T_a + 460)^4] \tag{26}$$

Solar Heat Gain

The solar gain is given by [20]:

$$H_s = a_s \tau I A s \tag{27}$$

where

a_s = absorptivity of the surface for solar radiation, decimal
I = solar intensity on a horizontal surface, Btu-hr^{-1}-ft^{-2}
A_s = area of surface receiving solar radiation, ft^2

Again, the area receiving the solar radiation is essentially the area of the ground, Ag = 48,000 ft^2. For plants and soil, $a_s \approx 0.77$. Since there are three sheets of polyethylene, each with a solar transmittance of 0.9, the overall transmittance will be $(0.9)^3 = 0.729$. We see that:

$$H_s = (26943.84)I \tag{28}$$

Heat Loss by Ventilation

Ventilation heat losses include both sensible and latent heat. Ventilation sensible heat losses, H_{VS}, are given by:

$$H_{VS} = \frac{60CpF(T_i - T_a)}{V_s} \tag{29}$$

where

Cp = specific heat of air, 0.24 Btu-lb^{-1}-°F^{-1}
V_s = specific volume of air, 13.35 ft^3/lb
60 = conversion factor, min/hr
F = air flow rate into greenhouse, cfm

The air flow rate, F, is the sum of ventilation (F_{vent}) and infiltration (F_{inf}). The number of air changes per hour for a well-constructed, new double polyethylene greenhouse is 0.75 [21]. The number of air changes per hour can be converted to cfm by:

$$F_{inf} = \frac{nV}{60} \tag{30}$$

where

n = number of air changes per hour, hr^{-1}
60 = conversion factor, min/hr
V = volume of the greenhouse, ft^3
F_{inf} = infiltration, cfm

In order to combine equations (29) and (30), we must first perform some simple arithmetic. Hence:

Area of attic ends = 1/2 x 24 ft x 5 ft = 60 ft^2 each.

In the following calculations, we will consider the central access aisle volume as part of the other volume calculations by ignoring the partitions.

Growing volume = 8 ft x 240 ft x 200 ft = 384,000 ft^3
Attic volume = 60 ft^2 x 100 ft x 20 bays = 120,000 ft^3
Total volume = 384,000 ft^3 + 120,000 ft^3 = 504,000 ft^3

Using this value for V, equation (29) becomes:

$$H_{VS} = 1.08\,(6300 + F_{vent})\,(T_i - T_a) \tag{31}$$

To the ventilation sensible heat losses we must add the ventilation latent heat losses. These losses are caused by water, which has been evaporated by heat within the greenhouse, escaping as water vapor to the atmosphere and carrying the latent heat of vaporization with it. This term is much more pronounced than it would be in most structures, because we are dealing with a greenhouse having a high internal humidity.

For these latent heat losses, the expression is:

$$H_{VL} = H_{fg}(\omega_g - \omega_a)\left(\frac{60}{13.35}\right)(F_{vent} + F_{\infty}) \tag{32}$$

where

h_{fg} = latent heat of vaporization of water, Btu/lb
ω_g = specific humidity of greenhouse air, lb H_2O/lb air
ω_a = specific humidity of ambient air, lb H_2O/lb air
$\frac{60}{13.35}$ = conversion factor (60 min/hr) (1 lb air/13.35 ft^3)

Methods for calculating the specific humidities are discussed in the following section. The H_V term of equation (20) is determined by summing H_{VL} (equation (32)) and H_{VS} (equation (31)). Equation (20) is then used to determine, H_F, the heat output of the evaporative pad. Data requirements for this evaluation include site-specific climatic information (T_a, T_{gw}, and I), known constants (h_{fg}, V_s, Cp, U, a_s, σ, τ_τ, ε_s, and ε_a), and certain design parameters (T_i, $F_{vent,}$ n, A, A_g, A_s, and V). The heat output, H_f, will then be combined with the flow requirements as determined in the next section.

FLOW REQUIREMENTS FOR EVAPORATIVE PAD GREENHOUSE

Simulating the heat transfer through the evaporative pad is no easy task. A common approach is to subdivide the pad into elemental areas and use a numerical integration scheme [22, 23]. This method is too time consuming to be useful here. Instead, we will make some simplifying assumptions. The resulting model will give an approximate answer with far less computation time.

Figure 3 shows the pattern of air circulation during the heating and cooling modes. The following principles form the boundary conditions for this problem. During the heating mode, warm greenhouse air is continually circulated through the evaporative pad by way of the attic. The air is very nearly saturated both when it enters and when it leaves the pad. Thus, we can use the specific humidities of saturated air in our calculations. During the cooling model, the air entering the pad has come from the outside and has the specific humidity and temperature of the outside air. By the time the air has left the evaporative pad, it is very nearly saturated, and has the specific humidity of saturated air. The temperature of the water entering the pad is the temperature of the warm water that's being piped to the greenhouse. The temperature of the air leaving the pad is the thermostat set point temperature of the greenhouse. Over a wide range of temperature and flow conditions [23], the water leaves the pad 3°C (5°F) warmer than the air leaving the pad. The air enters the pad at the outside temperature

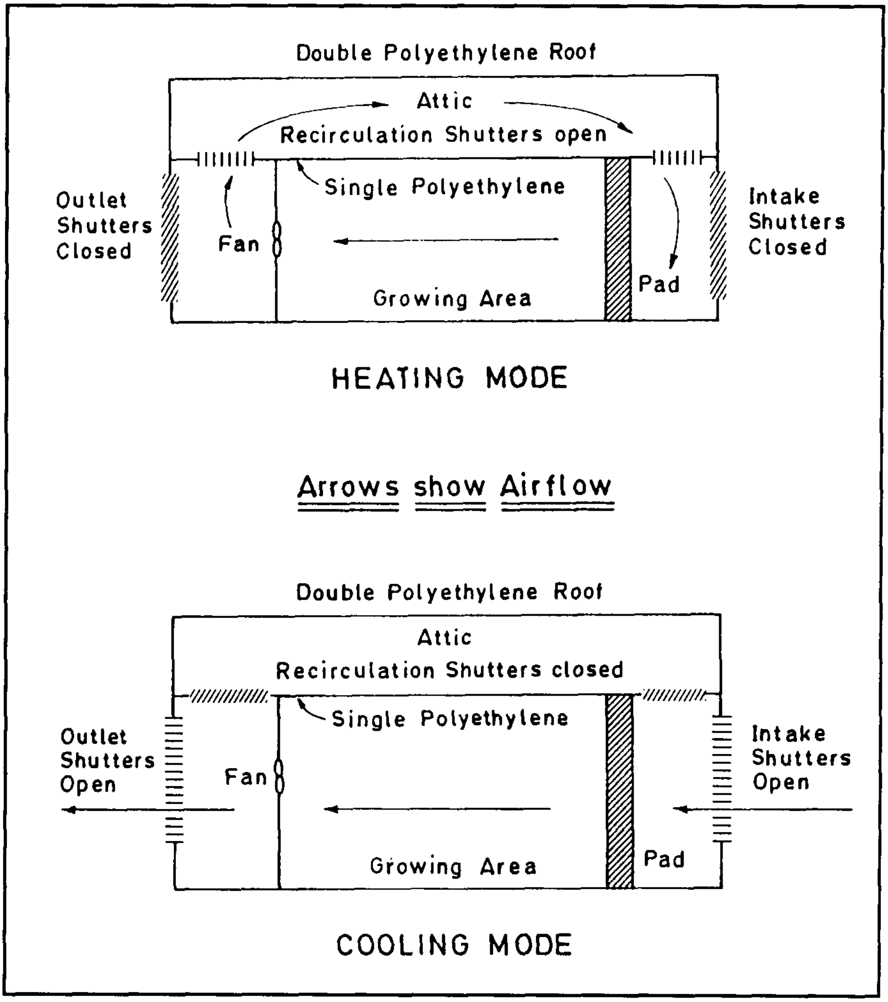

Figure 3. Evaporative pad greenhouse air circulation.

during the cooling mode. During the heating mode, the air enters the pad at a temperature determined by the heat loss of the greenhouse.

An empirical heat transfer equation and a heat balance on the warm water stream allow us to solve for the unknowns. The heat transfer rate of the CELdek evaporative pad is given graphically [23]. The following equation has been fitted to Olszewski's data:

$$Up = 120\, F_L^{0.7187} \tag{33}$$

where

Up = convective heat transfer coefficient per unit area of pad, $Btu\text{-}hr^{-1}\text{-}°F^{-1}\text{-}ft^{-2}$
F_L = flow per linear foot of pad, gpm/ft

The total heat transfer coefficient is obtained by multiplying by the area of the pads as follows.

$$U_{PT} = U_P A_P = (120F_L^{0.7187})\ (24\ \text{ft} \times 8\ \text{ft} \times 20\ \text{pads})$$
$$= 460800F_L^{0.7187} \tag{34}$$

where

U_{PT} = convective heat transfer coefficient of pads, $Btu\text{-}hr^{-1}\text{-}°F^{-1}$
A_p = total area of pads in the greenhouse = 3840 ft^2

We would like express equation (34) in terms of m, which is the mass flow rate of water (lb/hr) into the greenhouse. Since $m = 239512F_L$ (by unit conversion), we obtain from equation (34):

$$U_{PT} = 62.7162m^{0.7187} \tag{35}$$

The convective heat loss from the pads is calculated using [24]:

$$H_A = U_{PT}(T_{w_{av}} - T_{a_{av}}) \tag{36}$$

$$T_{w_{av}} = \frac{T_{w_2} + T_{w_1}}{2} \tag{37}$$

$$T_{a_{av}} = \frac{T_{a_2} + T_{a_1}}{2} \tag{38}$$

where

H_A = convective heat loss from the pads, Btu/hr
T_{w_2} = temperature of water exiting pad, °F
T_{w_1} = temperature of water entering pad, °F
T_{a_2} = temperature of air exiting pad, °F
T_{a_1} = temperature of air entering pad, °F

The evaporative heat transfer coefficient is related to the convective heat transfer coefficient by the Lewis Number [24]:

$$L_e = \frac{U_{PT}}{E_e C_P} \tag{39}$$

where

E_e = evaporative mass transfer coefficient, lb/hr
L_e = Lewis Number (0.894 for air and water at these temperatures).

The evaporative heat loss from the pads is given by [24]:

$$H_E = m_e h_{fg} \tag{40}$$

with

$$m_e = E_e\,(\omega_r - \omega_{aav}) \tag{41}$$

$$\omega_{aav} = \frac{\omega_2 + \omega_1}{2} \tag{42}$$

where

m_e = evaporative mass loss, lb/hr
ω_{aav} = average specific humidity of the air, lb H_2O/lb air
ω_r = saturated specific humidity at the average pad temperature, T_{wav}, lb H_2O/lb air
ω_2 = specific humidity of air exiting pad, lb H_2O/lb air
ω_1 = specific humidity of air entering pad, lb H_2O/lb
H_E = evaporative heat loss from the pads, Btu/hr

The enthalpy balance on the water stream in the pad is given by [24]:

$$-H_L = m(h_{L_2} - h_{L_1}) - m_c h_{L_2} \tag{43}$$

where

H_L = heat lost by water stream passing through pad, Btu/hr
h_{L2} = enthalpy of water exiting pad, Btu/lr
h_{L1} = enthalpy of water entering pad, Btu/lb

We can solve this equation for m:

$$m = \frac{m_c h_{L_2} - H_L}{H_{L_2} - H_{L_1}} \tag{44}$$

Note that the heat lost by the water as it passes through the pad is the sum of the convective and evaporative losses:

$$H_L = H_A + H_E \tag{45}$$

We can substitute equation (45) for H_L in equation (44) to get:

$$m = \frac{H_A + H_E - m_c h_{L_2}}{H_{L_1} - h_{L_2}} \tag{46}$$

The next step is to replace H_A and H_E with equations (36) and (40), respectively, in equation (46):

$$m = \frac{U_{PT}(T_{w_{av}} - T_{a_{av}}) + m_c(h_{fg} - h_{L_2})}{h_{L_1} - h_{L_2}} \tag{47}$$

We can use equation (41) in place of m_e and write:

$$m = \frac{U_{PT}(T_{w_{av}} - T_{a_{av}}) + E_c(\omega_r - \omega_{a_{av}})(h_{fg} - H_{L_2})}{h_{L_1} - h_{L_2}} \tag{48}$$

Now recall equation (39), with numerical values inserted:

$$E_e = \frac{U_{PT}}{L_e C_p} = \frac{U_{PT}}{(0.894)(0.24)} = 4.6607U_{PT} \tag{49}$$

We can express this in terms of m through equation (35):

$$E_e = (4.6607)(62.7162)m^{0.7187} = 292.3m^{0.7187} \tag{50}$$

Equations (35) and (50) replace H and E_e in equation (47):

$$m = \frac{62.7162m^{0.7187}(T_{w_{av}} - T_{a_{av}})}{h_{L_1} - h_{L_2}} + \frac{292.3m^{0.7187}(\omega_r - \omega_{a_{av}})(h_{fg} - h_{L_2})}{h_{L_1} - h_{L_2}} \tag{51}$$

which can be simplified as:

$$m = \left[\frac{62.7162(T_{w_{av}} - T_{a_{av}}) + 292.3(\omega_r - \omega_{a_{av}})(h_{fg} - h_{L_2})}{h_{L_1} - h_{L_2}}\right]^{3.555} \tag{52}$$

We now have an expression for the flow requirements in terms of the enthalpies, temperatures, and specific humidities which are easily determined. These variables may be calculated from the temperature and relative humidity data which will be available $T_{w_{av}}$ and $T_{a_{av}}$ are determined using equations (37) and (38).

For water at these temperatures, a good approximation is [25]:

$$h_L = 0.999T - 31.86 \tag{53}$$

$$h_{fg} = 1093.9 - 0.57T \tag{54}$$

where

h_L = enthalpy of water at temperature T, Btu/lb.

Since the water enters at T_{w_1} and leaves at T_{w_2}:

$$h_{L_1} = 0.999T_{w_1} - 31.86 \tag{55}$$

$$h_{L_2} = 0.999T_{w_2} - 31.86 \tag{56}$$

$$h_{fg} = 1093.9 - 0.57T_{w_{av}} \tag{57}$$

Expressions for saturation vapor pressure and atmospheric pressure have been presented by Thackston and Parker [26]:

$$p_g = 0.491 \exp\left[17.62 - \frac{9501}{T + 460}\right] \tag{58}$$

$$p = \left[\frac{14.7}{\exp\left(\dfrac{32.15E}{1545(T_{aa} + 460)}\right)}\right] \tag{59}$$

where

E = elevation of the site, ft
p_g = saturation vapor pressure, psi
p = atmospheric pressure, psi

Specific humidity for water-air mixtures can be found using [25]:

$$\omega = 0.622\left(\frac{p_g\Phi}{p - p_g\Phi}\right) \tag{60}$$

where

ω = specific humidity, lb water/lb air
ϕ = relative humidity $\dfrac{\text{actual vapor pressure}}{\text{saturation vapor pressure}}$
0.622 = ratio of molar masses of water to air.

Therefore, we need to calculate:

$$p_{g_1} = 0.491 \exp \; 17.62 - \frac{9501}{T_{a_1} + 460} \tag{61}$$

$$p_{g_2} = 0.491 \exp \; 17.62 - \frac{9501}{T_{a_2} + 460} \tag{62}$$

$$\omega_1 = 0.622 \left(\frac{p_{g_1}\Phi_1}{p - p_{g_1}\Phi_1}\right) \tag{63}$$

$$\omega_2 = 0.622 \left(\frac{p_{g_2}\Phi_2}{P - p_{g_2}\Phi_2}\right) \tag{64}$$

$$p_{gr} = 0.491 \exp \left[17.62 - \frac{9501}{T_{w_{av}} + 460}\right] \tag{65}$$

$$\omega_r = 0.622 \left(\frac{p_{gr}}{P - p_{gr}}\right) \tag{66}$$

where

p_{g_1} = saturation vapor pressure at T_{a_1}, psi
p_{g_2} = saturation vapor pressure at T_{a_2}, psi
p_{gr} = saturation vapor pressure at $T_{w_{av}}$, psi
ϕ_1 = relative humidity of air entering pad
ϕ_2 = relative humidity of air exiting pad.

Note that since ω_r is a saturated specific humidity, in equation (66) the relative humidity, ϕ, equals 1, and so it vanishes.

Equations (61) and (62) are used to calculated the vapor pressure needed for equations (63) and (64), respectively. Equations (63) and (64) provide the specific humidities needed for equation (42).

The assumptions which were made in the conceptual discussion at the beginning of this section allow us to know the temperatures of the water and the air, as well as the relative humidity of the air, both when they enter the pad and when they leave the pad. Now that we have derived the equations into which they will be substituted, we can refer to Table 2 which summarizes these known values.

Table 2. Summary of Initial and Final Conditions for Evaporative Pad Air and Water Streams

Description	*Parameter*	*Heating Mode*	*Cooling Mode*
Entering water temperature, °F	T_{w_1}	T_{ww}	T_{ww}
Exiting water temperature, °F	T_{w_2}	$T_1 + 5$	$T_i + 5$
Entering air temperature, °F	T_{a_1}	T_f	T_{aa}
Exiting air temperature, °F	T_{a_2}	T_i	T_i
Entering air relative humidity	ϕ_1	1.0	ϕ_{aa}
Exiting air relative humidity	ϕ_2	1.0	1.0

There is an additional point to consider in this discussion, and that is the temperature of the greenhouse air after it has passed through the growing area and returned to the pad via the attic. This is called T_F in Table 2.

The heat lost by the air as it passes through the greenhouse is:

$$-H_f = m_a C_P (T_f - T_i) \tag{67}$$

where

m_a = mass flow rate of the air, lb/hr.

The H_f needed for equation (67) is calculated using the heat balance of equation (20). Solving for equation (67) for T_f.

$$T_f = T_i - \frac{H_f}{m_a C_P} \tag{68}$$

The recommended air flow rate for evaporative pads is one pass through the greenhouse per minute [27]. The value of the greenhouse is 504,000 ft^3, calculated when deriving equation (31). We can use this to find the denominator of equation (68):

$$m_a C_P = \left(\frac{504000\ ft^3}{1 min}\right) \left(\frac{60\ min}{1\ hr}\right) \left(\frac{1\ lb}{13.35\ ft^3}\right) \left(\frac{0.24\ Btu}{1\ lb\text{–}°F}\right)$$

$$= 543640\ Btu\text{-}hr^{-1}\text{-}°F^{-1}$$

The numerical version of equation (68) is:

$$T_f = T_i - \frac{H_f}{543640} \tag{69}$$

And, finally, the flow of water out of the greenhouse can be found by subtracting the evaporative losses, m_e, from the total flow of water into the greenhouse:

$$m_f = m - m_e \tag{70}$$

where

m_f = mass flow of water out of greenhouse, lb/hr.

The principal items in this discussion, m and m_e, are determined using equations (52) and (41), respectively. In addition to the initial and final conditions listed in Table 1, and H_f from equation (20), the data requirements include site-specific weather data (T_{aa}, p, and ϕ_{aa}), known constants (C_p and L_e), and design parameters (T_i, T_{ww}, A_p, and E).

SUMMARY AND CONCLUSIONS

A mathematical model describing the behavior of evaporative pad-heated greenhouses has been developed. The model has been based upon a materials balance and a heat balance. The former provides, as output, growth rates and yields as a function of solar insolation, time to harvest, and temperature. The heat balance is used to determine the mass flow of water in and out of the greenhouse needed to maintain a particular greenhouse temperature. This evaporative pad greenhouse model has been designed to rely solely on readily available site-specific weather data.

NOMENCLATURE

The following is a listing of the nomenclature used throughout this article.

A = Area expand to outside, ft^2
A_p = Total area of greenhouse pads, ft^2
A_g = Area of greenhouse floor, ft^2
A_s = Area of insolated surface, ft^2
A_t = Area of the surface radiating thermally, ft^2
C = e^{-a}
C_p = Specific heat of air, Btu-lb^{-1}-$°F^{-1}$
E = Site elevation, ft
E_e = Evaporative mass transfer coefficient, lb/hr
F = Air flow rate between interior of greenhouse and surrounding, cfm
F_L = Flow per linear foot of pad, gpm/ft

F_{inf} = Infiltration air flow rate, cfm
F_{vent} = Ventilation air flow rate, cfm
H_A = Convective heat loss from pads, Btu/hr
H_E = Evaporative heat loss from pads, Btu/hr
H_L = Heat lost by water stream passing through pad, Btu/hr
H_{PH} = Solar energy used for photosynthesis, Btu/hr
H_V = Heat exchange with ventilating air, Btu/hr
H_{VL} = Ventilation latent heat loss, Btu/hr
H_{VS} = Ventilation sensible heat loss, Btu/hr
H_c = Heat lost by conduction, Btu/hr
H_e = Heat released by equipment, Btu/hr
H_f = Heat output of evaporative pad, Btu/hr
H_g = Heat lost to ground, Btu/hr
H_r = Heat released from plant respiration, Btu/hr
H_s = Solar heat gain, Btu/hr
H_t = Thermal radiation heat loss, Btu/hr
I = Solar intensity on a horizontal surface, Btu-hr^{-1}-ft^{-2}
I = Mean light intensity, ft-c
I_{eff} = Effective light intensity, ft-c
I_i = Incident light intensity, ft-c
I_s = Saturation light intensity, ft-c
K = Carrying capacity, lb
L_e = Lewis number, dimensionless
S = Total insolation, Btu-ft^{-2}-d^{-1}
T = Temperature, °F
T_H = Upper zero-growth temperature, °F
T_L = Lower zero-growth temperature, °F
T_a = Temperature of ambient air, °F
T_{aa} = Temperature of atmosphere near ground, °R
$T_{a_{av}}$ = Average temperature of air in pad, °F
T_{a_1} =Temperature of air entering pad, °F
T_{a_2} =Temperature of air exiting pad, °F
T_f = Greenhouse air return temperature, °F
T_{gw} = Ground water temperature, °F
T_i = Temperature inside of greenhouse, °F
T_s = Temperature of the surface, °R
$T_{w_{av}}$ = Average water temperature in pad, °F
T_{ww} = Temperature of water piped to greenhouse, °F
T_{w_1} = Temperature of water entering pad, °F
T_{w_2} = Temperature of water exiting pad, °F
U = Heat transfer coefficient, Btu-hr^{-1}-ft^{-2}-°F^{-1}

U_p = Convective heat transfer coefficient per unit area of pad, Btu-hr^{-1}-°F^{-1}-ft^{-2}
U_{PT} = Convective heat transfer coefficient of pads, Btu-hr^{-1}-°F^{-1}
V = Greenhouse volume, ft^3
V_s = Specific volume of air, ft^3/lb
X = Plant biomass, lb
X_0 = Plant biomass at planting, lb
X_{harv} = Plant biomass at harvest, lb
X_t = Plant biomass at time = t, lb

$$a = \ln \frac{X_o}{1 - \frac{X_o}{K}}$$

a_s = Absorptivity of the surface for solar radiation, decimal
h_L = Enthalpy of water, at temperature T, Btu/hr
h_{L_1} = Enthalpy of entering water, Btu/lb
h_{L_2} = Enthalpy of exiting water, Btu/lb
h_{fg} = Latent heat of vaporization of water, Btu/hr
m = Mass flow rate of water into greenhouse, lb/hr
m_a = Mass flow rate of air, lb/hr
m_e = Evaporative mass loss, lb/hr
m_f = Mass flow rate of water out of greenhouse, lb/hr
n = Air changes per hour, hr^{-1}
p = Atmospheric pressure, psi
p_g = Saturation vapor pressure, psi
p_{gr} = Saturation vapor pressure at $T_{w_{av}}$, psi
p_{g_1} = Saturation vapor pressure at T_{a_1}, psi
p_{g_2} = Saturation vapor pressure at T_{a_2}, psi
r = Intrinsic growth rate, hr^{-1}
r_o = Growth rate under normal light conditions, hr^{-1}
r_i = Growth rate at I_i, hr^{-1}
r_{max} = Maximum growth rate, hr^{-1}
r_s = Growth rate at τS, hr^{-1}
t = Time, hr
t_{harv} = Time of harvest, hr
ϕ = Relative humidity
ϕ_{aa} = Relative humidity of ambient air
ϕ_1 = Relative humidity of entering air
ϕ_2 = Relative humidity of exiting air
ε_a = Apparent emissivity of atmosphere, decimal
ε_s = Surface emissivity, decimal
σ = Stefan-Boltzman constant

τ = Solar radiation transmittance, decimal
τ_t = Thermal radiation transmittance, decimal
ω = Specific humidity, lb H_2O/lb air
ω_a = Specific humidity of greenhouse air, lb H_2O/lb air
ω_{aav} = Average specific humidity of air, lb H_2O/lb air
ω_g = Specific humidity of greenhouse air, lb H_2O/lb air
ω_r = Saturated specific humidity at $T_{w_{av}}$, lb H_2O/lb air
ω_1 = Specific humidity of entering air, lb H_2O/lb air
ω_2 = Specific humidity of exiting air, lb H_2O/lb air

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