CALCULATION OF THE AMOUNT OF ENERGY RELEASED BY HEATING PIPES IN A GREENHOUSE AND ITS ALLOCATION BETWEEN CONVECTION AND RADIATION

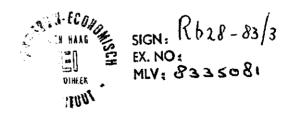
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ABSTRACT

The energy output from a heating system in a greenhouse is mostly known only within broad limits and for long periods of time.

The values for heat transfer coefficients calculated for a single pipe in laboratory conditions cannot easily be applied to conditions in the greenhouse where heating pipes are usually grouped within the canopy. The method described allows the heat transfer coefficient of a given pipe system to be estimated using simple temperature data, measured when the system is cooling. The parameters of the theoretical cooling function, i.e. the total heat transfer coefficient and the apparent thermal conductivity of the pipe system, can thus be derived by best-fitting. The small difference between the latter parameter and the thermal conductivity of water is discussed.

It is further shown that, when the effect of radiation is eliminated from the total heat transfer coefficient, the Nusselt number can be calculated for the system. The resulting relation is then applied to estimate the system's energy output and temperature when cooling.



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SYMBOLS

Capital

Gr = Grashof number =
$$\frac{gl^3}{v^2} \frac{T_s - T_f}{T_f}$$

$$H = energy (J)$$

$$J_n$$
 = Bessel function of order n

Nu = Nusselt number =
$$l/\delta$$

$$Pr = Prandtl number = V/K$$

$$Q = \text{heat flux } (W m^{-2})$$

Re = Reynolds number =
$$ul/v$$

Lower case

Italic

$$c$$
 = specific heat (J kg⁻¹)

$$d = diameter (m)$$

$$g$$
 = acceleration due to gravity (9.81 m s⁻²)

$$s$$
 = standard deviation

$$t = time (s)$$

$$u = air velocity (m s^{-1})$$

Greek

$$\alpha$$
 = heat transfer coefficient (Wm⁻²K⁻¹)

$$\delta$$
 = thickness (m)

$$\kappa$$
 = thermal diffusivity = $\lambda/\rho c$ (m²s⁻¹)

$$\lambda$$
 = thermal conductivity (Wm⁻¹K⁻¹)

$$v = \text{kinematic viscosity } (m^2 s^{-1})$$

$$\rho$$
 = density (kg m⁻³)

$$\sigma$$
 = Stefan-Boltzmann constant (5.67 $10^{-8} \text{ Wm}^{-2}\text{K}^{-4}$)

Subscripts

- a = air
- c = convection
- f = fluid
- m = mean
- r = radiation
- s = surface
- w = water

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To assess the energy balance of a greenhouse the amount of energy released by the heating system must be known. Moreover, since the canopy (and the greenhouse environment in general) does not necessarily make the same use of radiation and sensible heat, one should also be able to separately evaluate the amount of energy transferred in the radiative and convective mode respectively.

1. THEORY

In general, heat is being transferred in three ways: by radiation, convection, i.e. by air in motion, and conduction, which is the exchange of kinetic energy at molecular level in solids or still fluids. Moreover, two types of convection are important: forced convection, or transfer through an air stream (the rate of heat transfer thus depends on the velocity of the flow); and free convection, which depends on vertical displacement of air masses due to temperature gradients.

In the present study, radiative heat transfer between two bodies at temperatures T_1 and T_2 respectively will be estimated according to the much used linearization of the Stefan-Boltzmann formula, i.e.:

$$Q_r = \varepsilon \sigma (T_1^4 - T_2^4) \cong 4\varepsilon \sigma T_m^3 (T_1 - T_2) = \alpha_r (T_1 - T_2) \quad wm^{-2}$$
 (1)

where temperatures are in K and the subscript m denotes the mean of the two temperatures.

On the other hand, conductive heat transfer through an air mass in a greenhouse environment can be neglected due to the small conductivity of still air (λ_a = 2.53·10⁻² Wm⁻¹K⁻¹, for dry air at 15°C) which is many orders of magnitude smaller than an "apparent conductivity" for convection even at the fairly low air velocities experienced in a greenhouse environment.

Heat flux due to convection can be described formally by the equation for conduction of heat in a solid:

$$Q = -\lambda \nabla T \qquad Wm^{-2} \quad (2)$$

When such an equation is meant for heat transfer through a moving fluid, λ will rather be an "apparent conductivity" and the thickness of the layer where the temperature gradient has to be considered will be an "equivalent thickness", accounting for the net effect of the presence of a surface losing heat to the moving fluid.

Non-dimensional groups, such as the various numbers, help in solving particular transfer problems. Thus, the Grashof number (Gr) is a measure of the importance of buoyancy due to the difference in temperature between the surface and the fluid, while the Reynolds number (Re) describes the effect of the surface itself on turbulence in the fluid. At a large Reynolds number, heat transfer is due more to the fluid stream than to buoyancy and forced convection takes place, while with large Grashof numbers ($Gr > Re^2$) the reverse is true, and the prevailing transfer mechanism is free convection.

In a greenhouse climate it is often a case of mixed convection, since it is usually impossible to specify the prevailing driving force for heat transfer. Another non-dimensional group, the Nusselt number (Nu), can be helpful, being the ratio between a characteristic dimension of the surface to the thickness of an "equivalent boundary layer" through which the transfer of heat takes place. The Nusselt number is a function either of the Grashof number in free convection or of the Reynolds number in forced convection.

In the intermediate region it is a good practice to calculate Nu as a function of either and use the largest Nu to estimate the rate of heat transfer. The heat transfer coefficient for convection $\alpha_{_{\rm C}}$ is easily related to Nu. In fact (2) can be written for one-dimensional convection through the equivalent boundary layer of finite thickness δ

$$Q_{c} = \lambda \frac{(T_{s} - T_{f})}{\delta}$$
 Wm⁻² (3)

where T_f denotes the prevailing temperature of the fluid at distance δ from the surface, and the sign is positive for fluxes leaving the surface. After substitution of Nu = l/δ , (3) becomes

$$Q_{c} = Nu \frac{\lambda}{l} (T_{s} - T_{f}) = \alpha_{c} (T_{s} - T_{f})$$
 wm⁻² (4)

where I is a typical dimension of the surface.

Since radiative heat loss is also present when a difference in temperature is experienced, a surface immersed in a fluid loses heat according to

$$Q = (\alpha_r + \alpha_c) (T_S - T_f) = \alpha (T_S - T_f)$$
 wm⁻² (5)

where a total heat transfer coefficient α is defined as the sum of the radiative and convective coefficients.

If it is assumed that the energy input in a greenhouse by radiation of the heating system is known when the temperature of the heating elements is known (eq.(1)), and conduction is neglected, the problem of estimating the energy released by the heating system is reduced to the knowledge of $\alpha_{\rm C}$.

First, it has to be decided whether the prevailing transfer mechanism is free or forced convection, according to the method briefly described above.

For heating pipes the diameter is a natural choice for the typical length; then for pipes of diameter $d \approx 0.06$ m, 15° C warmer than the surrounding air, and an air velocity $u \approx 0.1$ ms⁻¹, Re $\approx 4 \cdot 10^2$ and Gr $\approx 4 \cdot 10^5$ (kinematic viscosity of air $v_{\rm a} = 1.5 \cdot 10^{-5} {\rm m}^2 {\rm s}^{-1}$). It appears then that free convection should be more important than forced convection and Nu should be calculated as a function of Gr.

There is agreement in the literature that for systems with ${\rm Gr}$ <10 9

$$Nu = C(Pr \cdot Gr)^{1/4}$$
 (6)

with C a constant and Pr the Prandtl number of the fluid (0.71 for air). The value of the constant depends on the shape of the surface losing heat, and mostly there is no complete agreement on it: Jodlbauer (1933) proposed C = 0.455 for long horizontal cylinders, while McAdams (1954) suggested a value of 0.525. Monteith (1975) quotes C = 0.523 from previous works and Stoffers (1976) calculated C = 0.614 from measurements in a greenhouse environment.

Since, as substitution of (6) in (4) shows,

$$\alpha_{\rm c} = c \frac{\lambda}{d} \left(\text{Gr-Pr} \right)^{1/4} \qquad \qquad w_{\rm m}^{-2} \kappa^{-1} \quad (7)$$

an inaccuracy of 30%, say, in C means the same inaccuracy in the convective heat flux; it appears then that a good estimate of C for a given system is the bottleneck to the knowledge of its energy output.

2. MATERIALS AND METHODS

As stated above, an estimate of the energy input from the heating system and of its allocation between convection and radiation, was a much needed part of a larger experiment aimed at defining the energy balance of a greenhouse. The experiment is described elsewhere: Stanghellini (1981); Bot et al. (1983); van 't Ooster (1983); Stanghellini (1983), and only the relevant instrumental set-up for the present subject will be described here.

The greenhouse is a Venlo-type, 8 spans 3.20 m each, 22 m long, oriented E-W. Heating is provided by hot water circulated in pipes (diameter: 5.735·10⁻² m) laid a few centimeters above ground, one element (supply and return lines) for each canopy row. A secondary system is at gutter level, one pipe for each span. To prevent side effects, a set of four pipes one above the other is brought to each wall. The wall network is directly coupled to the ground one, while the gutter system is switched in only if necessary. The total length of the ground network is 588 m, that of the gutter one 164 m, while the wall system is 433 m in length. This adds up to a total volume of water in the pipe system of 3.06 m³. The water in the pipe system is heated in a heat exchanger in which steam is circulated. A buffer vessel (0.4 m³) is placed in parallel between the heat exchanger and the pipe system. Water is continuously circulated in the pipe system, even when the valve between it and the heat exchanger-vessel system is closed.

The total energy input to the greenhouse was measured in two ways:

- a. by measuring the amount and temperature of condensed water flowing at the outlet of the heat exchanger in the greenhouse;
- b. by measuring the water flow to the pipe system and temperature decay between the inlet and outlet of that system.

It soon appeared that energy input estimates with the first method were by far too high, leading to the conclusion that some steam is being condensed already at the inlet of the heat exchanger; no further attempt was made to improve the reliability of this method. For the second one, on the other hand, a commercial device was installed, which worked properly for about three months. It ceased working for reasons

still unclear; it is, however, possible that a fault in the heating system, when boiling water was pushed into the pipes, has damaged the sensor of the flow meter. The temperature of the ground pipeline system was measured by two thermocouples (on the supply and return lines) at one section in the middle of the house; that of the gutter one was measured in the same way, at about the same spot, while the temperature of the wall system was not checked.

The temperature of the air was measured by Assmann aspirated psychrometers at various heights above ground. Air temperature referred to hereafter was measured 0.8 m above the pipes.

The air velocity was measured by four hot-bulb anemometers (den Ouden, 1958) also at various heights above the pipe system. Hereafter the average of the four values will be used for each measurement.

The total heat transfer from the pipe system could be calculated while warming it up and then letting it cool down. In fact, the temperature (T) as a function of time (t) and distance from the axis (x) of a circular cylinder of radius (r) and infinite length, losing heat at its surface to a medium at zero temperature according to a linear law, as eq. (5), is known (Carslaw and Jaeger, 1948)

$$\frac{\mathbf{T}}{\mathbf{T}_{0}} = \sum_{n=1}^{\infty} e^{-\beta_{n}^{2} \frac{K}{r^{2}}} \frac{t}{\beta_{n}^{2} + \lambda^{2}} \frac{J_{0}(\frac{\mathbf{x}\beta_{n}}{r})}{J_{0}(\beta_{n})}$$
(8)

where T_0 is the constant temperature of the cylinder at time t=0; β_n are the roots of the transcedent equation

$$\beta J_1(\beta) = AJ_0(\beta) \tag{9}$$

and A is related to a by:

$$A = \frac{r}{\lambda} \alpha = r \frac{\alpha}{\kappa \rho \sigma} \tag{10}$$

i.e. a can be calculated from A and K if they are known.

This was achieved by defining a "mean pipe" with a surface temperature which is the mean of the measured ones, and then finding by the least square method the best fit of (8) on the cooling portion of the temperature vs. time function. It should be noted that when temperature is measured at a distance x from the pipe axis, equal to its radius r,

Table 1 - Average values over each time interval, on which pipe surface was cooling, used for the best fit procedure.

Symbols are as used in the text, temperatures are in °C and times in hours and minutes.

	Time in	terval				
Date	from	to	T a	T _s	T-T	T m
Feb.20, '81	18.10	19.40	14.80	42.85	28,05	28.83
11	20.10	21.30	14.64	44.35	29.70	29.50
Ħ	22.00	23.30	14.39	45.49	31.11	29.94
Feb.21, '81	0.00	1.20	14.30	45,64	31.34	29.97
11	1.50	3.10	14.13	46.23	32.11	30.18
D	3.40	5.00	14.60	45.90	31.30	30.25
o	5.30	6.50	14.49	46.30	31.81	30.39
n	7.20	8.10	16.04	52.28	36.24	34.16
Mar.14, '81	19.40	23.40	14.07	28.11	14.04	21.09
Mar.15, '81	0.00	3.50	14.33	28.67	14.34	21.50
11	4.20	7.20	15.31	32.75	17.44	24.30
Apr.15, '82	22.44	1.26	17.14	33.13	16.00	25.13
Apr.16, '82	1.56	4.23	17.06	34.64	17.58	25.85

the last factor on the right-hand side of (8) equals one. Moreover, only the first term of the summation was fitted.

Since the surrounding air could not be considered at constant temperature, the difference between the measured pipe and air temperature was used for each step. Anyhow, in order to decrease inaccuracy due to this, the calculations were made only on night-time values, thus minimizing air temperature variations not due to pipe heating.

Calculations were performed on 13 cooling periods of three nights with various climatic conditions. Average values for each cooling are summarized in Table I. The strict relationship should be noted between air velocity and pipe-air temperature difference, shown in Fig. 1 for the same data. This is actually an indication that most of the air movement is due to buoyancy. It appears that a function of air velocity could as well be substituted in eq. (7) instead of the function of the difference in temperature, leading to the suggestion that in such conditions Nu could also be calculated as a function of the Reynolds number. In any case, this is more an academic question than a practical one, since it is easier and cheaper to measure temperatures than air velocities at such low levels.

The regression coefficients calculated for the procedure outlined above were always above r = 0.99 and no influence of the time interval between subsequent data was revealed; data every three or ten minutes were used. The magnitude of the 2nd order term in (8) was calculated a posteriori: its contribution was about 0.5° C in the first few steps and became quite negligible after a few minutes.

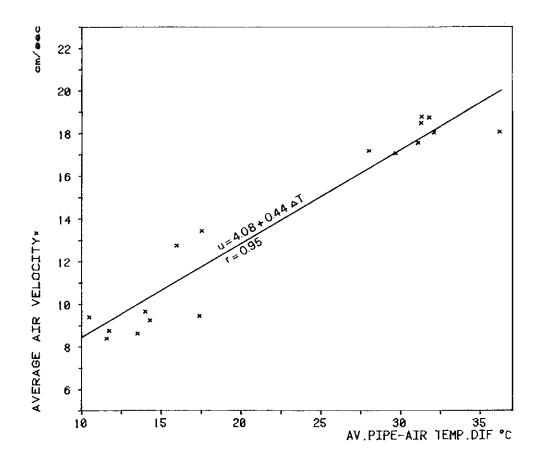


Fig. 1 - Air velocity as a function of the difference in temperature between the pipe surface and air 0.8 m above ground. Each point shows averages over a whole cooling interval.

3. RESULTS

The continuous circulation of water in the pipe system introduces a perturbation in the cooling pattern. Due to the forced circulation, it must be assumed that convection and not conduction is the main heat transfer mechanism, also inside the pipe: it has to be expected that temperature decay of the pipe surface points to a higher conductivity (and hence diffusivity) than if still water was present. On the other hand, it is acknowledged that the heat exchanger-buffer vessel system is always gaining heat due to leakages through the valve placed at the steam inlet of the heat exchanger. The valve between the heat exchanger-vessel loop and the pipe loop is also leaking, although to a lower degree; it may be concluded then that the water steadily brought to any given section of the pipe system is warmer than the water it replaces, i.e. the cooling rate (and diffusivity directly related to it) is smaller than it should be.

These considerations and the presence of many different thermal conductivities between the axis of the pipe and the surface where the temperature is measured (water, iron, coating) lead to the conclusion that diffusivity K has to be an output parameter of the best-fit procedure and not an input, i.e. an apparent diffusivity for such a system is calculated.

The calculated κ as the average of the resulting diffusivities for each cooling interval is $1.12 \cdot 10^{-7} \text{ m}^2\text{s}^{-1}$ (with a standard deviation $s = 0.314 \cdot 10^{-7}$) which is indeed smaller than the diffusivity of water $\kappa_w = 1.52 \cdot 10^{-7} \text{ m}^2\text{s}^{-1}$ at 40°C . It means that the effect of leakages from the heat exchanger outweighs the effect of convection inside the pipe section, and that of the much greater diffusivity of a part of it $(\kappa_{\text{iron}} = 1.2 \cdot 10^{-5} \text{ m}^2\text{s}^{-1})$.

Using the total heat transfer coefficient α calculated from (10) the energy H released by the pipe system was calculated as the integral of Q over pipe area and time for the two nights for which energy consumption data of the greenhouse were available. The total amount of heat released during the 8 cooling intervals of February 20-21, 1981, was $H = 2.81 \cdot 10^9$ J (the gutter system was coupled to the main network) and the time interval over which readings of the energy meter are

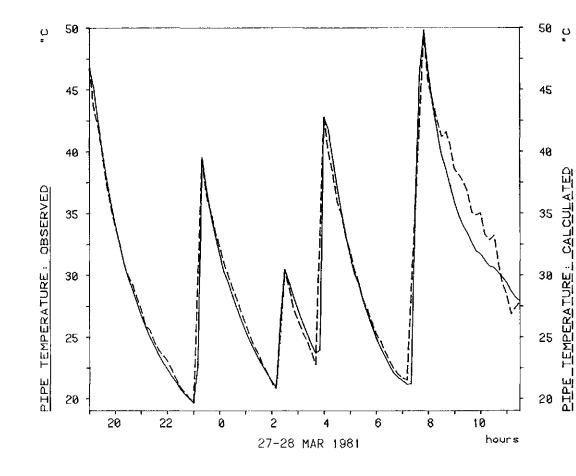


Fig. 2 - Observed and calculated pipe surface temperature (using eq. (8)). The starting temperature for each cooling is input into the procedure. Note that when the air temperature is influenced by other factors than pipe heating (after 8.30 a.m.) the predicted temperature deviates more from the measured one.

available is 33% longer (some time and one more warming-cooling after the last temperature measurement, plus all the intervals when the pipe temperature was rising); this points to a value $H = 3.72 \cdot 10^9$ J for the whole time, which is in good agreement with $H = 3.96 \cdot 10^9$ J provided by the meter. It is suggested that some energy is lost outside the heating system, in the connecting sections. On the other hand, for the measurements of 14-15 March, 1981, the time interval between readings of the meter was 10% longer than the total cooling time, and the gutter system was not connected. The total heat released in the three cooling intervals was $H = 1.01 \cdot 10^9$ J x 1.1 = 1.11 $\cdot 10^9$ J that is again a small underestimate of $H = 1.16 \cdot 10^9$ J provided by the meter.

In order to enable estimates of heat released in other circumstances, without having to go through this cumbersome procedure all over again, $\alpha_{\rm C}$ was calculated from each α by means of (5) $(\alpha_{\rm r}$ was calculated from (1) with $\epsilon=0.95)$, and the corresponding Nusselt number according to (4) was written, using the above-mentioned value for $\lambda_{\rm a}.$ It was then calculated as a function of the Prandtl and Grashof numbers, i.e. the value of the constant C of (6) was estimated. It was found that for such a system

$$Nu = 0.330 (Gr.Pr)^{1/4}$$
 (11)

with a standard deviation of C, s = 0.048.

The value of the coefficient is lower than any value found for similar systems (with still water, however) in the literature cited, and can be explained by the considerations made above.

Eq. (1), (4) and (11) were substituted in (5) to get

$$\alpha = 2.155 \cdot 10^{-7} \, \text{T}_{\text{m}}^{3} + 3.505 \left(\frac{\text{T}_{\text{s}} - \text{T}_{\text{a}}}{\text{T}_{\text{a}} \cdot d} \right)^{1/4} \, \text{Wm}^{-2} \text{K}^{-1} \quad (12)$$

and (8) with substitution of (12) was applied to data of a fourth night, to calculate the pipe temperatures. The results are shown in Fig. 2, together with measured values and allow the conclusion that (12) gives satisfactory estimates of the convective (and radiative) heat transfer coefficient for such a system.

4. CONCLUSIONS

The heat transfer coefficient thus calculated for convection averaged 65% of that for radiation. This shows that for such a system radiation is a more efficient way of losing energy than convection. From literature values, such as the ones mentioned, it is mostly concluded that in the present range of temperatures, such a system should lose about half of its energy either way, while the findings of Stoffers (1976) indicated that convection should be more efficient. It has to be stressed that the conditions of the various experiments were hardly similar, and no attempt is made here to investigate the combined effect of the many differences.

However, it may be concluded that the method outlined here allows a good estimate of the heat transfer coefficients to be made for a given system, by means of a few and simple measurements. It has to be stressed that no action had to be undertaken on the system to make such calculations possible. In this way, the energy output from a given system under its normal working condition through both convection and radiation can be easily estimated.

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