INFRARED HEATING OF GREENHOUSES REVISITED: AN EXPERIMENTAL AND MODELING STUDY

A. Kavga, T. Panidis, V. Bontozoglou, S. Pantelakis

ABSTRACT. *The potential advantages of night-time heating of greenhouses by modern infrared (IR) radiative sources, instead of forced hot air, are examined experimentally and theoretically. Measurements of indoor and outdoor conditions during typical cold nights in central Greece were taken in an experimental greenhouse using either forced hot air or IR heating. A simple theoretical model that contains all the essential physics was developed and subsequently used in parametric studies. Experimental and simulation results confirmed that, with IR heating, inside air temperatures several degrees lower than the desired plant canopy temperature were sustained, and that this temperature difference increased proportionally to the nightly drop in outside temperature. The model estimated energy savings in the order of 45% to 50% using the IR sources currently available, and predicted significant further benefits from improvements in the radiative efficiency of the IR sources.*

Keywords. *Canopy temperature, Convection, Energy balance, Radiation, Simulation.*

Energy consumption for greenhouse heating represents a serious concern for greenhouse operators throughout the world (Bot, 2001; Tiwari, 2003). Conventionally, thermal energy is transmitted to the greenhouse either by circulation of hot water through a piping system or by air heaters (Van de Braak, 1988; Teitel et al., 1999). In order for the plants to reach the required temperature by these methods, the interior of the greenhouse has to be heated to the same or even to a slightly higher temperature than the value targeted for the plants. This practice results in increased heat losses due to conduction and radiation through the cover, and also due to ventilation and leakages through unintended openings.

Aiming to reduce energy consumption, a variety of modifications have been proposed and implemented during the last decade. Among them, we note the use of double glazing (Gupta and Chandra, 2002), the insulation of side walls (Singh and Tiwari, 2000; Gupta and Chandra, 2002), the introduction of thermal screens (Chandra and Albright, 1989; Kittas et al., 2003; Glosal and Tiwari, 2004), the use of different types of covering materials such as glass, PE, PVC, etc. (Zhang et al., 1996; Cemek et al., 2006), the development of zigzag covering in order to restrict transmission losses (Swinkels et al., 2001), and the development of Fresnel lenses for the south-facing roof cover (Tripanagnostopoulos et al., 2005). Although the above methods clearly result in energy savings, they do not address the basic premise of heating the entire greenhouse environment and then letting the plants gain energy from it.

An alternative method of heating the plants in a greenhouse, which was originally proposed in the late 1970s as a result of the first oil crisis (Youngsman, 1978; Itagi and Takahashi, 1978; Rotz and Heins, 1982; Nelson, 2003), is the use of low-intensity infrared (IR) radiation. The main potential advantage of IR heating is the direct delivery of thermal energy from the power source to the canopy, thus eliminating the need to increase the inside air temperature in order to deliver the necessary heat by convection. As a result, the cover and inside air may remain at significantly lower temperatures than the target value for the plants, with a concomitant reduction of energy losses.

Although IR heating has been advocated as having in principle the potential for improved energy performance (Hanan, 1998; Nelson, 2003), it has not been implemented at a significant scale. It is conjectured that the main obstacle hindering the method's potential from being realized is the radiative efficiency of the power source. For example, emission of IR radiation with peak intensity at 3 μm by a plain heated surface necessitates a temperature above 650°C. Such a hot surface inside the greenhouse will evidently suffer intense heat losses by natural convection to the surrounding air and by radiation to the cover, thus negating the advantages of IR heating. Evidence pointing in this direction was a comparative study of the vertical temperature distribution in a forced-air heated greenhouse and an IR-heated greenhouse (Blom and Ingratta, 1981) that indicated no significant temperature differences and actually more thermal stratification in the latter.

Unlike conventional hot surfaces, modern IR sources (gas radiators or electric infrared lamps) have improved efficien-

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cy, converting a significant percentage of the consumed energy to IR radiation. They emit highly concentrated radiation while their outer surface remains at a moderate temperature. Moreover, IR sources have fast response times and are suitable for modern automatic control implementation. The current wide availability of IR sources in the market, combined with prospects of forthcoming improvements in radiant efficiency and the expected drop in prices due to mass production, justify revisiting the idea of IR heating of a greenhouse (Kavga et al., 2008).

Thus, the goal of the study presented here was to examine the conditions under which IR heating may be energetically beneficial, and to investigate the potential for improvement in the method when implemented with the use of modern IR sources. To this end, we present a series of measurements in a small-scale greenhouse that was alternatively heated conventionally and by IR radiation. Both systems were electricity operated, using either an air heater or four IR lamps, to facilitate control and monitoring of their performance. Analysis of the data was assisted by a model that was also developed for the current study, and it incorporated all the essential thermal phenomena. A key issue, investigated by the model predictions, was the impact of convective energy losses by the hot radiating surfaces on the overall efficiency of the system. This work is focused on the heating needs during the night, since they constitute the main heating demand in the region where the greenhouse is located, which does not allow for the collection of statistically significant measurements during daytime heating.

IR radiation heating seems to be ideally suitable for relatively short plants. Existing greenhouses are often used only for short plants, and these are expected to be the first candidates for IR heating implementation. Therefore, the current work focused on greenhouses with canopies that can be approximated as two-dimensional, and lettuce plants were used in the experiments. A common problem observed in previous experiments utilizing IR heating was the uneven distribution of the radiation, resulting in uneven plant growth and development. The construction of modern radiative sources takes advantage of improved materials, minimizing heat losses from the source. In addition, modern IR heaters are equipped with efficient reflectors, providing superior control of the directional heat flux distribution. Engineering practices for artificial lighting designs may provide the necessary tools that, along with well defined specifications for the IR sources, can ensure efficient design of IR heating systems by considering the distribution and orientation of the IR sources. We believe that development of these techniques will allow efficient implementation of IR heating in more complex cases incorporating taller or thicker plant canopies.

Although energy efficiency provided the main motivation for the present study, another favorable side-effect is worth noting: When the plants are maintained at a higher temperature than the rest of the greenhouse, the possibility of moisture condensation on their surface is drastically reduced. Thus, IR heating of greenhouses is also expected to contribute to improved product quality (Teitel et al., 2000; Nelson, 2003), as preliminary qualitative observations indicate. Thus, our suggestion is in line with the modern trend of substituting forced-air heating with IR heating in many food processing applications (Sakai and Hanzawa, 1994; Nowak and Lewicki, 2004; Galindo et al., 2005; Hebbar et al., 2005, Tanaka et al., 2007) that are also predominantly driven by improvements in product quality.

METHODOLOGY

THEORETICAL MODEL

In this section, a thermal model is developed that is simple enough to avoid extensive computation but at the same time incorporates all the essential phenomena differentiating between forced-air and IR heating. The model is first validated by comparison with data and then used in a parametric study of the effect of outdoor conditions on the thermal losses associated with each heating method.

Based on the degree of detail in the measurements, spatially uniform temperatures of the cover, inside air, and plant canopy are assumed. Further, a pseudosteady-state assumption is invoked whereby the thermal problem is formulated as steady, and time variation is introduced only through changes in the outdoor temperature and wind speed. This last assumption is permissible because the time scale for changes in the external conditions significantly exceeds the characteristic times of thermal response of the main greenhouse components, i.e., the cover and the inside air.

Following the above-mentioned formulation, input variables to the problem are: the outdoor temperature (T_o) and wind velocity (u_w); the geometric and operational characteristics of the greenhouse, i.e., the base surface area (A_p) assumed fully covered with plants, the cover surface area (A_c), the greenhouse volume (V), the required plant temperature (T_p), the number of air changes per hour, N), and the mean emissivities of the cover surface (ε_c) and of the canopy (ε_p); and the convective heat transfer coefficients between the inside air and cover (h_{ac}), the inside air and canopy (h_{ap}), and the outside air and cover (h_{co}). The latter were derived from appropriate correlations. The unknowns are the temperatures of the cover (T_c) and of the inside air (T_α).

Further progress necessitates differentiation between the two heating alternatives considered. With forced-air heating, energy is primarily delivered to the inside air, and the plants are subsequently heated from the air by convection. In greenhouses without air circulation, significant temperature stratification develops, and the average T_a is considerably higher than T_p. In the present model, the most advantageous limiting case is adopted, corresponding to a greenhouse with complete mixing, assuming $T_a = T_p$. Thus, the only remaining unknown is the cover temperature, which can be computed from an energy balance using the cover itself as the control volume.

The cover gains heat by radiation and convection with the inside air and, at the same time, loses heat by radiation towards the sky and by convection to the outside air. A balance of these terms at steady-state leads to the following equation:

$$\frac{\sigma\left(T_p^4 - T_c^4\right)}{\dfrac{1-\varepsilon_p}{A_p \varepsilon_p} + \dfrac{1}{A_p F_{p\to c}} + \dfrac{1-\varepsilon_c}{A_c \varepsilon_c}} + A_c h_{ac}(T_\alpha - T_c)$$

$$- \varepsilon_c A_c \sigma\left(T_c^4 - T_{sky}^4\right) - h_{co} A_c (T_c - T_o) = 0 \tag{1a}$$

The first term corresponds to radiation exchange between the canopy and the cover, which are treated as two gray surfaces that form an enclosure. For the case study considered (a full house of lettuce), we simplify the above equation by making the assumption that the surface of the canopy is flat, and consequently the view factor appearing in equation 1a is equal to one. With this simplification, the equation becomes:

$$\frac{\sigma\varepsilon_p A_p\left(T_p^4 - T_c^4\right)}{1+(1-\varepsilon_c)(A_p\varepsilon_p / A_c\varepsilon_c)} + A_c h_{ac}(T_\alpha - T_c)$$
$$-\varepsilon_c A_c \sigma\left(T_c^4 - T_{sky}^4\right) - h_{co}A_c(T_c - T_o) = 0 \quad (1b)$$

We note that all the results in the present work are based on equation 1b, but the model can readily be applied to a canopy of plants projecting above the floor by using equation 1a with an appropriately selected view factor.

With IR heating, energy is delivered directly to the plant canopy, which is consequently at a higher temperature than its surroundings. Part of the energy offered to the plants is lost by convection to the inside air and by thermal radiation to the cover. Thus, with this mode of heat transfer, the inside air temperature is unknown and is expected to be lower than that of the plants. The additional equation needed for the calculation of T_a comes from the energy balance around a control volume that contains only the inside air. The air exchanges heat by convection with the plant canopy and the inside surface of the cover, and also exchanges mass with the exterior (air renewal by leakage and ventilation). An additional input of particular significance is the convective loss to the air from the hot IR lamps, ($Q_{lamps,conv}$). Convective losses from the lamps are estimated as $Q_{lamps,conv} = (1 - \eta)Q_{total}$, where Q_{total} is the total thermal losses, and η is the lamps' radiative efficiency. As we will see later, $Q_{lamps,conv}$ is inversely proportional to the efficiency of IR heating, and should ideally be kept as low as possible. This may be accomplished by using IR sources whose external surfaces do not heat up significantly during operation. The balance of the contributions mentioned above is expressed at steady-state by the equation:

$$A_p h_{ap}(T_p - T_\alpha) - A_c h_{ac}(T_\alpha - T_c)$$
$$-0.36NV(T_\alpha - T_o) + Q_{lamps,conv} = 0 \quad (2)$$

The numerical coefficient in the air renewal term stems from the air density and specific heat, as shown below in equation 3.

The convective heat transfer coefficients are estimated from correlations. More specifically, the heat transfer coefficient between the cover and external air is taken as $h_{co} = 0.95 + 6.76u_w^{0.49}$, where u_w is in m s^{-1}, and h_{co} is in W m^{-2} K^{-1} (Papadakis et al., 1992). The internal heat transfer coefficients are considered constant in the conventionally heated case at $h_{ac} = h_{ap} = 8.5$ W m^{-2} K^{-1}. This corresponds to a weak, forced-air agitation, as is imposed in the experiments by an internal fan (Perdigones et al., 2006). In the case of IR heating, internal recirculation is imposed by natural convection, and the correlation used is $h_{ac} = 2.21(T_a - T_c)^{0.33}$, where T is in °C or K, and h_{ac} is in W m^{-2} K^{-1} (Papadakis et al., 1992).

The system of equations is closed by computing the thermal losses contributing to Q_{total}. Three additive terms are considered for either heating alternative:

Thermal losses caused by air renewal are given by:

$$Q_1 = \frac{C_{p\alpha}\rho_\alpha NV}{3600}(T_a - T_o) - 0.36NV(T_a - T_o) \quad (3)$$

Losses from the greenhouse cover are combined convective and radiative, and are given by:

$$Q_2 = h_{co}A_c(T_c - T_o) + \varepsilon_c A_c \sigma\left(T_c^4 - T_{sky}^4\right) \quad (4)$$

where the sky temperature is set equal to the outdoor air temperature, as in equation 1.

Finally, we estimate losses from the greenhouse floor as proportional to the area of the floor and to the temperature difference between the floor and deep soil. We further approximate the above two temperatures by the temperature of the plants and of the outside air, respectively. Thus, in line with the one-dimensional approach of the rest of the model, we write:

$$Q_3 = K_p A_p(T_p - T_o) \quad (5)$$

where the overall heat transfer coefficient from the greenhouse floor (K_p) depends directly on the thermal conductivity of the soil (k_{soil}) and is here taken as $K_p = 1.85$ W m^{-2} K^{-1} (Tiwari, 2003). A check of the magnitude of K_p is provided by the theoretical result $Q_3 = k_{soil}(2D)(\Delta T)$, which is rigorously valid for an isothermal circular floor of diameter D (Carslaw and Jaeger, 1959). Combining the two equations and using $A_p = 4.26$ m^2 and $k_{soil} = 1$ to 2 W m^{-1} K^{-1} (dry to wet soil, respectively) results in the range $K_p = 1.1$ to 2.2 W m^{-2} K^{-1}. The total thermal losses are then estimated as $Q_{total} = Q_1 + Q_2 + Q_3$.

Summarizing, in the case of forced-air heating, it is assumed that $T_a = T_p$ and equation 1 is solved for the unknown cover temperature (T_c). In the case of IR heating, the system of equations (eqs. 1 to 5) is solved for T_a and T_c. These are nonlinear, algebraic equations that can be treated with standard numerical schemes. We have used one of the NAG routines contained in Matlab.

The presented model incorporates several assumptions implicit in generic models for the estimation of greenhouse heating needs. The most important of these assumptions are briefly discussed in the following paragraphs. The assumption that the equivalent sky temperature is equal to the outside air temperature, although it is a standard simplification used in many similar models, may lead to a significant underestimation of heat losses during a clear (cloudless) night, since the actual sky temperature can be 5 to 15 degrees lower (Adelard et al., 1998). To assess the importance of this assumption, the present calculations were repeated using a sky temperature 6 degrees below that of the outside temperature (Tiwari, 2003). Although the calculated energy losses increased by nearly 15%, in the worst case, the percentage of energy savings for the infrared system in comparison to the forced-air system did not change by more than 4%. Thus, the present simplified treatment of the sky temperature does not weaken the conclusions of the work about the improved efficiency of IR heating. An additional argument in favor of the substitution of T_{sky} by T_o comes from the observation that a significant part of the greenhouse radiation (mostly from the walls) faces not the sky but the ground and adjacent plants or buildings. All of the above surfaces are probably at a temperature much closer to T_o than to T_{sky}. However, the key point is that the true sky temperature is unavailable in most cases, so generic models estimating greenhouse heat losses based on a mean sky temperature are subject to increased uncertainty.

Another assumption implicit in the model is the omission of the heat losses from the crop to the sky by radiation in the calculation of total losses. This may be significant for plastic cover materials, but since our data are taken in a glass greenhouse, this term can be reasonably considered negligible (Nelson, 2003).

The greenhouse energy balance does not take into account two effects related to mass transfer phenomena: transpiration of water from the canopy, and condensation of water on the cover. Either or both phenomena may become significant under different conditions, but our experimental observations indicate that neither is of importance to the present study. More specifically, transpiration is a minor effect during night-time mainly because the stomata of the leaves are practically closed, presenting a significant bulk mass transfer resistivity. In addition, the driving force for transpiration is weak, because the relative humidity of the inside air is typically high during the night, although it is always lower than that of the outside air (see next paragraph).

Condensation is also of limited importance during most of the night because, due to the lack of significant transpiration, the absolute humidity does not vary strongly between inside and outside air. Incoming air encounters the canopy and cover surfaces that typically have temperatures higher than that of the ambient, and thus the relative humidity of the air decreases. The only time period when condensation is actually observed is during the steep temperature drop accompanying sunset. The droplets that then accumulate on the cover walls reach equilibrium without draining to the ground, and gradually re-evaporate. Thus, condensation is a transient phenomenon at the beginning of the night cycle that may temporarily alter the heat capacity of the cover walls but does not affect the average heat losses.

The characteristics of the presented model are common to most models used in the estimation of heat losses from greenhouses and buildings. These models typically rely only on average values of outside temperature and wind velocity, and incorporate reasonable assumptions for all the other variables. In engineering calculations, the uncertainties introduced are compensated by suitable design factors used for the design of a heating system, whereas suitably implemented control systems ensure the energetically and economically efficient operation of actual heating systems. In relation to the scope of the present work, the impact of these assumptions on the comparison of the IR and forced-air heating systems is expected to be significantly less than the uncertainties they introduce, since they would influence the estimated heat losses for both cases in a similar manner.

Uncertainties in the above thermal calculation stem mainly from two sources. First, the overall heat transfer coefficients, although taking into account the specific characteristics of the reference greenhouse (wind conditions, orientation, materials, etc.), are derived from correlations that may be trusted with a precision $P = \pm 15\%$. Factors contributing to these uncertainties include averaging of properties and conditions, geometric characteristics, material degradation, and unaccounted for effects (Campbell, 1977; Chapman, 1984). Second, radiation losses are based on a sky temperature equal to the outside air temperature, an approximation representative of cloudy nights. During a night with clear skies, these losses will be significantly higher. From this second error source, a bias of up to $B = 25\%$ is anticipated (Duffie and Beckman, 1991). Thus, the overall uncertainty is estimated as:

$$U = \sqrt{P^2 + B^2} = 30\%$$

It is worth noting that the above uncertainty sources enter similarly in the calculations of the forced-air and the IR heated greenhouse. Thus, a possible error in the two calculations will be in the same direction, and the predicted improvement by the substitution of forced-air heating with IR heating will be very weakly affected.

Experimental Setup and Procedure

The experiments were performed in a pilot span greenhouse (fig. 1) and involved systematic measurement and recording of the parameters that determine the interior microclimate. The exterior climatic conditions were also recorded by a properly equipped meteorological station. The cultivation tested was lettuce, arranged with a density of 20 plants per square meter.

The experimental greenhouse is constructed of an aluminum framework, with 3 mm thick glass sheet as covering material. Its dimensions are 2.13 m width, 2.00 m length, 1.00 m eave height, and 1.50 m total height to the top. The base area of the greenhouse (A_p) is 4.26 m^2, the area of the cover (A_c) is 14.05 m^2, and the volume of the greenhouse (V) is 5.33 m^3. The design of the greenhouse has taken into consideration several constrains, such as the similarity to real production greenhouses, the implementation of the measuring equipment, and the efficient cultivation and servicing of the plants. A real greenhouse, typical for the geographic location, with 25 m length, 6.4 m width, and 5.3 m height, was used as a reference. The geometrical characteristics of the experimental greenhouse compare reasonably well with those of the reference greenhouse (cover area = 432 m^2, base area = 160 m^2, volume = 744 m^3). The corresponding characteristic linear length ratios are in the range 5.2 to 6.1, whereas the cover to base area ratios are 2.7 for the reference greenhouse and 3.3 for the experimental greenhouse. Since flow and thermal similarity depend on several factors besides the geometrical characteristics, the quality of the model is important for the transfer of the experimental findings to real production greenhouses.

The experimental greenhouse is of very solid construction and located in an area characterized by low winds, justifying the low rate of air infiltration selected. Sealing is significantly better than average, and the roof opening is the only one used during winter but never during the night. For the 80 days of operation of both configurations, the average wind velocity was lower than 1 m s^{-1} for 90% of the nights, whereas the maximum measured wind speed reached approximately 4 m s^{-1} (fig. 2). The number of air changes per hour (N) is estimated as 1 h^{-1}, referring to a new greenhouse of very solid construction and better than average sealing of windows and openings.

The experimental greenhouse was equipped with heating and ventilation systems. Two alternative heating systems were available: (1) a forced-air unit with two power levels (1 and 2 kW) and a small fan that promoted air mixing, and (2) an IR heating system consisting of four lamps with blown-bulb reflectors (1 kW total power, 50° beam angle) placed at the greenhouse corners and an elevation of 1 m above the

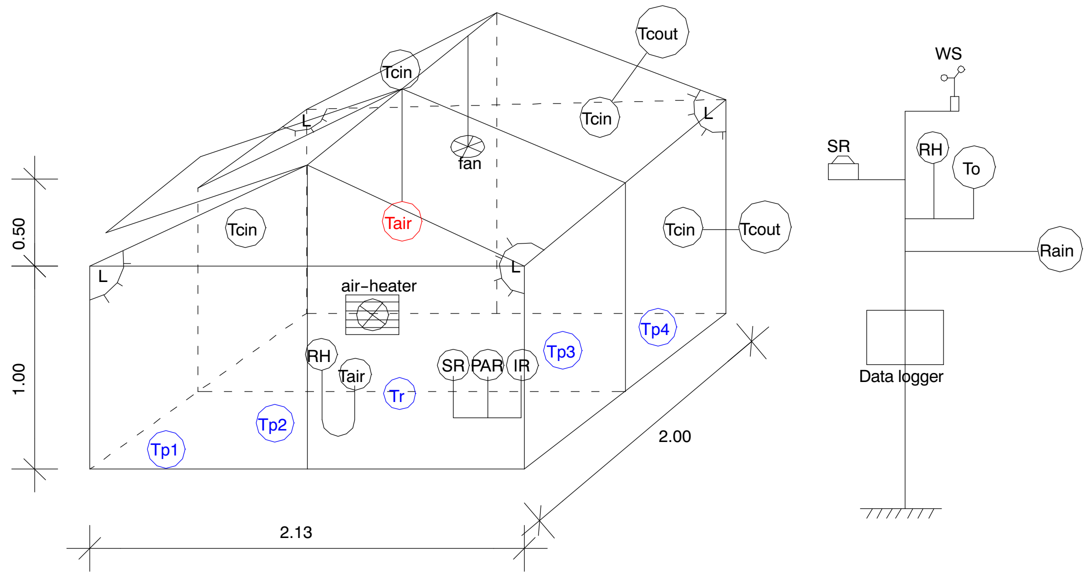

Figure 1. Experimental greenhouse and meteorological station.

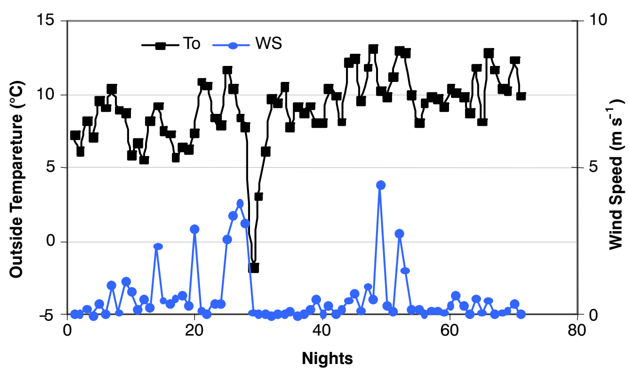

Figure 2. Average nightly wind speed variation during the measurement period.

plants (fig. 1). Ventilation during winter daytime is accomplished, whenever the reference temperature rises above 22°C, by a forced-air flow of 68 m^3 h^{-1} imposed by a ventilator and simultaneous opening of the roof window. During summer, the door is also left open to increase air exchange.

The microclimate of the inside environment was monitored and controlled with several sensors (fig. 1) including a silicon-type pyranometer (model SP-LITE, spectral range 400-1100 nm, accuracy ±5%, Kipp & Zonen, Delft, The Netherlands), a thermopile-type pyranometer (model CMP3, spectral range 300-3000 nm, accuracy ±5%, Kipp & Zonen), a "photosynthetically active" radiometer (model PAR-LITE, spectral range 400-700 nm, accuracy ±5%, Kipp & Zonen), a temperature and relative humidity probe (model S3CO3, accuracy ±1% RH, ± 0.3 K, Rotronic, Huntington, N.Y.), and a number of thermocouples (type T, copper-constantan, 0.5 mm diameter, accuracy 0.5°C) distributed throughout the greenhouse, including five thermocouples measuring the plant temperature. All thermocouples were shaded from the radiation of the IR lamps with suitable reflective surfaces. Thermocouples measuring plant temperature were attached on the lower sides of the lower leaves. Since transpiration during the night is minimal and heat transfer underneath the leaves is accomplished mainly by free convection, the actual temperature of the leaves is expected to be very close to the measured plant temperature. The meteorological station recording the outdoor environmental conditions was equipped with a data logger with two relay analog multiplexer units and sensors, including a silicon-type pyranometer (model SP-LITE, spectral range 400-1100 nm, accuracy ±10%, Kipp & Zonen), a temperature and relative humidity probe (model MP101A, accuracy ±1% RH, ±0.2°C, Rotronic), a rain gauge (model 52203, accuracy 2%, R.M. Young, Traverse City, Mich.), and an anemometer (model A100K, accuracy 1%. Windspeed Ltd., trading as Vector Instruments, Rhyl, U.K.) with threshold sensitivity 0.15 m s^{-1}. All the above instruments and sensors were calibrated either using corresponding certified instruments as reference or using standard samples traceable to European or International standards.

The parameters of central interest to the present study were the temperatures of the plants, of the inside and outside air, and of the glass cover, as well as the outdoor wind speed. As their values generally change with time, the data were scanned every minute, and 10 min averages were computed and recorded on a 24 h basis by the station's data logger. Attention was focused on greenhouse performance during the night, when the heating system was automatically turned on. Overall night-time mean values were also computed from the respective time series, based on the interval between the steep temperature changes at sunrise and sunset.

Preliminary measurements indicated that the plant canopy temperature was spatially uniform within the central area of the greenhouse base, but close to the walls temperatures as much as 1.5°C lower were measured during cold nights (as shown in fig. 3 for a typical cold night). Therefore, canopy temperatures were represented by a reference temperature ($T_p = T_r$) taken at the center of the greenhouse that also served as the measured variable for the thermal control of the greenhouse. For lettuce cultivation, the reference temperature was set to T_p = 15°C ±1°C, i.e., the heating system turned on when T_p dropped below 14°C and turned off when T_p exceeded 16°C. The cover temperature was also found to vary a little with location, and thus an average of the relevant thermocouple data is reported. Finally, the outdoor temperature (T_o) and

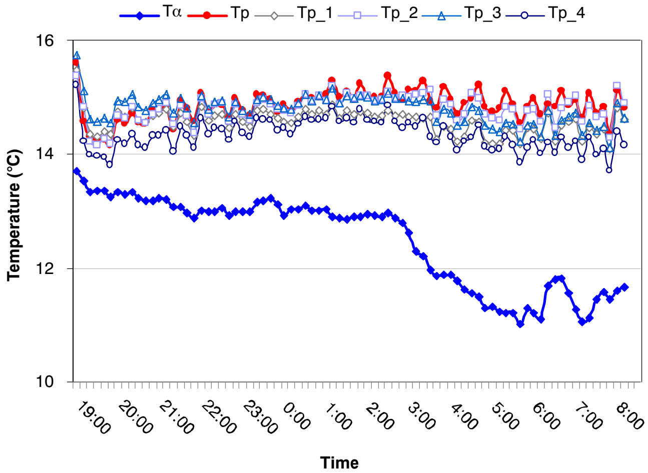

Figure 3. Temperature variation at the canopy during the typical night (exact location of thermocouples is shown in fig. 1; for outside temperature, see fig. 8).

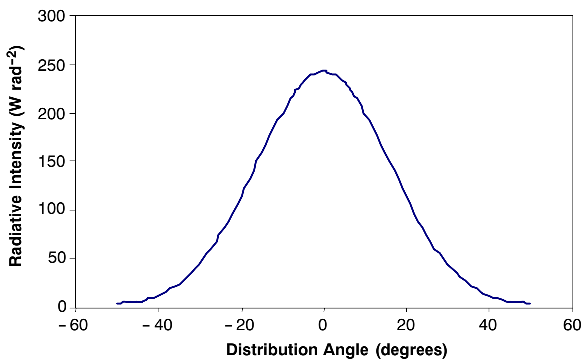

Figure 4. Radiative intensity distribution.

wind speed (u_w) were measured at a height of 2.50 m above the ground, and the inside air temperature (T_a) was measured at the center of the greenhouse and at 1.20 m above the ground. During the daytime, greenhouse overheating was avoided by automatic operation of the ventilation system as soon as T_a rose above 22°C.

The IR lamps that were used were made of hard glass with high mechanical strength and resistance to thermal shocks (e.g., water splashes). The lamps emit a high proportion of infrared light and a low proportion of visible light. The distribution of the radiation intensity of the type of lamps used has been measured with the thermopile-type pyranometer at a distance of 1 m (fig. 4). Lamps were positioned at the four corners of the greenhouse, 1 m above the ground. The heat flux patterns presented in figure 5 (as estimated with Mathematica software on the basis of the orientation and intensity distribution of each lamp, and as measured in the greenhouse with the thermopile pyranometer) indicated a more or less uniform pattern for a large area around the center of the greenhouse, whereas the values decreased in a small range close to the walls. This distribution proved sufficiently uniform for the cultivation of the plants, which grew evenly in the greenhouse. It is expected that in larger greenhouses, which provide more design options for the number, location, and orientation of the heating sources, superior distribution of the heat flux can be achieved. The key characteristic of the IR lamps with respect to the present study is their radiative

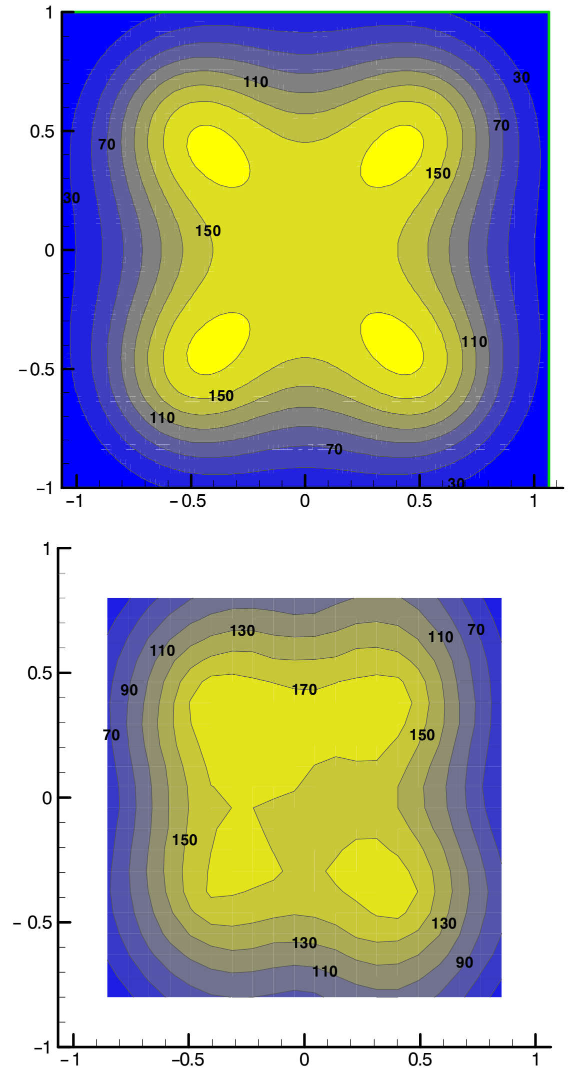

Figure 5. Distribution of the energy flux (W m^{-2}) from the IR lamps at the canopy level: (top) as estimated with Mathematica software, and (bottom) as measured in the greenhouse.

efficiency (η), i.e., the percent of electrical energy consumed that is delivered as thermal radiation. Surface integration of the data shown in figure 4 gives the value η = 60%. The remaining 40% of the input energy is predominantly lost by convection to the surrounding air. Losses by radiation from the lamp reflector to the greenhouse cover above the lamp are estimated as at least an order of magnitude smaller and are neglected in the simulation model. Given that the lamps operate intermittently, making up for all the energy losses of the greenhouse, the convective energy loss term ($Q_{lamps,conv}$) in equation 2 was modeled as:

$$Q_{lamps,conv} = (1-\eta)(Q_1 + Q_2 + Q_3) \qquad (6)$$

Experimental Results

Experimental results collected during a period of 80 days comprise 52 nights with infrared heating and 20 nights with forced-air heating with recirculation. In the beginning of the thermal period, the forced-air heating system was used with no recirculation for about 8 nights. The results for each mode of operation indicate a rather similar pattern, and typical nights were selected from each set, having relatively low out-

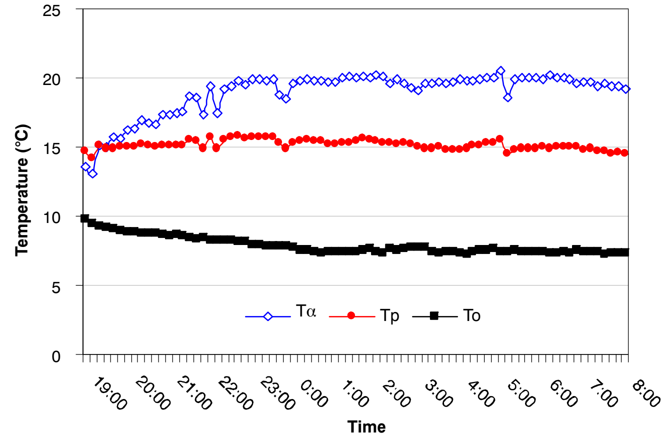

Figure 6. Hot air heating system without operation of the recirculation fan.

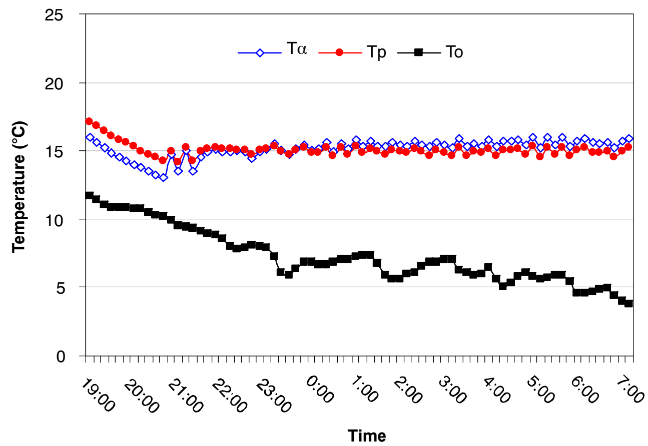

Figure 7. Hot air heating system with operation of the recirculation fan.

side temperatures but with no otherwise extreme conditions (i.e., the wind speed was low, and there was no rain).

Typical results of the night-time temperature variations are presented here, characterizing each mode of operation. Furthermore, the mean nightly values for the 52 nights of IR operation and the 20 nights of forced-air heating with recirculation are presented, and the relative merits of IR heating are outlined.

Night-Time Temperature Variation

First, the forced-air heating system is considered, and the variations in outdoor, indoor, and reference temperatures are compared. Recall that the recorded temperatures are 10 min averages, and note that the time duration plotted in the following figures corresponds to one night. Figures 6 and 7 refer to forced-air heating without and with the recirculation fan, respectively. In both figures, it is confirmed that the reference temperature (T_p) is successfully kept at the desired value of 15°C ±1°C. However, figure 6 indicates that, without forced recirculation of the interior air, very significant stratification is established. More specifically, the air temperature at the measurement location (1.20 m above ground level, 0.3 m below the top of the roof) was observed to rise up to 5°C above the value targeted for the plants. It is further noted that this trend increased as the outdoor temperature dropped.

Results with the air recirculation fan turned on are shown in figure 7. They indicate that stratification is completely alleviated, and the internal air temperature always remains at a slightly higher value than that of the plants. This experimental observation justifies the assumption of equal temperatures of plants and inside air, adopted in the theoretical model of the conventionally heated greenhouse.

Next, operation of the greenhouse with the IR heating system is considered, and a typical night-time variation of the recorded temperatures is shown in figure 8. The IR lamps are successful in keeping the plant temperature at the desired value, despite the variation of outdoor temperature. However, the most important observation is that the inside air always remains cooler than the plants, providing an initial indication in favor of the IR heating concept. It is further observed that, as the outdoor temperature drops, the inside air temperature drops as well. Thus, the difference $T_a - T_o$, which determines thermal losses, does not increase in proportion to the decrease in the outdoor temperature. This is unlike what happens with forced-air heating, where the above difference increases proportionally to (with recirculation) or even faster than (without recirculation) the decrease in T_o.

Effect of Outside Temperature

The merit of IR heating is related to the ability of the system to sustain the desired temperature of the canopy while maintaining the greenhouse air at a significantly lower temperature. Data presently used to compare the two alternatives are mean night-time temperatures, computed as outlined in the Experimental Results section. More specifically, figure 9 shows the difference between the inside and outside air tem-

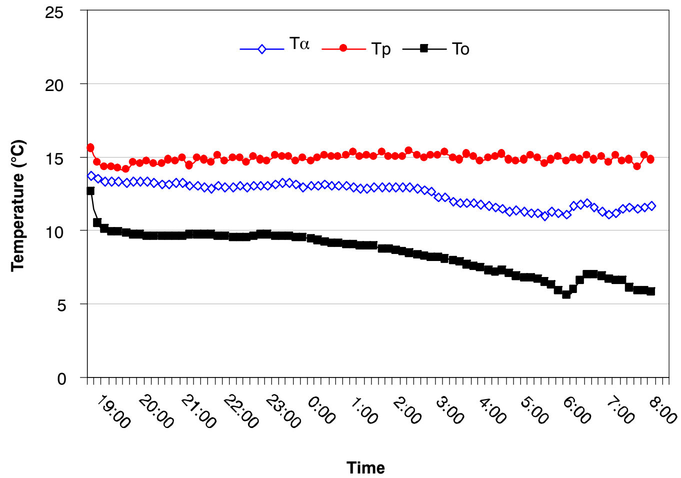

Figure 8. IR heating system with lamps and reflectors.

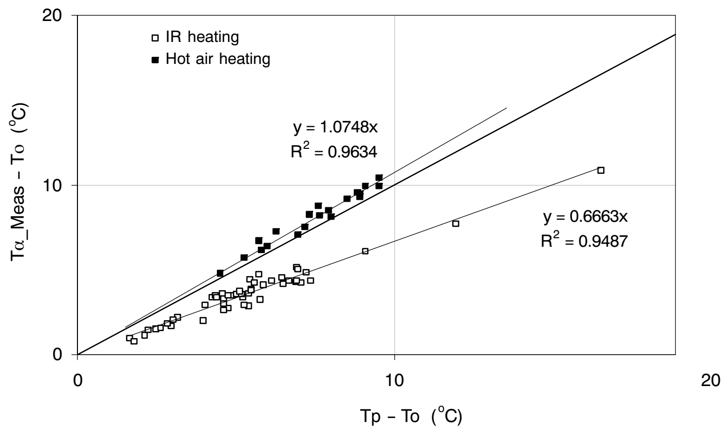

Figure 9. Temperature differences indicative of the overall heating losses for IR and forced-air heating.

peratures, indicative of the overall greenhouse heat losses, as a function of the difference between the plant and outside air temperatures. Open symbols correspond to infrared heating, and closed symbols correspond to forced-air heating. The diagonal represents the limit at which the inside air temperature is equal to the canopy temperature (recall that, for the lettuce crop, the plant temperature was set to $T_p = 15^\circ\text{C} \pm 1^\circ\text{C}$). Both IR and forced-air heating data are satisfactorily fitted by straight lines, indicating the expected result that thermal losses increase proportionally to a drop in outside temperature.

The potential benefit of IR heating is clearly demonstrated in figure 9. The inside air temperature is always lower for IR heating compared to the reference temperature indicated by the diagonal. On the other hand, the forced-air heating values are always slightly above the diagonal. Comparison of the slopes of the two fitting lines provides a rough estimate of the accomplished energy savings of approximately 40% [(1.07 – 0.67)/1.07]. It is also evident that the absolute energy saving increases linearly with the difference between the plant and outside temperatures, and thus makes infrared heating particularly appealing for colder climates. An important question is whether we expect these results to improve significantly if sources with higher radiative efficiency are used. A quantitative estimate of expected benefits is provided in the next section through the use of a simulation model.

Theoretical Predictions

Reliability of Model

Our theoretical model is designed to predict steady-state values of the inside air and cover temperatures for steady external conditions. Next, we use as inputs the measured mean night-time conditions and compare the predicted inside air and cover temperatures to the measured mean values. The predictive capability of the model is shown in figures 10 and 11, where mean inside air temperatures and cover temperatures, respectively, are plotted for all 50 nights with IR heating. The general impression is that the model is quantitatively reliable. More specifically, the average difference between the data and the predictions is 0.37°C, and the standard deviation of the differences is 0.59°C. Deviations are mainly attributed to the use of mean nightly values and the assumption that the sky temperature is equal to the outside air temperature. Similar remarks hold for the predictive capability of the model for forced-air heating, as shown in figure 12.

Prediction of Energy Losses

The theoretical model was used to predict the greenhouse energy losses for each of the two alternative heating systems. Computations were undertaken for a range of outside temperatures (–6°C to +10°C) and a reference air speed of 0.5 m s^{-1}. The setpoint temperature for plant growth was 15°C. Results are shown in figure 13 for forced-air heating and in figure 14

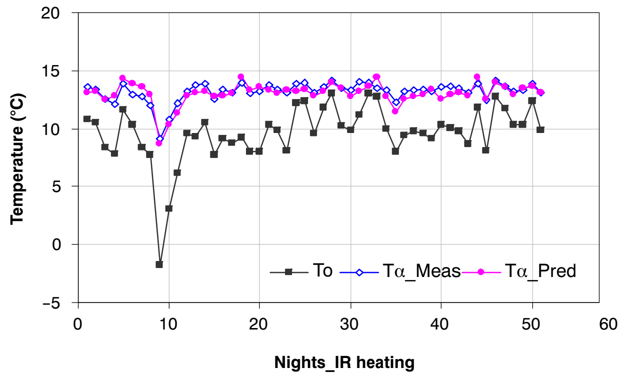

Figure 10. Mean nightly measured (Tα_Meas) and predicted (Tα_Pred) inside air temperature for nights with IR heating.

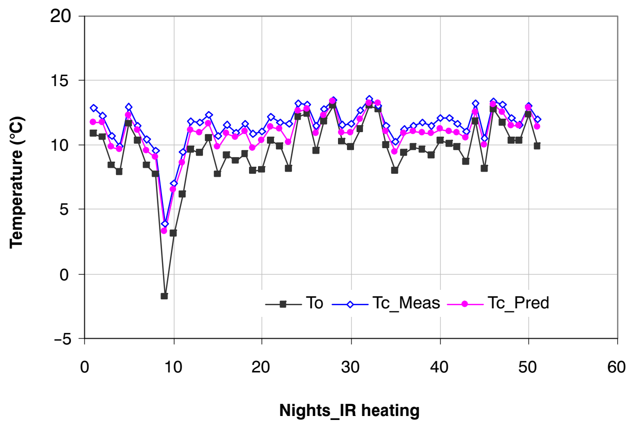

Figure 11. Mean nightly measured (Tc_Meas) and predicted (Tc_Pred) cover temperature for nights with IR heating.

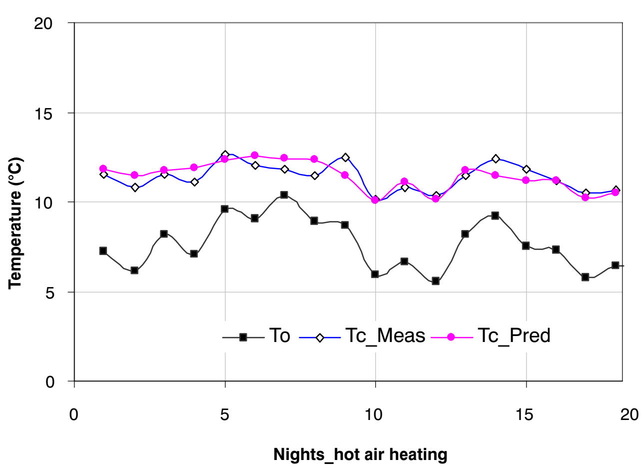

Figure 12. Mean nightly measured (Tc_Meas) and predicted (Tc_Pred) cover temperature for nights with forced-air heating.

for IR heating, and losses are grouped in three categories: (1) infiltration and convection to the soil, (2) radiation through the cover, and (3) convection from the cover. Measured heat losses, which indicate a good agreement with the predictions, are also included in these figures. With the exception of the conductive losses through the soil (that remain by definition identical), all other thermal losses are appreciably reduced in the case of IR heating. The total result is a 45% to 50% reduction in the energy demand for the IR-heating case, in comparison to the forced-air heating case, during

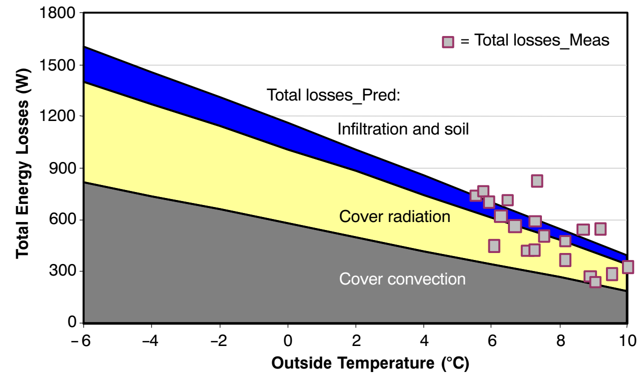

Figure 13. Total energy losses during forced-air heating.

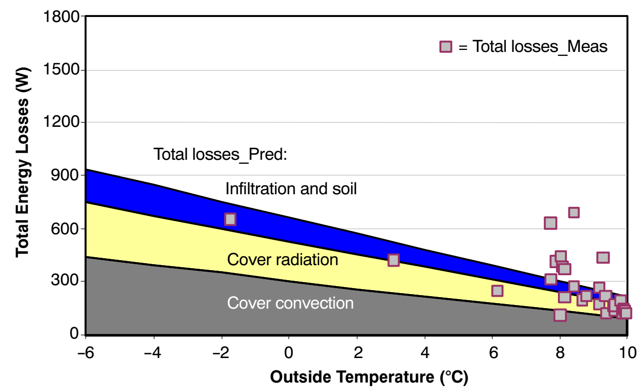

Figure 14. Total energy losses during IR heating.

steady-state operation. It is worth mentioning that convective and radiative heat losses from the cover during IR heating, although appreciably reduced in absolute value as compared to forced-air heating, still represent the most significant component of the total energy demand and are as high as 85% of the total. Hence, research for novel cover materials with improved insulation properties will remain a key goal even if IR heating is adopted.

Parametric Studies

The effect of the decrease of the outside temperature on the efficiency of IR heating in comparison to conventional forced-air heating has been already discussed on the basis of figure 9. Next, the theoretical model is used in parametric studies to examine the effect of other conditions and scenarios on this comparison.

Effect of Wind Speed and Infiltration Losses

To assess the effect of convective and infiltration losses due to a change in wind speed, the model was used to simulate an outdoor temperature (T_o) of 0°C and a target plant temperature (T_p) of 15°C with varying wind speed (u_w) over the range 0.5 to 5.0 m s^{-1}. For the calculations, infiltration losses were assumed to be a linear function of the wind speed, setting the number of air exchanges per hour (N) equal to the velocity magnitude in m s^{-1} (N = 0.5 to 5.0 h^{-1}). The combined effect of a higher convective heat transfer coefficient between the cover and outside air along with the increase in air changes due to a higher wind speed resulted in increased heat losses. As shown in figure 15, this increase was lower in the case of the IR-heated greenhouse, compared to the baseline

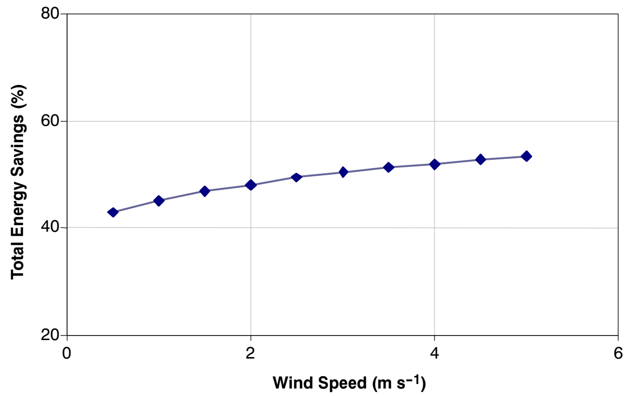

Figure 15. Effect of wind speed (WS) variation on energy saving with IR heating.

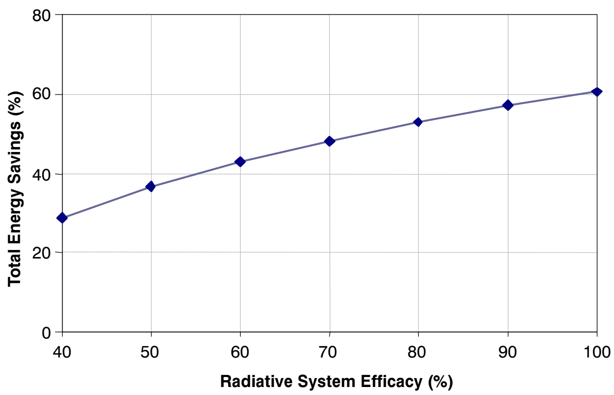

Figure 16. Effect of IR source radiative efficacy on energy saving under IR heating.

case of forced-air heating, increasing the total energy savings to more than 50% for wind speeds over 3.0 m s^{-1}.

Effect of IR Sources Radiative Efficacy

The theoretical model was finally used in a parametric study to examine the potential benefits from the development of IR sources with improved thermal efficiency. Recalling that the experimentally determined radiative efficiency (η) of the lamps used was 60%, we examined the range η = 40% to 100% for representative outdoor conditions $T_o = 0^\circ$C, u_w = 0.5 m s^{-1}, and $T_p = 15^\circ$C. The results in figure 16 show the percentage energy saving achieved by IR heating (over the baseline case of forced-air heating) as a function of the radiative efficacy of the IR source. It is evident that IR heating will become significantly more attractive with the availability of improved sources.

Concluding Remarks

Measurements of indoor and outdoor conditions during typical cold nights in central Greece were obtained in an experimental greenhouse equipped with forced-air heating and infrared heating. A theoretical model was developed, initially validated by comparison with measured data, and subsequently used to explore potential benefits of IR heating.

Greenhouse operation with forced-air heating resulted in inside air temperatures equal to or higher than the target value for the plants, whereas operation with IR heating allowed the inside air temperature to be several degrees cooler without a drop in plant temperature. The temperature difference between the inside air and the plants during IR heating was found to increase linearly with the drop in outdoor temperature, pointing to the potential appeal of this alternative for colder climates.

Parametric studies based on the developed theoretical model indicate that, with commercially available IR sources, energy savings over conventional forced-air heating in the order of 45% to 50% can be expected. These savings are expected to increase for higher wind velocities, when overall losses increase due to the increased convective heat transfer between the cover and outside air and due to the higher rate of infiltration. It is further predicted that improvements in the radiative efficacy of IR sources will result in significant growth of these savings, rendering IR heating a more attractive alternative.

Infrared heating appears to be a promising option for greenhouse heating when short plants are considered. For non-planar crops, the even distribution of the heating flux seems to be the key engineering challenge. The use of radiant IR heaters in open cafes, although intended to heat humans, who have a superior temperature control system, indicates that the concept is feasible. The increase of the radiative flux at the more exposed parts of a plant is not expected to cause a negative impact on the plants, which experience much higher heat fluxes during sunshine than during the night in the current study. The limiting factor in these cases is whether this increase will result in a corresponding increase in heating losses and therefore in increased heating cost, rendering the comparison of IR heating with conventional forced-air heating less advantageous. In this regard, the development of more efficient IR sources may be a critical factor for the widespread use of IR heating. Another alternative that may be worth considering is the combined use of radiative sources that also increase the photosynthesis of the plants (artificial lighting), in which case the gains could be many fold.

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Nomenclature

A_c	= greenhouse cover area (m^2)
A_p	= area of plant canopy (growing area) (m^2)
A_s	= greenhouse floor area (m^2)
C_{pa}	= specific heat of air (J kg^{-1} $^\circ C^{-1}$)
h_{ac}	= convective heat transfer coefficient between inside ambient air and cover (W m^{-2} K^{-1})
h_{ap}	= convective heat transfer coefficient between plant canopy and inside air (W m^{-2} K^{-1})
h_{co}	= convective heat transfer coefficient between cover and outside air (W m^{-2} K^{-1})
K_p	= heat transfer coefficient for conductive losses through the soil (W m^{-2} K^{-1})
n	= radiative efficiency of the IR lamps
N	= number of air exchanges per hour (h^{-1})
$Q_{lamps,conv}$	= convective loss of IR lamps (W)
Q_{total}	= total thermal losses (W)
Q_1	= thermal losses from infiltration (W)
Q_2	= thermal losses by combined convection and radiation from the greenhouse cover (W)
Q_3	= thermal losses from conduction to the soil below the greenhouse (W)
T_a	= inside ambient greenhouse temperature (°C)
T_c	= temperature of the cover (°C)
T_o	= outside air temperature (°C)
T_p	= canopy temperature (°C)
T_{sky}	= temperature of the sky (K)
V	= greenhouse volume (m^3)
u_w	= wind speed (m s^{-1})

Greek Letters

ε_c	= emissivity of the cover
ε_p	= emissivity of the plants
ρ_a	= air density (kg m^{-3})
σ	= Stefan-Boltzmann constant (W m^{-2} K^{-4})