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DIRECT DIGITAL CONTROL OF AIR HUMIDITY FOR PLANT RESEARCH

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EGUCHI H., KITANO M. and MATSUI T. *Direct digital control of air humidity for plant research.* BIOTRONICS **13**, 29–38, 1984. Direct digital control (DDC) system was employed in a growth chamber for accurate control of air humidity in wider regions. Control elements were composed of cooling brine coil, dehumidifying brine coil, reheating coil (electric heater), heating coil (electric heater) and steam boiler: Volumes of circulated brines, electric currents in heaters and steam volume were manipulated by PID control action (velocity algorithm). Furthermore, capacities of cooling and dehumidifying coils were optimalized for high accuracy by the feed-forward control of brine temperatures, where the optimum temperatures of respective brines were set on the basis of the desired values of air temperature and humidity. This system made it possible to control relative humidity from 10 or 20 to 85% within ±3%RH in an air temperature control region of 0 to 45 ±0.3°C (the lowest humidity was 10%RH at air temperatures higher than 10°C, but was 20%RH at 0°C): This control region covered saturation deficits of 0.73 to 58.77 g m^{-3}.

INTRODUCTION

Plants grow under various air conditions, and air temperature and humidity are basal factors in environment control for plant research. Air temperature control has been mostly generalized, and many kinds of systems have been applied to various environmental studies of plants (*1*, *2*). On the other hand, it is necessary to attach more importance to humidity control for developing more reasonable system of environment control, as humidity is responsible for physiological functions and plant growth (*3–5*, *7*, *8*, *14*, *15*). Present paper deals with computer control of air humidity with high accuracy in wider regions.

CONTROL SYSTEM

System diagram is illustrated in Fig. 1. Control elements consisted of cooling coil, dehumidifying coil, reheating coil, heating coil, and humidifying steam boiler. A computer system was used for direct digital control (DDC) (*6*, *13*), which was composed of central processing unit (CPU; TOSDIC-245, TOSHIBA CORPORATION), interfaces and CRT console. Manipulated variable was generated by PID control action (*10*, *11*): Theoretical PID algorithm of the position form is

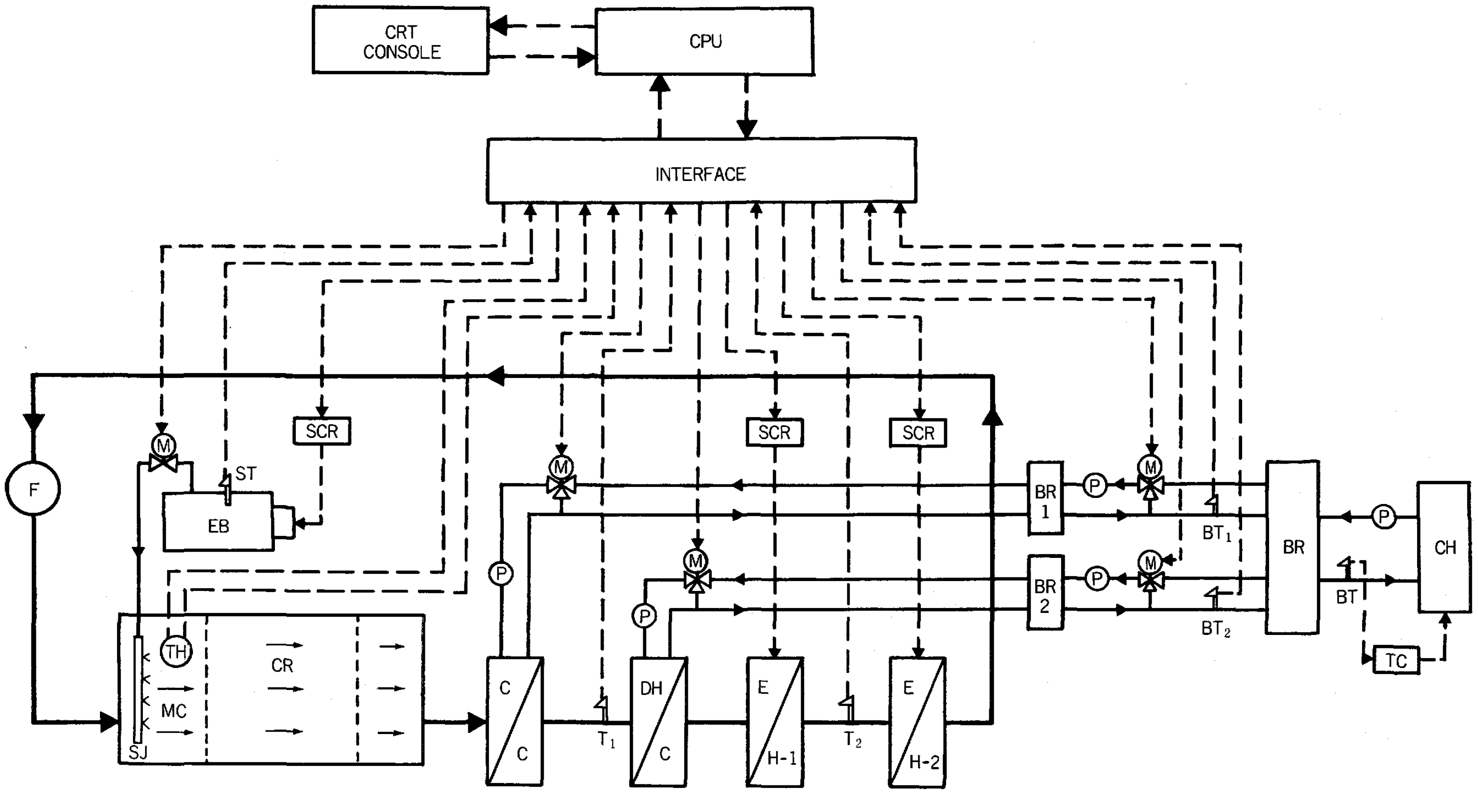

Fig. 1. System diagram of air temperature and humidity control: CR, controlled environment room; MC, mixing chamber; C/C, cooling coil; DH/C, dehumidifying coil; E/H-1, reheating coil; E/H-2, heating coil; SJ, steam jet; F, fan; CH, chiller unit; BR, BR1 and BR2, brine tanks; EB, electric steam boiler; M, motor valve; P, pump; TH, air temperature and humidity sensors; T_1, T_2, BT, BT_1, BT_2 and ST, temperature sensors; TC, temperature controller; SCR, silicon controlled rectifier.

$$Y_t = K_P\left(Z_t + \frac{1}{T_I}\int Z_t dt + T_D\frac{dZ_t}{dt}\right) \tag{1}$$

where: Y_t, manipulated variable; K_P, proportional gain; T_I, reset time; T_D, rate time; Z_t, controlled deviation ($Z_t = sv - pv$; sv, desired value; pv, controlled variable). Eq. (1) can be rewritten as following numerical equation,

$$Y_t = K_P\left(Z_t + \frac{\Delta t}{T_I}\sum_{i=0}^{n} Z_{t-i\Delta t} + T_D\frac{Z_t - Z_{t-\Delta t}}{\Delta t}\right) \tag{2}$$

where Δt is sampling interval.

In this system, the velocity algorithm of PID control action was employed and manipulated variable was given by

$$\begin{aligned} Y_t &= Y_{t-\Delta t} + \Delta Y_t \\ &= Y_{t-\Delta t} + K_P\left\{(Z_t - Z_{t-\Delta t}) + \frac{\Delta t}{T_I}Z_t + \frac{T_D}{\Delta t}(Z_t - 2Z_{t-\Delta t} + Z_{t-2\Delta t})\right\} \end{aligned} \tag{3}$$

where $K_P > 0$ in heating, humidifying and reheating, and $K_P < 0$ in cooling and dehumidifying.

For rapid compensation of overshooting, manipulation in Eq. (3) was treated as $Y_t = 0$ when $Z_t < Z_{t-\Delta t} < 0$ in heating, humidifying and reheating, and when $0 < Z_{t-\Delta t} < Z_t$ in cooling and dehumidifying.

List of K_P, T_I and T_D were filed in CPU, and these parameters were selected for optimum settings (*11*, *12*) corresponding to desired values of air temperature and humidity. The desired values and the other system parameters were set through CRT console. Air temperature (pv_A) and humidity (pv_H) were detected in the mixing chamber by sensors (TH) of Pt100Ω and electric capacitance meter (HMP 15, Vaisala, Oy) (*9*), respectively. The controlled variables and other signals detected by T_1, T_2, BT_1, BT_2 and ST were transmitted to CPU and used for computation and monitoring of the system in CRT displays. Figure 2 shows an example of the CRT display. Y_t calculated in Eq. (3) was used for manipulation of the final control element; respective capacities of cooling, dehumidifying, reheating, heating and humidifying were manipulated by each Y_t at an interval of Δt.

MANIPULATION

The growth chamber was installed in the room where air temperature and relative humidity were controlled at about 23°C and about 60%. A volume of the controlled environment room of the growth chamber was 1.5 m (width) × 1.3 m (depth) × 1.5 m (height), in which controlled air flowed out of left side to right side with a velocity of about 0.3 m sec^{-1} (circulated air volume, Q was 2000 m^3 hr^{-1}); 5.5% of Q was the fresh air introduced from the room (23°C and 60%RH) into the growth chamber. Control regions of air temperature and humidity are shown in Fig. 3. Air temperature was controlled within a region of −10 to 45°C, and humidity control region distributed from 10 or 20 to 85% relative humidity (RH) in air temperature region of 0 to 45°C, which covered saturation deficit of 0.73 to 58.77 g m^{-3}. For air control in such wider regions, the control elements were provided with large capacities. In general, there is the delay in the feedback

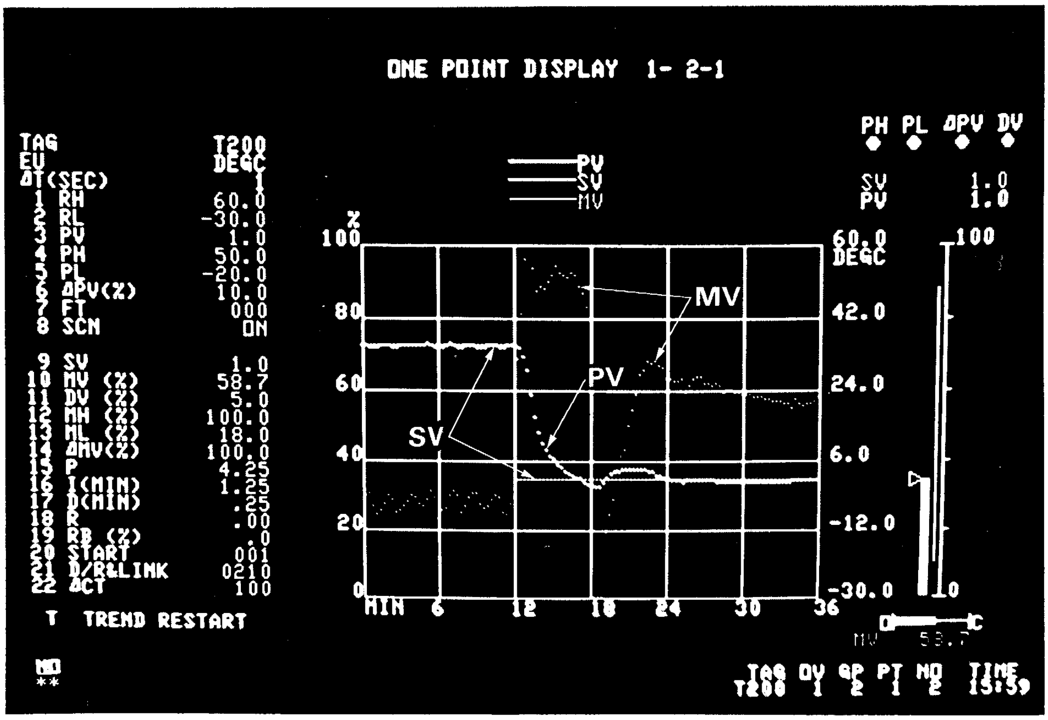

Fig. 2. CRT display of dew point step control from 35.5 to 1.0°C: PV, controlled variable (°C); SV, desired value (°C); MV, manipulated variable of dehumidifying (manipulating percentage of motor valve for brine circulation).

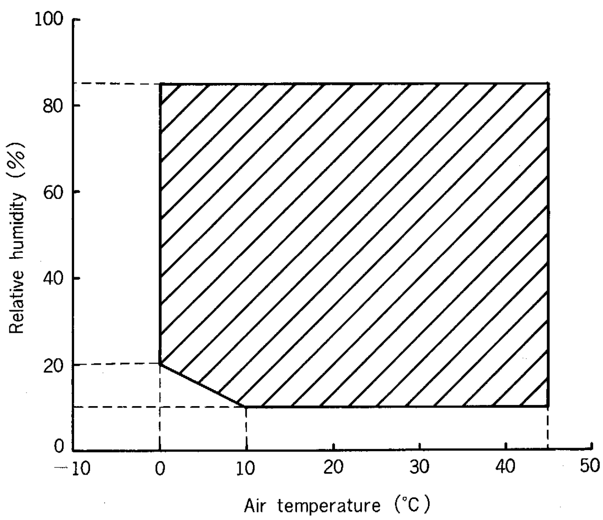

Fig. 3. Control regions of air temperature and humidity.

control system, and large capacities of the control elements result in decrease in control accuracy as affected by the overshoot of manipulation, which is caused by the delay. So, each element in this system was accommodated to the optimalizing control for high accuracy. On the other hand, in the case of air control at lower temperatures or lower dew points, frost forms on the surface of the cooling coil or the dehumidifying coil and results in decrease in the capacities. For such a case, two sets of the coils were equipped in two air passes; when one of the coils was frosted, the other was operated, and the air pass was exchanged into another by driving a damper with an interval of defrosting.

Preparation of cool brine

A chiller unit (CH) was used for cooling the source brine (aqueous solution of ethylene glycol) in a brine tank (BR); the lowest temperature of the brine was -25°C. The source brine was divided into two tanks of BR_1 for cooling and BR_2 for dehumidifying. Respective brine temperatures in BR_1 and BR_2 were controlled at the desired values for optimalizing capacities of the coil, which were calculated in feed-forward control system on the basis of desired values of air temperature and humidity; the brine temperatures were detected by BT_1 and BT_2, and volumes of brine circulated from BR to BR_1 and to BR_2 were manipulated through motor valves by respective computer signals. Thus, cool brines were prepared at respective optimum temperatures for cooling and dehumidifying.

Cooling

The brine coil was equipped for cooling, which consisted of 12 (row)$\times$9 (column)$\times$600 mm (length) of copper tubes with aluminium fins (fin pitch of 5 mm). The cooling capacity varies with brine temperature and volume of brine circulated into the coil. For optimalizing the cooling capacity, the brine temperature was set 10°C lower than the desired value of air temperature in the feed-forward control system, and volume of the circulated brine was manipulated through a motor valve by PID control action of Eq. (3). This cooling coil was provided with the maximum capacity of 10000 kcal hr^{-1} for sensible heat loads brought by introduced fresh air, heat conduction through the wall, artificial light radiation and humidification stated below. The maximum capacity was obtained in the case of brine temperature of -25°C and the circulated brine volume of 4080 l hr^{-1} (maximum volume).

Dehumidifying

Dehumidifying load reached a peak at air temperature of 10°C and relative humidity of 10%. At this point of the air condition, specific volume of air (v) is 0.803 m^3 kg^{-1}, and humidity ratio (x_0) is 0.00076 kg kg^{-1}. Water vapour (L) gained in controlled environment room was estimated by summation of L_f brought by introduced fresh air and L_P caused by evapo-transpiration in plants in controlled environment room. L_f was given by

$$L_f = \frac{Q_f}{v'}(x_f - x_0) \doteqdot 1.3 \text{ kg hr}^{-1} \tag{4}$$

where Q_f (volume of introduced fresh air) is 110 m^3 hr^{-1} (5.5% of Q), v' (specific volume of fresh air of 23°C and 60%RH) is 0.854 m^3 kg^{-1}, and x_f (humidity ratio of fresh air of 23°C and 60%RH) is 0.0106 kg kg^{-1}.

L_P was evaluated as 0.3 kg hr^{-1} which corresponded to enough volume of evapo-transpiration in 10 young cucumber plants (*8*). Thus, L was 1.6 kg hr^{-1}. Therefore, humidity ratio (x_1) of entering air (air entering to the dehumidifying coil) was given as

$$x_1 = x_0 + \frac{L}{Q} \cdot v = 0.00140 \text{ kg kg}^{-1} \tag{5}$$

So, enthalpy (i_1) of the entering air was obtained as $i_1 = 3.2$ kcal kg^{-1} from psychrometric chart.

For removing L, it is necessary to reduce the temperature of the leaving air (air coming out of the dehumidifying coil) to the dew point for x_0 by dropping the temperature of the coil surface to the apparatus dew point. At this point, enthalpy (i_2) of -3.9 kcal kg^{-1} was obtained from psychrometric chart. Thus, peak heat load (H_L) of dehumidifying coil was given as

$$H_L = \frac{Q}{v}(i_1 - i_2) \fallingdotseq 18000 \text{ kcal hr}^{-1} \tag{6}$$

For dehumidifying, the brine coil was used, which consisted of 14 (row)$\times$9 (column)$\times$600 mm (length) of copper tubes with aluminium fins (fin pitch of 5 mm). Difference (ΔT_B) in brine temperature between entrance and exit in this coil was 4.5°C at its maximum in the case of brine temperature of -25°C (at the entrance) and circulated brine volume (Q_B) of 5220 l hr^{-1} (maximum volume). Therefore, the maximum heat (H) transferred to the brine in this coil was given as

$$H = Q_B \cdot \rho \cdot c \cdot \Delta T_B \fallingdotseq 18000 \text{ kcal hr}^{-1} \tag{7}$$

where ρ (density of brine) is 1.10 kg l^{-1}, and c (specific heat of brine) is 0.70 kcal kg^{-1} °C^{-1}.

Thus, from Eq. (7), it could be estimated that the capacity of this coil can satisfy H_L. For optimalizing humidifying coil, the brine temperature was set about 5°C lower than the dew point at desired air temperature and humidity in the feed-forward control system; the dew point was calculated from the desired values of air temperature and humidity. Volume of the circulated brine was manipulated through a motor valve by PID control action of Eq. (3).

Reheating

In order to compensate air temperature drop caused by the dehumidifying coil, the reheating coil was equipped; the electric heaters were sheathed in stainless steel tubes with stainless steel fins (fin pitch of 4 mm), consisting of 3 (row)$\times$6 (column)$\times$ 750 mm (length). This coil was provided with the capacity corresponding to the heat load of Eq. (6), and electric current was manipulated through silicon controlled rectifier (SCR); the manipulated variable of this reheating coil was calculated from Eq. (3) where air temperature detected by T_1 was used as the desired value, and air temperature detected by T_2 was used as controlled variable (feedback signal); difference in temperature between air entering to the dehumidifying coil and air coming out of reheating coil was vanished by manipulating the reheating capacity.

Thus, this reheating coil compensated the disturbance caused by dehumidifying for air temperature control.

Heating

Heating coil consisted of 1 (row) × 12 (column) × 750 mm (length) of stainless steel tubes with stainless steel fins (fin pitch of 4 mm), in which electric heaters were sheathed. Electric current of this coil was manipulated through SCR by PID control action of Eq. (3). This heater was provided with the maximum capacity of 8600 kcal hr^{-1}.

Humidifying

Steam was used for humidifying which was generated by an electric steam boiler (*10*). Humidifying load reached a peak at desired values of 45°C and 85% RH in the case of introduced fresh air (23°C and 60%RH) of 110 m^3 hr^{-1} (Q_f=5.5% of Q). For this peak humidifying load, water vapour (L') required was given by

$$L'=\frac{Q_f}{v'}(x_2-x_f)=5.85 \text{ kg hr}^{-1} \tag{8}$$

where x_2 (humidity ratio of air of 45°C and 85%RH) is 0.056 kg kg^{-1}, and v' and x_f are defined in Eq. (4).

In practical process, too much water vapour is removed by excessive dehumidification which is accompanied by cooling in the cooling coil and caused by overshoot of manipulation and residual cooling in the dehumidifying coil. For these extra loads, the safety factor of 2.0 was applied to this system, and the boiler was provided with maximum capacity (water vapour) of 12 kg hr^{-1}. Steam temperature in the boiler was set within a region of 105 to 120°C, and steam was supplied from jet nozzles in the mixing chamber; volume of steam supply was manipulated through a motor valve by PID control action of Eq. (3).

PERFORMANCE

Figure 4 shows controlled variables of air temperature and relative humidity at the lowest and the highest saturation deficits. The lowest saturation deficits (A) of 0.73 g m^{-3} was obtained in the control region at air temperature of 0°C and relative humidity of 85%. Controlled air temperature and relative humidity fluctuated to some extent, but were maintained within ±0.3°C and ±3%RH. At the highest saturation deficit (B) of 58.77 g m^{-3} (45°C and 10%RH), reliable characteristics were found in the controlled variables of air temperature and relative humidity.

Figures 5 and 6 show humidity step responses at air temperatures of 10°C and 40°C, respectively. In the regions at lower and higher humidities, small overshoot and delay were found, but the controlled variables settled within 10 min. The air temperature control was not disturbed by the humidity changes, and the accuracy was kept within ±0.3°C.

These static and dynamic characteristics indicate that this DDC system makes it possible to control air temperature and humidity in wider regions with high

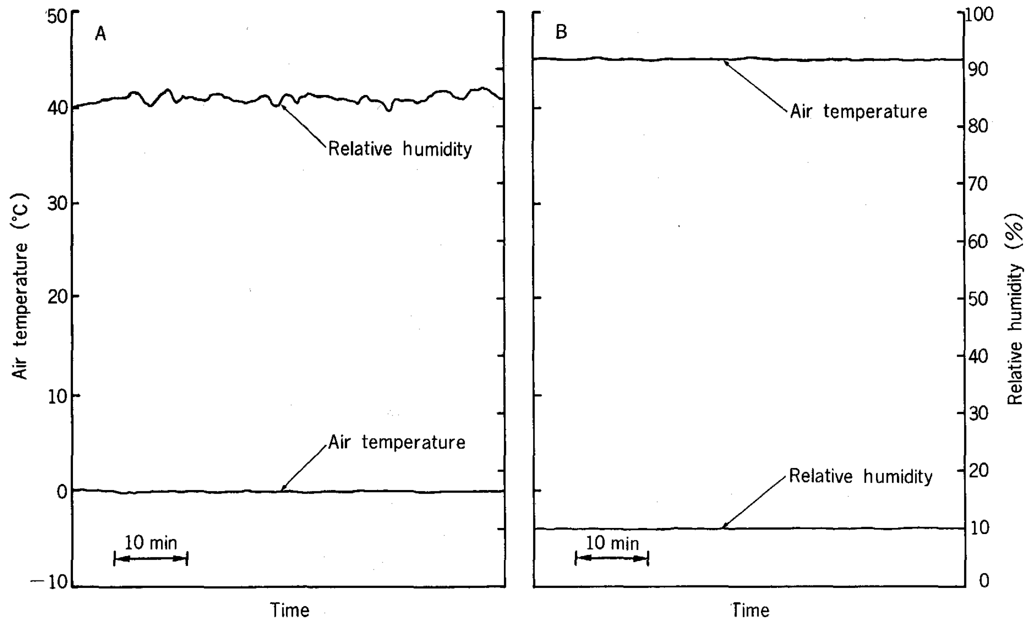

Fig. 4. Static characteristics of air temperature and humidity controls at the lowest saturation deficit (A) of 0.73 g m^{-3} (0°C and 85%RH), and at the highest saturation deficit (B) of 58.77 g m^{-3} (45°C and 10%RH).

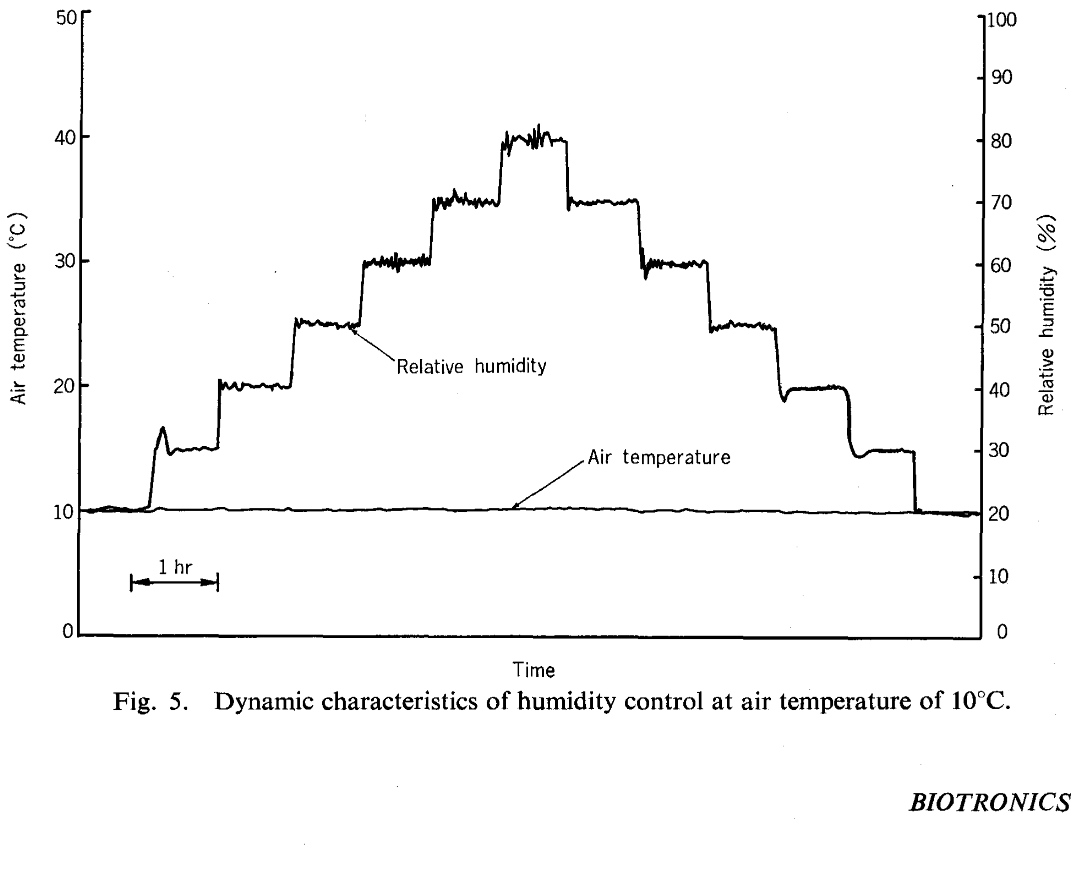

Fig. 5. Dynamic characteristics of humidity control at air temperature of 10°C.

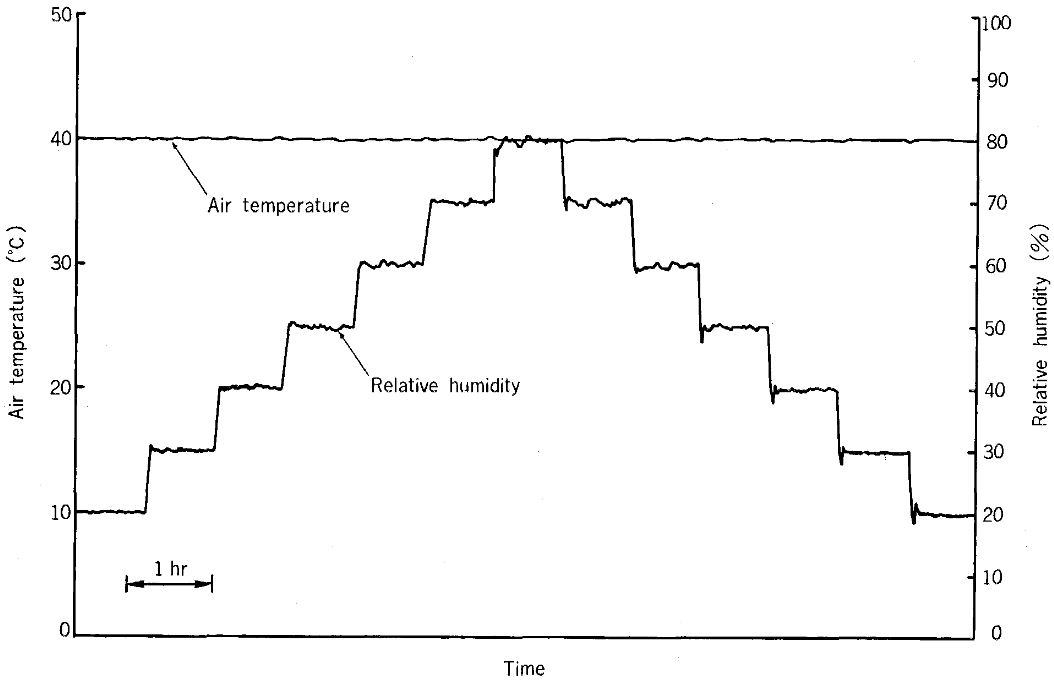

Fig. 6. Dynamic characteristics of humidity control at air temperature of 40°C.

accuracy and can help to analyse various plant responses to air humidity at different air temperatures.

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