ENERGY CONSERVATION THROUGH CONTROL OF GREENHOUSE HUMIDITY. I. CONDENSATION HEAT LOSSES

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Calculations were made on a quasi steady-state model describing heat loss in a greenhouse. The object was to assess the relative contribution of condensation to the total nighttime heat loss during cold weather operation, with a view to controlling severe condensation losses by dehumidification. Two cases were chosen for study — a single-glazed glasshouse and a greenhouse covered by a double layer of plastic film. Results show that significant heat loss by condensation (up to 20% of the total) may be expected only in plastic houses because of a lack of infiltration of air from outdoors. The use of mechanical dehumidifiers to eliminate this source of heat loss cannot be justified on the basis of energy savings alone.

INTRODUCTION

In climates where winter nighttime temperatures drop well below freezing point, condensation in greenhouses is observed, usually in the first few hours after nightfall. A high latent heat of condensation and relatively high rates of heat transfer combine to make condensation an efficient means of heat transfer. Certainly, early in the night, condensation should provide an important contribution to greenhouse heat losses.

Condensation in greenhouses is generally viewed by operators in terms of problems of plant health and discomfort to greenhouse workers and not in energy-economic terms. This is understandable, as capital charges and labor costs constitute the major cost of greenhouse operation. However, the fuel cost component has more than tripled since 1973. Fuel costs are expected to continue rising so that investigating means of reducing greenhouse energy requirements is timely.

Thus, the purpose of this study is to assess the magnitude of condensation heat loss in greenhouses, in cold weather, and, if significant, to investigate means and costs of humidity control. In order to cover a variety of situations, heat loss calculations are carried out on a single-glazed glasshouse, and on a double-layer plastic film greenhouse of identical dimensions.

Studies of greenhouse heat loss have been undertaken by Walker (1965), Walker and Cotter (1968), Walker and Walton (1971), Hoare and Morris (1956) and Whittle and Lawrence (1960). Walker (1965) used a steady-state heat balance to calculate temperatures inside plastic-covered houses and showed that temperatures can be predicted to within an error of about 1.5°C even though thermal capacity effects have been neglected.

Takakura et al. (1969) simulated the hour-to-hour climate in a greenhouse and concluded from a comparison of inside air, shaded leaf surface and soil surface temperature with measurements made on two summer days that a satisfactory simulation was achieved. Takakura et al. (1969) established that the leaf temperatures follow inside air temperature closely, an observation used in this study.

Condensation has been recognized as an important contribution to heat loss (Hoare and Morris 1956; Takakura et al. 1969), but the magnitude of its contribution has not been assessed. Takakura et al. (1969) included a condensation term in their dynamic model of a greenhouse, but they did not estimate its magnitude. Walker and Cotter (1968) and Walker and Walton (1971) dealt with condensation and humidity in plastic greenhouses. The former paper demonstrates that condensation is an effective means of reducing humidity after sunset during the cold months; this nighttime humidity is quite constant and is accurately estimated through heat transfer calculations. Walker and Walton further argued that condensation reduces thermal radiation loss through the polyethylene film.

Temperature distribution in heated greenhouses has been of considerable concern to operators. A sizeable literature on this subject exists (Anonymous 1947, 1971; Bark and Carpenter 1969; Carpenter et al. 1970; Carpenter and Bark 1967; Whittle and Lawrence 1960). This literature, taken as a whole, suggests that appreciable temperature differences exist between the heat source (unit heater, polysleeve) and the plant mass. However, for heat loss calculations these temperature variations contribute only small errors.

Only one study was found that dealt with the use of mechanical humidification in greenhouses (Wolfe 1970). This paper was concerned with means for avoiding the ventilation of greenhouses in order to maintain high CO_2 levels. The conclusion reached was that refrigeration for both cooling and humidity control was far too expensive in relation to the returns expected.

METHOD OF ATTACK

Since condensation in a greenhouse varies with the time of day, it is tempting to use a dynamic model to investigate the importance of condensation heat losses. However, the objective of our study was only to assess the magnitude of heat losses by condensation compared to those through other mechanisms so that the detail provided by a difficult and costly dynamic analysis was not required.

A comparison of heat losses by various mechanisms may be made by employing a quasi steady-state model, that is, by assuming that the changes in humidity and temperature with time are slow. In mathematical terms this means that the time derivatives are small when compared with the flux terms in the dynamic heat balance. Comparison of the heat loss magnitudes may then be made by calculating the individual heat losses at different states of humditiy and temperature inside the greenhouse for different ambient temperatures, cloud covers and wind speeds.

Figure 1 shows the terms used in our quasi steady-state heat balance on the internal volume of the greenhouse. This volume is indicated by the dashed lines in the figure. The letter L refers to rates of heat loss, S to rates of heat supply and M to rates of mass flows. Temperature and relative humidity are given by t and H, respectively. The notation defines the terms used in the figure.

An energy balance simply states that the flow of energy supplied to the greenhouse must either be lost, absorbed by the plant mass, or stored in a part of the greenhouse through an increase in temperature. Since part of the energy input or loss is associated with vaporizing water, the heat balance must be solved together with a water balance which states that all water entering the greenhouse within a span of time must leave (for the most part as humidity) or be absorbed in to either the plant mass, soil or air. Thus, to find the heat S, which must be added to maintain greenhouse temperature at t_I:

$$S_1 = L_1 + L_2 + L_3 + M_a C_p (t_I - t_E) - S_2 + Q_g - \lambda (M_{w_4} - M_{w_3}) \qquad (1)$$

where sensible heat in the water vapor has been neglected. The water balance is

$$M_{w_i} + M_{w_3} = M_{w_o} + M_{w_4} + M_{w_5} \qquad (2)$$

Preliminary calculations suggested that heat losses through the soil are considerably smaller than those through the shell and vary little with the ambient temperature. To simplify the analysis, soil losses have been treated as a constant (Silveston et al. 1976). Convective heat loss from the greenhouse depends on the temperature difference ($T_I - t_{gl}$), while the radiative loss depends on the difference between the fourth powers of temperatures ($t_I^4 - t_{gl}^4$). Thus, the glass or shell temperature (t_{gl}) must be known to calculate the heat losses. This temperature is readily obtained from a heat balance on the shell,

$$\sigma F_{ext}(t_{gl}^4 - t_{sky}^4) + h_E(t_{gl} - t_E) = \sigma F_{int}(t_I^4 - t_{gl}) + h_I(t_I - t_{gl}) + \lambda k_c(\bar{C}_A - C_{AI}) \qquad (3)$$

The rather small heat transfer resistance of the shell has been neglected. The shell temperature (t_{gl}) was obtained numerically by successive approximation rather than by solving the quartic equation. With t_{gl} known, greenhouse heat losses can be evaluated from the heat loss terms in the right-hand side of Eq. 3.

A similar approach was used when a double-glazed plastic house was considered. However, each plastic sheet assumes a different temperature, so a set of two equations with two unknowns must be solved. The equations are

$$h_I(t_I - t_{g1}) + \lambda k_c(\bar{C}_A - C_{AI}) + \sigma F(t_I^4 - t_{g1}^4) = (k^*/x)(t_{g1} - t_{g2}) + \sigma F(t_{g1}^4 - t_{g2}^4) \qquad (4)$$

$$k^*/x(t_{g1} - t_{g2})x + \sigma F(t_{g1}^4 - t_{g2}^4) = h_E(t_{g2} - t_E) + \sigma F(t_{g2}^4 - t_{sky}^4) \qquad (5)$$

Again, we neglect the temperature difference across the plastic film and a condensation layer, but do consider the temperature difference across the air gap. Successive approximation was used to solve the set of four equations.

Equation 1 is like the form given in the ASHRAE Guide (1967) and that used by Walker (1965) to investigate temperatures in ventilated greenhouses under daylight conditions. However, Walker expressed radiation in terms of transmittance and inside temperatures. This avoids the need for Eq. 3–5, but transmittance becomes a difficult parameter to evaluate. A reliable nighttime value for long-wave radiation does not seem to be available.

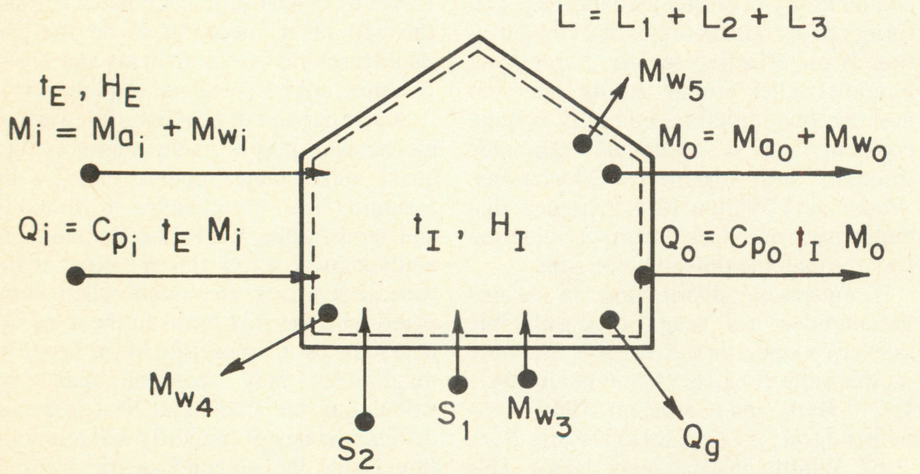

Figure 1. Schematic diagram of a greenhouse heat balance. The dashed line indicates the control volume for the mass and heat balance.

Polyethylene film widely used for plastic houses is not opaque to infrared radiation (Walker and Walton 1971). As an approximation, the transmission in the 5–25 μm range was assumed to be the same as the daytime transmittance and heat loss by transmittance was estimated. Separate calculations were performed assuming polyester films which can be regarded as opaque.

For glasshouses, infiltration ranges widely. Heating calculations (for example, Gray (1954)) generally assume three air changes per hour; we have done the same. Actual values vary widely and depend on tightness of the glasshouse, its location, wind speed and external and internal temperatures. However, the double-glazed plastic houses were assumed to have *no* infiltration during nighttime. The assumption of the number of air changes per hour is important and controls the relative importance of heat losses through infiltration.

Evapotranspiration behavior is not well defined for greenhouse conditions and greenhouse plants, particularly for nighttime conditions. Morris et al. (1957) have shown that there is a high degree of correlation between evaporation and solar radiation. On a clear, warm day, when solar radiation is high, an evapotranspiration rate of 0.66 mm H_2O/h was measured. Walker (1965), using different data, obtained a similar value for high foliage cover. Evapotranspiration rate has been observed to drop off drastically as the sun begins to set. Measured nighttime evapotranspiration rate drops to somewhere between 5 and 8% of the daytime maximum (Morris et al. 1957). Shorter days and perhaps colder greenhouse temperatures in winter will probably reduce nighttime rates further. However, for our calculations, 8% of the maximum was used for early evening. The contribution of evapotranspiration (ET) rate in Eq. 1 is given by Walker (1965) (the coefficient having been modified to agree with the units in the notation) as follows:

$$\lambda M_{w_3} = 203\, A_{gr}(ET) \qquad (6)$$

Equations 1–5 assume uniform temperature and humidity within the air space of the greenhouse. The radiation terms further assume that foliage temperature is the same as the air temperature and that foliage covers the greenhouse floor. To simplify further the complex radiation exchange problem, view factors were calculated assuming that the greenhouse shape could be approximated by an infinite half-cylinder in which the cylinder was the greenhouse shell and the intersecting plane was the plant mass. The shell was assumed to be black to partially compensate for transmission, while the emissivity of the plant mass was taken as 0.95 (Takakura et al. 1969). Radiation from the greenhouse shell to the night sky, either clear or cloud-covered, was assumed to be black body and a unity direct view factor was employed. Black body radiation was also assumed between the plastic films in a double-glazed greenhouse. View factors used in the calculations are given in Table I. Details of the calculation are summarized in Appendix A.

Convective heat transfer coefficients used are given in Table I and were taken from the ASHRAE Guide and Data Book (1967). Whittle and Lawrence (1960) used the same values. The air layer between plastic sheets in a double-glazed house acts as a heat barrier. However, orientation of the surfaces as well as movement of the outer film caused by wind induce motion in the layer so that the effective conductivity is many times that of still air. For this study, a uniform 25-mm air barrier was used with an effective conductivity four times the conductivity in still air. When these values are used, the reduction in heat loss calculated for double glazing agrees reasonably well with reductions found in practice (ASHRAE Guide 1967; Whittle and Lawrence 1960). The mass transfer coefficients for condensation given in Table I was calculated from a *j*-factor correlation for flow across a surface (McCabe and Smith 1967) assuming an inside air velocity of 0.5 m/sec (conforming with ASHRAE data). If the same calculation is done for the heat transfer coefficient, a value of about the same magnitude given in Table I is obtained. Details are given elsewhere (Silveston et al. 1976).

TABLE I. TRANSPORT AND RADIATION PARAMETERS USED IN GREENHOUSE HEAT LOSS MODEL

Radiative view factors		
$\mathcal{F}_{ext}$	=	1.0 (radiation between effectively black bodies)
$\mathcal{F}_{int}$	=	0.58
$\mathcal{F}_{films}$	=	0.9 (view factor between infinite parallel plane of $\epsilon = 0.9$)
Transport parameters		
h_I	=	7.4 W/m²/K
h_E	=	9.1 W/m²/K (still air)
	=	34.1 W/m²/K (wind)
k^*	=	0.10 W/m/K (fourfold conductivity of still air)
k_c	=	mass transfer coefficient = 4.44 (10^{-4}) m/sec

Greenhouse sizes have not been standardized. Dimensions of the greenhouse, however, play only a small role in the relative magnitudes of different modes of heat loss. Thus, for present purposes, a model house was selected. The dimensions of this house (Fig. 2) were based on a review of current practice. A further consideration was the economic constraint that greenhouse dimensions should be small enough to permit dehumidification within an hour by a single, commercial-sized, mechanical dehumidifier.

Calculations of heat losses were performed for a single internal air temperature of 21°C and for relative humidities (RH) of 60, 70, 80, 90, and 100%. The justification of 21°C is that it is the optimal temperature for tomatoes (Anonymous 1971; Tiessen and Wiebe 1974), the dominant winter greenhouse crop in Ontario. It is also the median between day and night temperatures for other major crops such as lettuce and cucumbers (Anonymous 1971). Crude calculations done at 18 and 24°C showed little change in the relative sizes of the various modes of heat losses. Of course, their absolute sizes are very sensitive to the inside temperatuve. The relative humidity (RH) range corresponds to the changes which would be experienced at the end of the day when condensation sets in.

External air temperatures used were −18, −12, −7, and −1°C, covering what would be anticipated in Southern Ontario for the late autumn, mid-winter and early spring heating period. External RH of 50% were used. Since evening and night skies are frequently covered in the heating season in Southern Ontario, calculations were repeated for two sky conditions: (1) covered sky with no wind, representing favorable conditions from a heat loss standpoint, and (2) clear night sky and a steady wind of about 28 km/h.

Of course, the situation represented by these cases occurs only for a short period of time at the close of the day. After sunset, the internal temperatures drop, plant respiration and evaporation from the soil fall to just small fractions of daytime values. The vapor supply processes no longer keep pace with condensation and vapor loss through infiltration. Both absolute and relative humidities decrease. Heat loss through condensation decreases accordingly and becomes much smaller than the other loss sources, for most of the night.

IMPORTANCE OF CONDENSATION AS A HEAT LOSS

Figure 3 shows the percent contribution of condensation of total heat loss near sunset from a single-glazed glass greenhouse for various levels of RH and ambient air temperatures. The solid lines assume a cloud-covered night sky without wind, while the dashed lines represent a clear sky with a 28-km/h wind. Near sunset, condensation is evidently a significant heat loss even in a moderately tight glasshouse. It accounts for between 25 and 40% of the heat loss under typical Southern Ontario winter conditions.

As would be expected, the percent contribution of condensation is roughly proportional to the RH. On the other hand, it is virtually insensitive to the air temperature and sky conditions, although the absolute heat loss from the glasshouse will increase about linearly with the decrease in the air temperature. The clear sky, 28-km/h wind condition increases heat loss by ca. 40% at each ambient temperature.

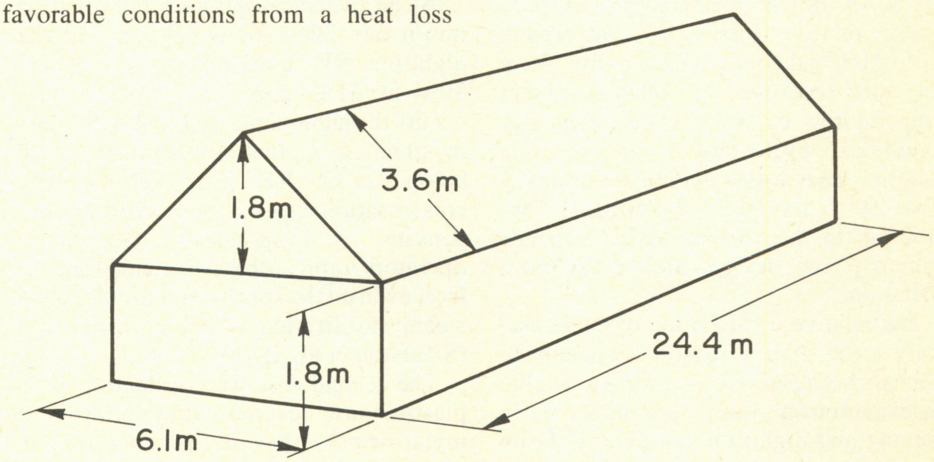

Figure 2. Model greenhouse assumed for the study. Greenhouse volume $V = 400$ m³; ground area $A_{gr} = 150$ m²; wall and roof area $A_{gl} = 300$ m².

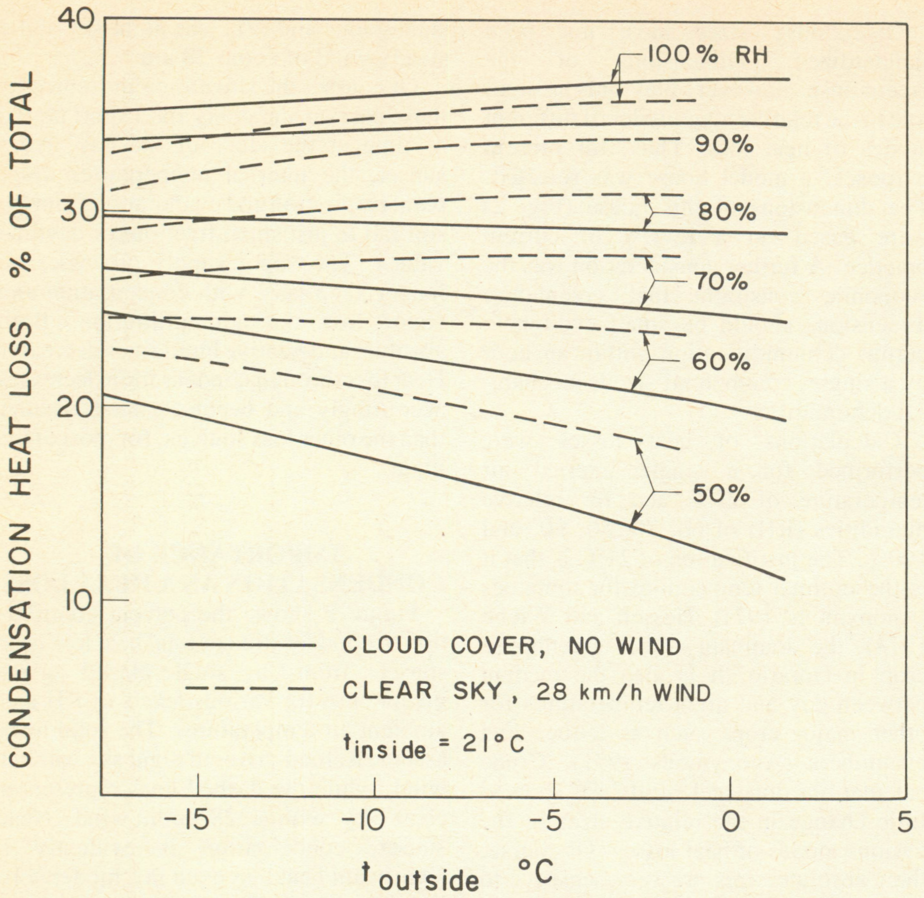

Figure 3. Condensation heat loss as a percentage of total heat loss for various inside and outside conditions in a single-glazed house.

The roles of other sources of heat loss inside the greenhouse are shown in Fig. 4. Changes in the contributions of radiation, convection, infiltration and condensation with RH are given assuming −18°C ambient temperature, no wind and a cloud cover. Radiation heat loss is seen to account for less than 10% of the total. Infiltration is significant at about 30% of the heat loss; its importance depends, of course, on the rate of air change assumed.

The relative losses by convection, infiltration and condensation in Fig. 4 are not too sensitive to other ambient temperatures or to windy, clear-sky conditions, even though the absolute radiative heat losses would increase by some 50%. Evidently the percent contribution to the various losses depends primarily on design and greenhouse operation.

The relative contribution of condensation for the heating season is difficult to estimate because so few data are available on evapotranspiration rates under early evening and nighttime conditions. From an energy conservation standpoint, nighttime, rather than early evening condensation heat loss, is the significant number. If we assume that evapotranspiration drops to 0.15 mm H_2O/h at sunset and an hour later to 0.05 mm/h, it takes about 1 h to reduce the RH from 90% to below 60%. During this period, about 36 kg of water condense. For a 12-h nighttime heating period, assuming a −12°C ambient temperature and a covered sky, condensation accounts for only about 3% of the heat loss.

At an evapotranspiration rate of 0.05 mm/h our calculations suggest that the nighttime RH drops to between 20 and 30% at 21°C and some condensation occurs throughout the night. What occurs in practice is that temperatures drop below 21°C, RH goes below 50%, condensation ceases, and indeed, condensate re-evaporates. Glasshouse operators rarely observe condensate or frost even on the coldest nights unless live steam sterilization is being carried out (Silveston et al. 1976).

The second case we considered was a plastic house covered with two layers of mylar or polyethylene film enclosing an air gap. In this construction, infiltration is essentially zero. The air gap provides higher heat transfer resistances, dropping the total heat loss in a double-film house to about 40% of that in a single-glazed glasshouse for the same outside conditions.

Figure 5 shows how the contribution to heat loss from condensation varies in the plastic house. With infiltration absent, condensation is considerably more important than in the glasshouse. As much as 60% of the heat loss in the early evening hours can be due to condensation. At 90% RH, neither the ambient air temperature nor the sky conditions make much difference in the relative importance of condensation, even though both conditions strongly affect the total heat loss. However, the relative contribution of condensation drops rapidly with RH so that below 70% RH, the role of condensation is severely affected by both the sky condition and t_E. The inside film temperature for the double-film plastic house is quite high, so condensation will not occur at 21°C for a RH under 60% if the sky is covered, or below 50% if the sky is clear for the outside temperatures investigated.

Polyethylene films transmit a large part of the incident thermal radiation. Calculations have been repeated allowing for transmittance, as discussed in Appendix A, but the behavior is similar to that shown in Fig. 5. Condensation, however, makes a smaller percentage contribution to the total heat loss.

Figure 6 is the counterpart of Fig. 4 for the plastic house. The two sets of curves allow for the differing transmittance of two common plastic covering materials. The figure shows that condensation is the largest source of heat loss at RH above 85%. In this range, it accounts for between 35 and 55% of the heat loss. However, the importance of condensation drops off rapidly as RH decreases. The contribution of radiation to heat transport in the greenhouse once again is small for a mylar films, but becomes very significant if polyethylene is the covering material. The rapid increase in the contribution as condensation ceases comes about because condensate on the plastic surface drastically reduces the transmittance. As was true for glasshouses, the more severe conditions of wind and a clear night sky changes the relationships very little, as shown in Fig. 6.

Condensation makes a much larger contribution to the winter energy demand for plastic houses than for glass ones. Not only is condensation more important as a loss early in the evening; it also takes place over a longer time period because it

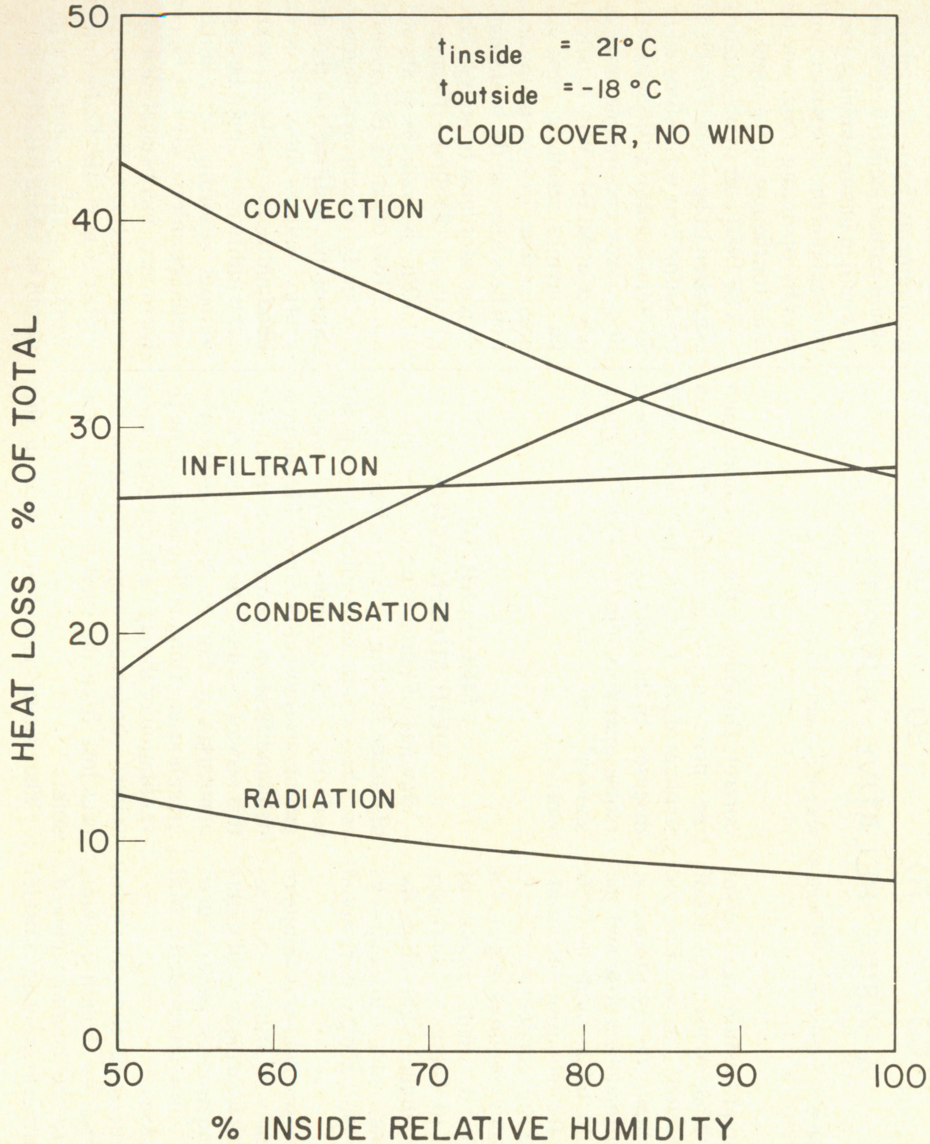

Figure 4. Comparison of heat losses for a single-glazed greenhouse.

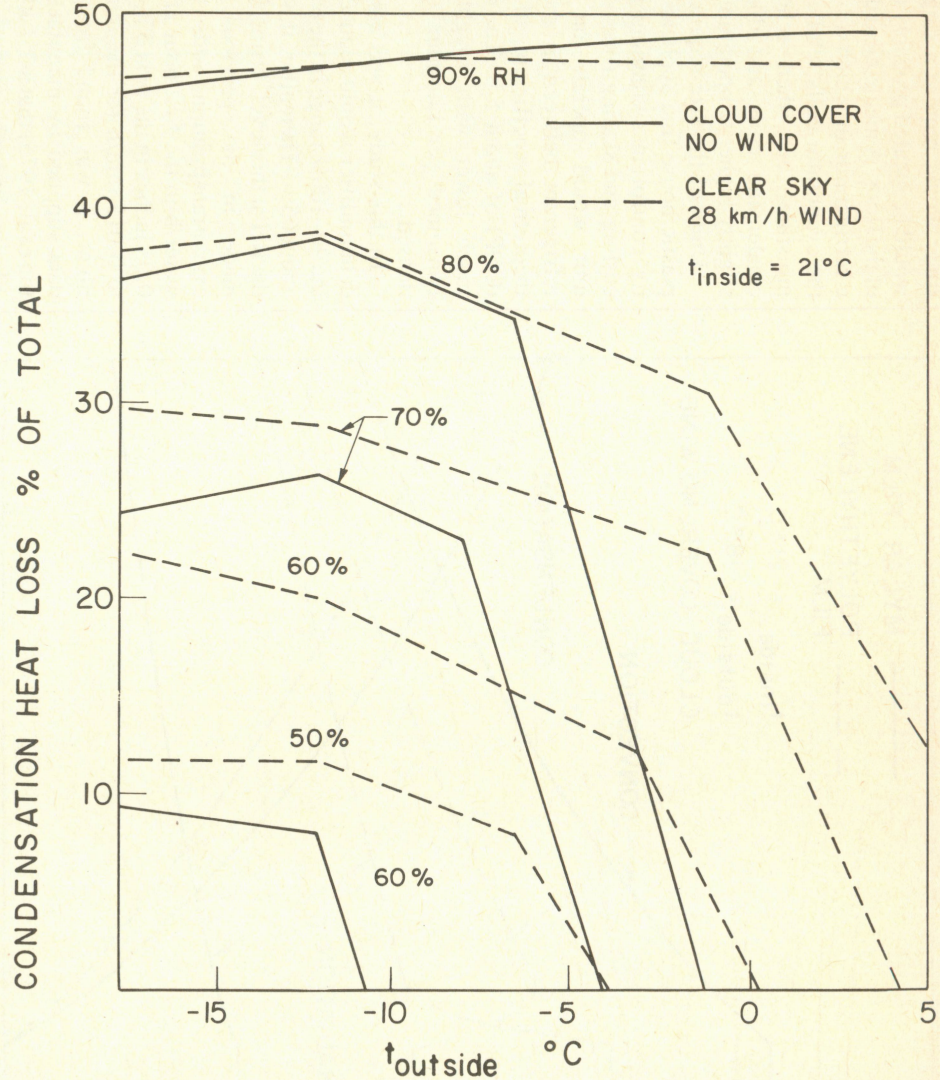

Figure 5. Condensation heat loss as a percentage of total heat loss for various inside and outside conditions in a double mylar film greenhouse.

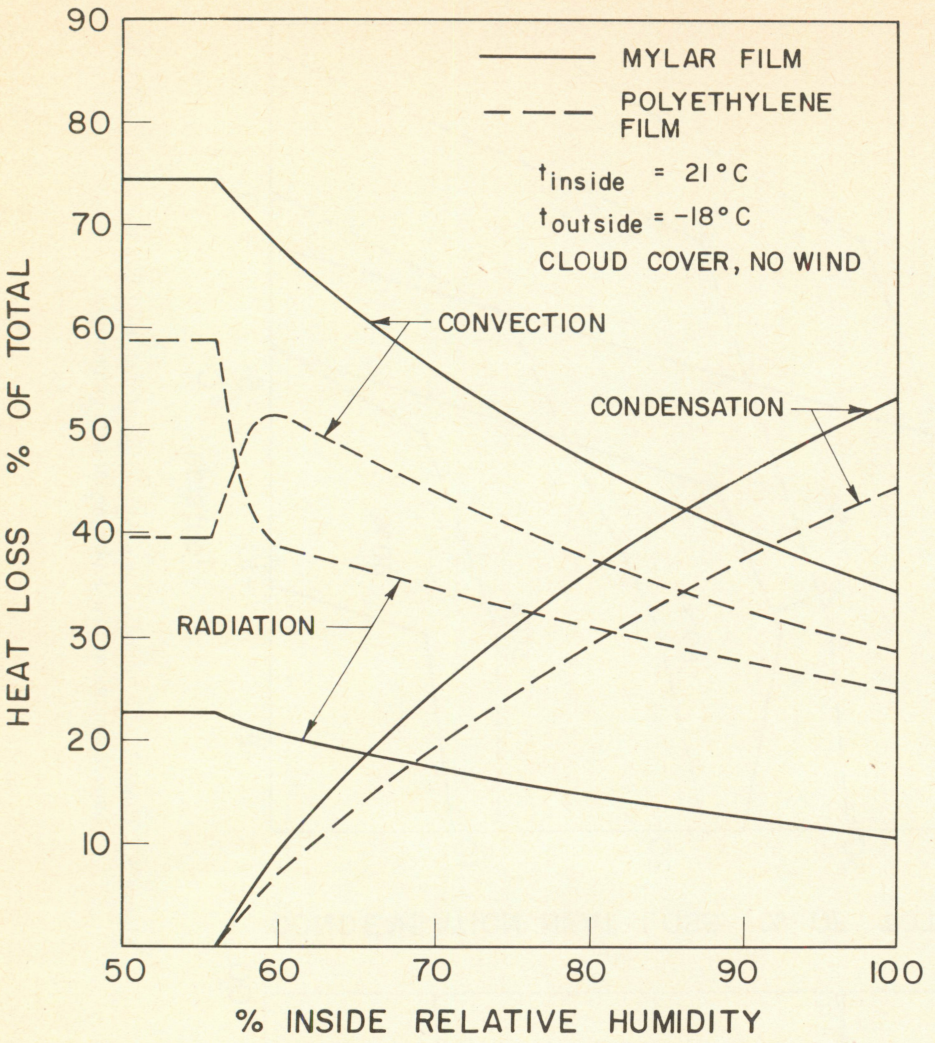

Figure 6. Comparison of heat losses for a double-film greenhouse.

is the only dehumidification mechanism in a tight house. Making the same set of assumptions as used above for the glasshouse, condensation contributes at least 15% of the total heat loss and the contribution could be as high as 20% when mylar is the covering material. Dehumidification by condensation to a level of about 65% requires about 4 h after sunset and a small amout of condensation probably continues throughout the winter night. For the model greenhouse under the conditions assumed, about 80 kg of water would condense.

Operators of plastic houses in Southern Ontario report heavy nighttime condensation in winter which on clear, cold nights forms frost. Condensation persists until morning. These observations agree well with our calculations.

It would seem desirable to try to control nighttime condensation since it accounts for up to 20% of the heating demand. There are two further reasons to consider control. Dripping of the condensate onto the plant mass at night has been implicated in botrytis and other leaf diseases. In the morning, heavy dripping (some operators refer to it as "morning shower") is a nuisance to greenhouse workers.

MECHANICAL DEHUMIDIFICATION

Savings of perhaps up to 10% of the nighttime heating cost, possibly as high as 15%, could be realized by rapidly reducing the humidity in plastic greenhouses just after sunset. Mechanical dehumidification seems best suited for this service because the heat released on condensation can be held inside the greenhouse. Dehumidification through ventilation or through a direct (e.g., cold water) condenser simply transfers the heat loss from the shell to another heat sink.

The model greenhouse used in this study would require a single unit which would be located probably near the center of the floor. A mechanical dehumidifier is a refrigerator in which air is drawn over the evaporator coils, chilled to condense moisture, and then reheated by the condensing coils and discharged at a slightly higher temperature.

Units of the size required for a single plastic house find application for humidity control in swimming pools, and in curling and skating rinks. A greenhouse unit would operate on a Freon 22 refrigerant between 2 and 49°C for a theoretical coefficient of performance (COP) of about 5.8. The actual COP is much lower with roughly 0.55 kWh consumed per kilogram of water condensed.

In this study, it was assumed that the dehumidifier would reduce the RH from the 90–100% range in the early evening to about 65%. Thereafter, it would maintain the RH at this level. Using the assumption made in the previous section that evapotranspiration adds 0.10 mm of water/h in the first hour after sunset and 0.05 inches in the next 3 h, finally falling to 0.015 mm/h for the remainder of the night, it was found from performance data supplied by manufacturers (Silveston et al. 1976) that a 3.7-kW unit rated at 14 kg water/h seemed to meet the requirements best. A unit of this size would reduce the RH to 65% in about 1 h. It would maintain humidity thereafter and re-evaporate the condensate on the inside film by operating about one-third of the time for the rest of the night.

The 1980 cost of an installed unit was estimated to be about $7000. The annual operating cost was estimated to be about $1600 assuming a 10-yr life for depreciation, taxes and maintenance at 8% and power at $0.04 kWh. Against this cost, the savings resulting from a somewhat greater than 75% reduction in condensation heat loss and the heat supplied by the refrigerator must be added. This amounts to about $170/yr if natural gas at $0.10/m^3 is available or about $270/yr if no. 2 fuel oil at $0.16/L is used.

Clearly, mechanical dehumidification cannot be justified on the grounds of energy savings alone. If a mechanical dehumidifier is available, daytime winter dehumidification could be accomplished without resorting to ventilation. This would permit the use of CO_2 enrichment. However, mechanical dehumidification would have to be justified on the basis of higher crop yields. Reduction in disease, worker comfort and energy savings would be added benefits.

DISCUSSION

The conclusions of a calculational study such as the one reported in this paper are only valid insofar as the results agree with those for the real systems modelled. We have used $t_E = -12°C$, a covered sky, a 12-h night and a 150-day heating year to make cost comparison. For the model greenhouse, these assumptions lead to a heating demand of about 538 GJ/yr for a single-glazed glasshouse. Allowing for furnace efficiency and assuming a natural-gas fuel at \$0.10/m^3, fuel cost for the model would be \$2100/yr. Translating to a hectare basis, the fuel cost amounts to about \$126 000/yr. Fuel cost for Southern Ontario commercial greenhouses is often quoted at \$110 000/ha · yr (Waterloo Research Institute 1976) corrected to 1980 fuel costs. Thus, our estimate is in reasonable agreement with widely used figures.

CONCLUSIONS

Condensation is an important mode of heat loss in both single-glazed glass greenhouses and double-filmed plastic houses in the first few hours of evening. However, condensation is only significant for the heating season as a whole for the plastic greenhouse where it may contribute as much as 15% of the heat loss.

Mechanical dehumidification is capable of reducing condensation heat losses in plastic houses by about 75%, but at the present cost of heating fuels, use of dehumidification cannot be justified.

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APPENDIX A
Calculation at Radiation Losses

Representing the greenhouse as half of a long, longitudinal-oriented cylinder such that the plane intersecting the cylinder is assumed to be the surface of the plant mass and the half cylinder, the greenhouse shell, an approximation can be made of the view factor between the plant mass and ground and the shell. We assume that equal emissivity for the plant mass and the glass or plastic skin (values quoted in the literature (Walker 1965; Whittle and Lawrence 1960) are 0.9–9.95 and approximate the grey body view factor by direct radiation view factors. Let 1 denote the plant mass (assumed to be a smooth plane) and 2 the shell; then following the method of Poljak (Siegel and Howell 1972) the pertinent equations are

$$q_1 = \frac{\epsilon}{1 - \epsilon}(\sigma' t_1^4 - q_{0,1}) = q_{0,1} - \mathscr{F}_{1\text{-}1} q_{0,1} - \mathscr{F}_{1\text{-}2} q_{0_2} \qquad \text{(A1)}$$

$$q_2 = \frac{\epsilon}{1 - \epsilon}(\sigma t_2^4 - q_{0,2}) = q_{0,2} - \mathscr{F}_{2\text{-}2} q_{0,2} - \mathscr{F}_{2\text{-}1} q_{0,1} \qquad \text{(A2)}$$

where q_1 is the net flux leaving the floor and represents the heat flux leaving the plant mass which is absorbed by the shell. For the geometry and view factors assumptions above, $\mathscr{F}_{1\text{-}1} = F_{1\text{-}1} = 1 - 2/\pi$, $\mathscr{F}_{1\text{-}2} = F_{1\text{-}2} = 2/\pi$, $\mathscr{F}_{2\text{-}1} = F_{2\text{-}1} = 1$, and $\mathscr{F}_{2\text{-}2} = F_{2\text{-}2} = 0$. Solving Eqs. A1 and A2 yields

$$q_1 = \frac{2\sigma' \epsilon}{\pi + 2(1 - \epsilon)}(t_1^4 - t_2^4)$$

The hourly heat loss is the product $q_1 A_1$. In the above equation, t_1 is the absolute temperature of the plant mass/100, while t_2 is the absolute shell temperature /100, and σ is the Stefan-Boltzmann constant × 10^8.

The grey body view factor in Eq. A3 works out to 0.58, almost three times the value used by Takakura (1969) for this exchange.

Equation A3 cannot be used if the film material is polyethylene because of the high transmittance of this material for thermal radiation. If we let t_c represent the absolute temperature of the atmospheric radiation sink (temperature of cloud bottom for an overcast sky) then

$$q_1 = \epsilon \tau \sigma' (t_1^4 - t_c^4) \frac{2\sigma' \epsilon'}{\pi + 2(1 - \epsilon')} (t_1^4 - t_2^4) \qquad \text{(A4)}$$

In this equation, the second term is a very crude approximation of the radiant exchange between the plant mass and shell assuming $\epsilon' = 1 - \tau$. The first term assumes equal emissivity for the plant mass and the atmospheric radiation sink. Walter (1965, 1968) used $\tau = 0.7$ for 0.12-mm polyethylene sheet and shows that condensate on the film sharply reduces this value. For calculations in this paper, we have allowed for condensation and a double film by letting $\tau = 0..25$ (Walker and Walton 1968).

NOTATION

A_{gl} Area of glass (m^2).

A_{gr} Ground area (m^2)

C_A Absolute humidity (kg/m^3).

$\bar{C}_A$ Mean absolute humidity (kg/m^3).

C_{p_o} Heat capacity of leaving water and air (= kJ/kg · K).

C_{p_i} Heat capacity of entering water and air (= kJ/kg · K).

ET Evapotranspiration rate (m water/s · m^2).

$\mathcal{F}$ Radiative view factors, see Table I.

h_I Interior convective heat transfer coefficient (W/m^2 · K).

h_E Exterior heat transfer coefficient (W/m^2 · K).

H_E Exterior relative humidity (dimensionless).

H_I Interior relative humidity (dimensionless).

k^* = 4 k_a where k_a= thermal conductivity of air (= 2.6 (10^{-2}) W/m · K) the factor of 4 accounts for air movement between the two films of plastic.

k_c Convective mass transfer coefficient (m/s).

L Heat loss from greenhouse.

L_1 Convection heat loss (W)
$= h_I A_{gl} (t_I - t_{gl})$.

L_2 Condensation heat loss (W)
$= \lambda k_c (\bar{C}_A - C_A) A_{gl}$.

L_3 Radiation heat loss (W)
$= \sigma F [t_{gr}{}^4 - t_{gl}{}^4] A_{gr}$.

M_{a_o} Mass of air leaving greenhouse (kg/s).

M_{a_i} Mass of air entering greenhouse (kg/s).

M_o Mass of air and water leaving (kg/s).

M_{w_o} Mass of water vapor leaving greenhouse *(kg/s)*.

M_i Mass of air and water entering (kg/s).

M_{w_i} Mass of water vapor entering greenhouse (kg/s).

M_{w_3} Evapotranspiration rate of plants and soil (kg H_2O/s).

M_{w_4} Mass of water removed by dehumidifier operation (kg/s).

M_{w_5} Condensation on glass surface (kg H_2O/s).

Q_g Heat absorbed into soil = 4.7 W/m^2 soil at 21°C.

Q_o Heat leaving greenhouse in air (W).

Q_i Heat entering greenhouse in air (W)...

S_1 Heat input from fuel (W).

S_2 Heat input from dehumidifier (W).

t_I Interior temperature (°C).

t_E Exterior temperature (°C).

t_{g1} Inside plastic film temperature (K).

t_{g2} Outside plastic film temperature (K).

t_{gl} Glass temperature (K).

t_{gr} Ground temperature (K).

t_{sky} Radiative sky temperature (K).

V Greenhouse volume (m^3).

Greek Letters

ϵ Emissivity.

λ Latent heat of condensation (= 2.5 kJ/g).

σ Stefan-Boltzmann constant
(= 5.67 (10^{-8}) W/m^2/K^4).

τ Transmissivity.

Subscripts

E Exterior.

I Interior.