DYNAMIC CHARACTERISTICS OF COOLING AND DEHUMIDIFICATION IN PHYTOTRON SYSTEM

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DYNAMIC CHARACTERISTICS OF COOLING AND DEHUMIDIFICATION IN PHYTOTRON SYSTEM

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CHIKUSHI J. and EGUCHI H. *Dynamic characteristics of cooling and dehumidification in phytotron system.* BIOTRONICS **16**, 75–80, 1987. Dynamic characteristics of cooling and dehumidification in phytotron system were examined for obtaining information about more reliable and suitable operation of the controlled environments. Temperatures and humidities were measured at different positions in the phytotron system by using thermocouples and electric capacitance meters, respectively. Cooling characteristics were clearly found in the delay times; the delay time at air outlet of the coil was larger than that at air inlet. Capacity and characteristics of dehumidification were analyzed by the difference in absolute humidity between at both sides of the coil. Distinct difference in peak time of dehumidification was found among initial air temperatures; dehumidification effect was lower at lower initial air temperatures. It is suggested that the enthalpy change at the coil acts predominantly for dehumidification of air at higher initial air temperature, and for air cooling at lower initial air temperature.

Key words: environment control; air cooling; air dehumidification; phytotron; delay time; enthalpy; air temperature; air humidity.

INTRODUCTION

In phytotron systems, two environmental factors of air temperature and air humidity are controlled basically, and cooling coil is an important device which manipulates cooling and dehumidification of air.

There are two kinds of coil systems which are a direct expansion coil and a brine coil (*1*). In a brine coil system, the chilled fluid of $CaCl_2$ or ethyleneglycol is circulated into the coil. The brine coil system is available for high accurate control (*2*), but there are some difficulties in general use for the phytotron because of its large size, complicated system, and higher costs of running and construction.

On the other hand, direct expansion coil prevails widely for manipulations of cooling and dehumidification in phytotron system, because of its simplicity and lower cost. But the control accuracy is, in general, limited by the on-off control action. That is, the on-off action causes the cycling of controlled variables in air temperature and humidity.

The present paper deals with examination of cooling and dehumidification

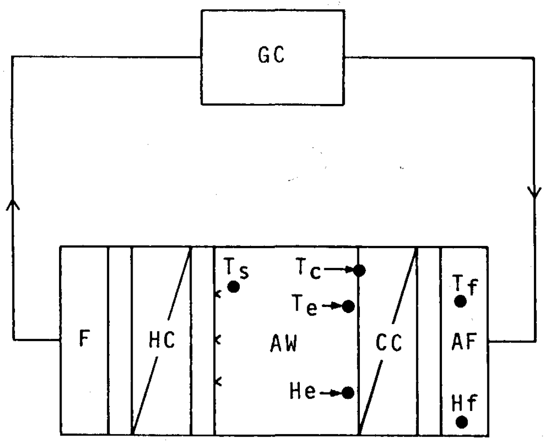

Fig. 1. Schematic diagram of the phytotron system, and measuring points of air temperatures and humidities: GC, glass room; AF, air filter; CC, cooling coil; AW, air washer; HD, heating coil. T_c, T_e, T_f and T_s are air temperatures. H_e and H_f are air humidities.

characteristics of direct expansion coil in the phytotron system under the step input of refrigerator in order to obtain information on more reliable control of air temperature and humidity.

APPARATUS AND METHODS

Direct expansion coil

The cooling coil consists of 6 (row) × 24 (column) × 120 cm (length) of copper tubes with aluminum fins (fin pitch of 5 mm). Chlorofluorocarbon R-22 ($CHClF_2$) is used as the refrigerant. Surface area of the coil for heat exchange is 94 m^2 in total. The air flow rate passing through the coil is 8,500 $m^3\,h^{-1}$, and cooling capacity is 32,000 kcal h^{-1}.

Disturbance to the phytotron system

There are mainly two kinds of disturbances in the phytotron system. One is the radiant energy (*3*) passing through the glasses. The other is heat exchange by ventilation of air flowing into and out the phytotron system.

The volume of ventilated air was measured by using CO_2 gas as a tracer. The ventilated air rate was determined from rate of decrease in concentration of CO_2 gas in time course on the basis of the following theory.

The equation of continuity of CO_2 gas in the system is

$$dc/dt = A\ (c_o - c) \qquad (1)$$

where c, CO_2 concentration inside the system; c_o, CO_2 concentration outside the system; t, time; A, ventilation count rate. Ventilation count rate A was given as ventilated air volume rate divided by air volume of the system (system volume). Assuming $c=c_i$ at $t=0$, then the integration of Eq. (1) leads to

$$At = -\ln\{(c-c_o)/c_i-c_o)\} \qquad (2)$$

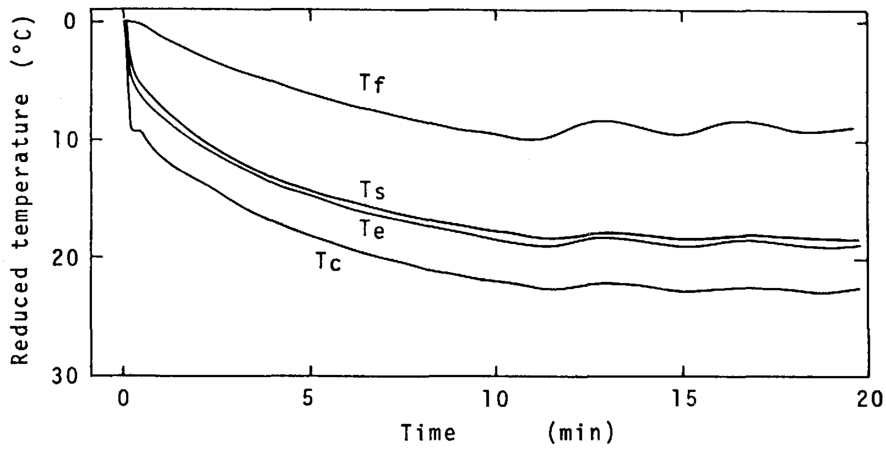

Fig. 2. Dynamic responses of air temperatures (at T_c, T_e, T_s and T_f in Fig. 1) to the step input of refrigerator.

From Eq. (2), A was determined as the drift of data of CO_2 concentration, and 1.03 $m^3\ h^{-1}\ m^{-3}$ was obtained in the phytotron system used in this experiment. This volume of ventilated air was corresponded to the system volume, i.e. 64 m^3. The disturbing action by the ventilated air was enlarged with increased differences in air temperature between inside and outside of the system, or with increased volumes of ventilated air.

Experimental procedure

Figure 1 shows the schematic diagram of the phytotron system (Phytotron II in Biotron Institute, Kyushu University; *4*) used in this experiments. The system consists of several devices such as an air filter, a cooling coil, an air washer (humidifier), a heating coil, a fan, and a glass room. These devices are connected with ducts in series. Air passes through these devices in the order mentioned above.

In this experiment, the dynamic responses of temperature and humidity were examined at the several points in the system; air temperatures were measured at the air filter T_f, surfaces of the cooling coil T_c, air outlet of the cooling coil T_e and water spray of air washer T_s, respectively. Furthermore relative humidities were also measured at the air filter H_f (front of the coil) and at the outlet of the cooling coil H_e (behind the coil). For measuring those temperatures and relative humidities, thermocouples and electric capacitance meters (HMP15, Vaisala Oy) were used; the humidity sensor was reliable for dynamic measurements as reported by Kitano *et al.* (*5*).

RESULTS AND DISCUSSION

Figure 2 shows dynamic responses of air temperature after switching on the refrigerator (step input), where the ordinate indicates the reduced temperature from the initial air temperature (T_i). The air temperatures dropped in order of T_c, T_e, T_s and T_f. That is, those temperatures became higher with distances apart from the cooling coil. The temperature difference between T_c and T_f varied from about

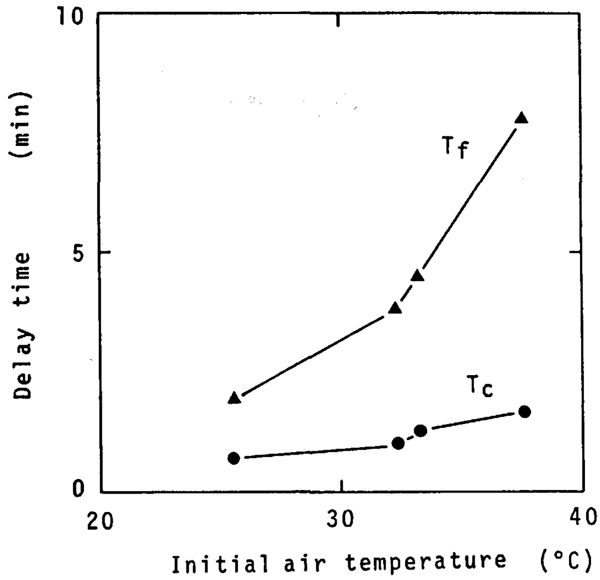

Fig. 3. Relation between initial temperature and delay times of air temperature at the air filter (T_f) and surface temperature of the cooling coil (T_c).

10 to 14°C in time course and became almost constant after the time when T_c reached to the lowest; at that time, those temperatures were more or less perturbed.

Figure 3 shows the changes in the delay time for T_f and T_c to T_i. In this phytotron system, the heat convecting process was able to be estimated as a multi-order system. So, to examine dynamic responses of the temperature, the delay time (*6*) was used for the analyses. The delay time was influenced greatly by T_i; delay time in T_f was larger than that in T_c. This fact suggests that T_c is influenced by the cooling coil more sensitively than other temperatures.

Figure 4 shows the differences in absolute humidity, air temperature and enthalpy between both sides of the cooling coil (*DH*, *DT* and *Dh* are respective differences in absolute humidity, air temperature and enthalpy). Difference in absolute humidity was obtained from relative humidities and air temperatures measured at inlet side (front of the coil) and at outlet side of the cooling coil (behind the coil). The change in energy of air passing through the cooling coil was evaluated by enthalpy change. In general, the enthalpy of air can be calculated from the following equation as defined in the field of air conditioning (*7*);

$$h = h_a + x\, h_v \qquad (\mathrm{kcal\ kg^{-1}}) \tag{3}$$

where h_a, enthalpy of dry air ($=0.24\,T$); h_v, enthalpy of water vapor ($=597.5+0.44T$); x, mixing ratio of air; T, dry-bulb temperature (°C). At respective initial air temperatures (T_i) of 37.6, 32.3 and 25.5°C, each peak of *DH* appeared at different times. Elevation of the peak at $T_i=25.5$°C was lowest. Time of the peak shifted to be later with higher T_i; times of the peaks were about 50, 8 and 1 min at respective T_i of 37.6, 32.3 and 25.5°C. This peak of *DH* was estimated to be caused by formation of dew and/or frost on the surface of the cooling coil. The frost made it

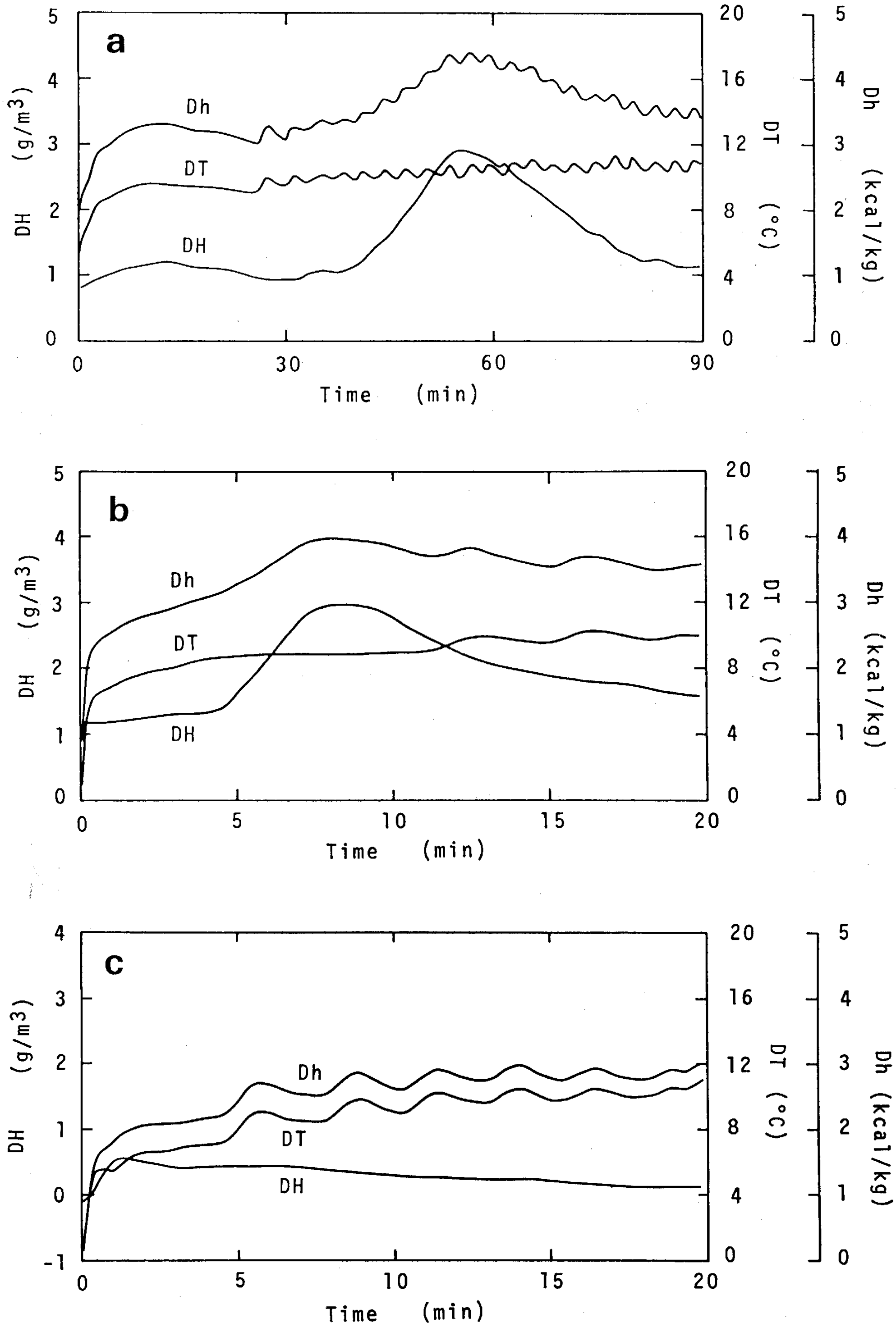

Fig. 4. Differences in absolute air humidity, air temperature, and enthalpy (*DH*, *DT* and *Dh*) between inlet and outlet of the cooling coil at respective initial air temperatures of 37.6°C (a), 32.3°C (b) and 25.5°C (c).

impossible to drop the air temperature lower than 0°C, and made it difficult to trap the water vapor enough. *DH* and *Dh* peaks appeared at the same time at respective T_i of 32.3 and 37.6°C. At T_i of 25.5°C, the peak of *DH* appeared just after the start of cooling, and clear peaks were not found in *Dh* and *DT*. The elevations of *DH* at respective T_i of 32.3 and 37.6°C were higher than that at T_i of 25.5°C. Variations of *DH* and *Dh* were almost parallel even at different T_i of 32.3 and 37.6°C. Furthermore, at T_i of 25.5°C, *DT* changed according to *Dh* variation. These facts indicate that the energy of *Dh* at the coil acts predominantly for dehumidification of air at higher T_i, and for air cooling at lower T_i.

These results could be applied to optimum setting of control parameters (*8*) in phytotron system.

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