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A PROTOTYPE HEAT EXCHANGER FOR HUMIDITY CONTROL IN GREENHOUSES

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SUMMARY:
Humidity control is becoming increasingly important as infiltration in greenhouses is eliminated and ventilation is reduced because of the increased cost of energy. This paper describes an air-to-air heat exchanger designed so that exiting air is used to preheat incoming ventilation air at a rate sufficient to control humidity levels in the greenhouse.

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Introduction

Greenhouses are structures designed to control the environment for plant growth. In most cases major consideration is given to the control of air temperature by heating or venting and control of light. In some cases independent control of soil temperature, carbon dioxide concentration and relative humidity may be sought. If an increase in humidity is required, this is usually provided by misting. The most common method of reducing humidity is by venting and if temperature is reduced below acceptable levels, heat must then be added.

It has been suggested by Walker and Cotter (1968) that the influences of humidity on plant response have not been as thoroughly investigated as those of light, temperature, and carbon dioxide levels. This is, in part, due to the difficulty in providing an accurate means for its measurement and control in a greenhouse. They state that for a glass-covered greenhouse internal humidity is regulated by condensation on the structure when it is cold outside. Cotter and Seay (1961) reported that in early British designed plastic-covered greenhouses there were higher humidity levels than in glass because there is less air infiltration. In double-covered plastic houses, due to the insulation of the entrapped air space, internal surfaces are warmer than in single-glazed houses so there is less dehumidification due to condensation.

In this era of rising fuel costs, there is a great incentive to use a movable curtain insulation system to reduce nighttime heat loss. Bailey (1978) and Smith (1978) report increases in relative humidity in glass houses using these systems. Manning (1980) reported the relative humidity in a double-covered plastic greenhouse using a night curtain to insulate a tomato crop produced a saturated environment between 10:00 p.m. and 6:00 a.m. in a fall crop.

Control of humidity at night by venting and then heating would defeat the purpose of energy conservation measures. However, by venting through a heat exchanger in which warm, moist, exiting

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greenhouse air gives up heat to incoming dry, cold, outside air, humidity could be reduced with less requirement for additional heating. The two air streams would need to be separated by vapor impermeable heat transfer surfaces of sufficient area to have a high heat exchange effectiveness. This paper describes a preliminary study conducted to determine the engineering parameters of such a heat exchanger.

Design and Construction

Greenhouse-grade polyethylene film was selected as the heat transfer surface material because large surfaces can be constructed at low cost. A series of design calculations were conducted to determine a reasonable heat exchanger design for the first test unit. In Japan, Mihara and Hayashi (1978) have developed a crossflow heat exchanger for greenhouse dehumidification with an ingenious fabrication procedure. In Canada, Besant, et al., (1979) have worked with counterflow units constructed of metal heat exchange surfaces for residences in which air flows are vertical to take advantage of natural convection. In our case a counterflow configuration was selected as the potential heat exchange effectiveness is greater than for cross flow or parallel flow units. Horizontal designs were mandated as the airflow requirements anticipated for greenhouses dictated flow lengths so long that vertical units would be impractical.

The research greenhouse in which the tests of the heat exchanger were to be conducted is 11 x 14.6 m with a gutter height of 2.4 m and an enclosed volume of 501 m^3. Mihara and Hayashi (1978) recommended an air exchange rate of 0.3 m^3/min per m^2 of floor area which would indicate a design flow rate for the prototype of 48 m^3/min. Considering the anticipated improvement in heat exchanger effectiveness of a counterflow unit and the fact that the test greenhouse could be segregated into 2 or 3 compartments, it was decided to design the test unit for an airflow of 12.7 m^3/min. In order to maintain turbulent airflow, a design Reynolds number of 10,000 was selected.

The length of the unit was 12.2 m which was the greatest practical length for the test greenhouse. The height of the unit was to be 1.0 m, which was a convenient size to construct under the given conditions. Figure 1 shows the unit in position in the greenhouse and the locations of fans, inlets and outlets for both air streams. In order to design the widths of the alternating flow channels and the number of them in each direction, a number of factors were considered. For turbulent flow in rectangular passages, the heat transfer coefficient increases as a function of the velocity raised to the power 0.8 (Notter and Sleicher, 1972). The frictional head loss varies with the length of the duct, the square of the flow velocity and inversely with the equivalent diameter which is twice the width of a narrow rectangular duct. For constant Reynolds and Prandtl numbers the heat transfer coefficient for any given mass flow of air will vary inversely with the equivalent diameter of the channel.

A detailed discussion of these factors is presented by LePoidevin (1980). The characteristics of the design selected to provide the required airflow at a low frictional heat loss were:

Reynolds number	10,000
Prandtl number	0.703
Nusselt number	31.5
Channel width	1.6 cm
Flow per channel each way	4.2 m^3/min
Number of channels each way	3
Theoretical heat transfer coefficient	25.6 W/m K
Frictional head loss in channels	0.8 mm of water
Total frictional head loss including estimated entrance and exit head loss	7.6 mm of water
Theoretical effectiveness	0.86

The testing of this prototype and the determination of actual effectiveness will verify the applicability of the theory to a real unit and give indications as to how changes in design parameters such as length, channel width, etc., would influence performance.

The construction of the main body of the unit consisted of sheets of polyethylene film spaced by 1.6 cm strips of plywood ripped from standard sheets. Figure 2 shows a cross section of the unit with 3 active flow channels in each direction and a static, inflated channel on each side. It was found that if the outer channels were used to move air, the central portions of the inner channels would collapse and limit flow to the outside channels. The entrances at each end were fabricated of flat sheets of fiberglass taped to the polyethylene sheets of the main body of the exchanger. Curved sheet-metal flutes were placed in each fiberglass passageway to guide the air. Efforts were made to ensure that there would be reasonably equal flow in each channel and that each would maintain uniform shape. LePoidevin (1980) discusses the measures taken to accomplish these objectives in detail.

Results and Discussions

In determining an energy balance for the operating unit it was necessary to account for energy added by the fans as the fan motors were partially in the airstream. In careful testing of the fan units it was found that heat input varied with line voltage, air-flow rate and air temperature. For the conditions of the recorded tests, energy input to the airstream which was flowing at 0.2 m^3/sec was 523 W, 62% of the measured energy input to the fan.

Tests were conducted with the sides of the unit insulated and uninsulated. Although a practical unit may not utilize side insulation, it was found that this facilitated making energy measurements and arriving at a satisfactory energy balance. This was needed for accurate determination of heat transfer coefficients and heat exchanger effectiveness. Figure 3 shows the axial temperature distribution in adjacent flow channels with the side walls insulated. The fans each increase the temperature of the respective air streams 1.7°C. Heat contribution from the fans must be taken into account in the determination of heat exchanger effectiveness. The effectiveness, E, was calculated by the formula:

$$E = \frac{DB_{oi} - DB_o + 1.7}{DB_i - DB_o + 3.4}$$

Definitions of these symbols and typical temperature data and values of E are presented in Table 1.

For the data presented in Table 1, the air flow rate of 0.212 m^3/sec and the enthalpies of the entering and leaving air were used to calculate the heat transfer coefficient. This is the energy transferred divided by the heat exchange area (42 m^2) and the log mean temperature difference. The temperatures used are taken between the fans and therefore account for the heat input of the fans. The heat transfer coefficients determined between 1:00 a.m. and 5:00 a.m. for the test shown in Table 1 are 22.7, 20.5, 20.1, 18.4 and 18.7 W/m^2K, respectively, for a mean value of 20.1 W/m^2K.

This heat transfer coefficient is substantially less than the predicted value of 25.6 W/m^2K for the flow rate tested. Estimates of experimental error on these tests are about 17% with the greatest uncertainty in the determination of actual airflow. Also, calculations and observations indicate that if spacings in airflow channels are not uniform or if one channel expands and another collapses, then heat transfer will be reduced as more air moves down the larger channel which therefore lowers the total heat transfer coefficient.

The effectiveness of the unit in terms of reducing greenhouse humidity can be shown in Table 1 where the temperatures, enthalpies and relative humidities of each airstream are tabulated for 5 hours. Those data are for a unit with the sidewalls insulated. In order to determine the potential for dehumidification in a more realistic configuration without sidewall insulation, inlet and outlet humidities are plotted in Fig. 4. In separate tests it was found that when the unit is started up after a long period of being off, the greenhouse humidity level is reduced to a new equilibrium condition within about 15 minutes.

Potential for Commercial Application

The basic engineering parameters of an air-to-air counterflow heat exchanger constructed of large sheets of polyethylene film have been determined. The mechanical and thermal performance of the unit can be improved by better design. It is important to balance airflow in each channel of the heat exchanger so the flexible polyethylene film will maintain constant and uniform channel dimensions. This will improve mechanical performance and enable the heat transfer coefficient to more nearly approach the theoretical value.

The actual sizing of a commercial unit would be determined primarily by the rate at which moisture must be removed to maintain the humidity level required in the greenhouse. There seems to be very little data available on rates of evaporation in a greenhouse at night, although there is data on a 24 hr/day basis. In order to make a rough estimate on the size of a unit required for a large greenhouse, the following factors have been assumed (Lake 1966):

Total evapotranspiration 1.5 mm of water per day

One third at night or 0.5 mm of water per night (Vanderpost 1981)

Desired inside condition 15°C and 80% relative humidity

Greenhouse is 3 m high

Under these conditions, an air exchange would remove approximately 22 g of water per m^2. In order to remove 500 g of water per m^2 per night about 23 air exchanges per night or roughly 1.5 per hour would be required. The test unit had 3 flow channels in each direction with the unit being about 1 m by 12 m and its airflow was 763 m^3 per hour or enough to provide 1.5 air exchanges per hour for 170 m^2 of 3-m-high greenhouse or about 1/60 of a hectare. Note that the test greenhouse area of 160 m^2 is in very close agreement with the actual measured performance capability under these typical operating conditions.

Clearly a great deal remains to be done before a commercially acceptable unit can be designed. Better techniques of construction, especially at the transitions from the fans to the unit are needed. Head loss in the transition sections must be kept low to avoid high fan horsepower requirements. Uniform airflow down each channel must be assured to achieve heat exchange effectiveness at least as good and preferably somewhat better than that of the first prototype. Most important is the operation of a second stage prototype in a real greenhouse with plants being grown under normal commercial practice. Actual performance can then be tracked for varying cropping and weather conditions.

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Table 1

ENERGY BALANCE CALCULATION WITH SIDE-WALLS INSULATED (5:17:80)

AIRSTREAM EXITING FROM GREENHOUSE

TIME	DB_i	WB_i	u_i	h_i	h_{afi}	DB_{io}	WB_{io}	u_{io}	h_{io}	Δh_{ex}	E
1.00	22.8	20.6	0.81	76.98	79.26	19.4	18.9	0.95	71.77	7.49	0.90
2.00	22.2	20.0	0.82	75.59	77.87	18.9	18.3	0.94	69.66	8.21	0.85
3.00	21.7	19.4	0.81	73.40	75.68	18.3	17.8	0.94	67.93	7.75	0.85
4.00	21.1	18.9	0.82	72.07	74.33	17.2	17.2	1.00	66.47	7.86	0.81
5.00	20.6	18.3	0.81	69.98	72.26	16.7	16.7	1.00	64.79	7.47	0.81

AIRSTREAM ENTERING GREENHOUSE

TIME	DB_o	WB_o	u_o	h_o	h_{afo}	DB_{oi}	WB_{oi}	u_{oi}	h_{oi}	Δh_{en}
1.00	14.4	11.1	0.67	49.86	52.14	23.3	15.0	0.40	59.86	7.72
2.00	13.9	10.6	0.66	48.41	50.69	22.2	14.4	0.43	58.77	8.07
3.00	13.9	10.0	0.60	46.93	49.21	21.7	13.9	0.40	56.32	7.12
4.00	12.8	10.0	0.72	47.51	49.79	20.6	13.9	0.48	57.19	7.38
5.00	12.2	9.4	0.74	45.60	47.88	20.0	13.3	0.47	55.65	7.77

SYMBOLS

DB = Dry-bulb temperature, ^0C
WB = Wet-bulb temperature, ^0C
u = Relative humidity, %
h = Enthalpy, kJ/kg dry air
Δh = Change in enthalpy, kJ/kg dry air
E = Heat exchanger effectiveness

SUBSCRIPTS

i = inside greenhouse
afi = after fan, inside
io = exit condition of greenhouse air
o = outside greenhouse
afo = after fan, outside
oi = exit condition of outside air
ex = exiting airstream
en = entering airstream

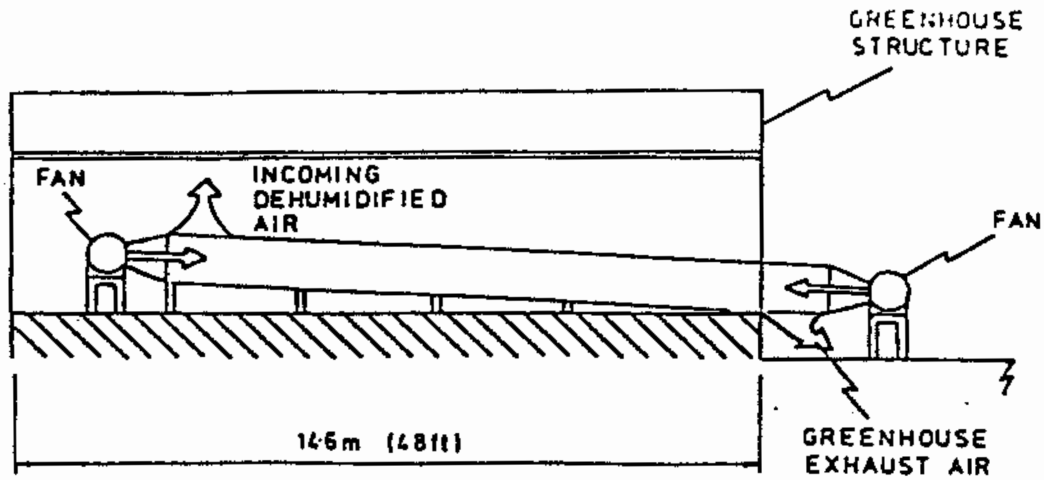

FIG 1
OVERALL LOCATION OF HEAT EXCHANGER IN TEST GREENHOUSE

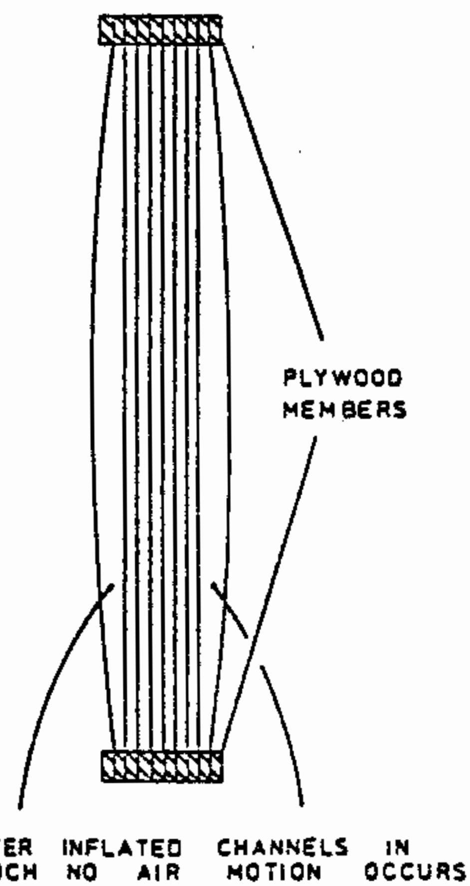

FIG 2
CROSS-SECTION THROUGH HEAT EXCHANGER WITH OUTER NON FLOW CHANNELS ADDED

FIG 3

TEMPERATURE DISTRIBUTION IN ADJACENT CHANNELS

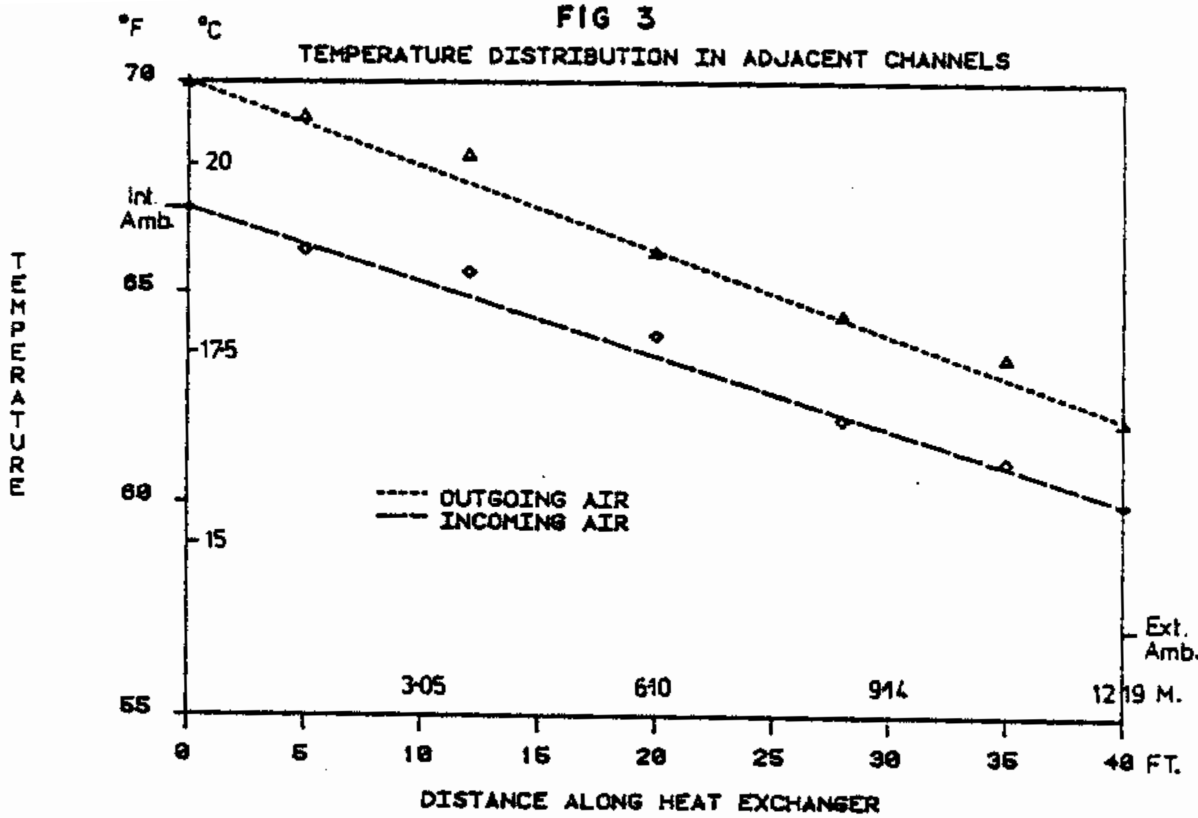

FIG 4

RELATIVE HUMIDITIES OF AIRSTREAMS WITH THE SIDEWALLS OF THE HEAT EXCHANGER UNINSULATED (5:15:80)