NEW DEHUMIDIFYING SYSTEM FOR GREENHOUSES

J.B. Campen
IMAG-DLO,
P.O. Box 43 6700 AA Wageningen
The Netherlands
E-mail: j.b.campen@imag.wag-ur.nl

G.P.A. Bot
Agricultural University
Department of Agricultural
Environmental and System Engineering
Bomenweg 4, 6703 HD Wageningen
The Netherlands

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Abstract

A new concept for a dehumidifier has been developed which reduces the need for ventilation and saves energy and CO_2. The objective of the system is a low energy demand. This is achieved by natural convection driven air circulation and integration of sensible heat recovery. Moreover a heat pump is applied for the transformation of latent heat into sensible heat. The system is positioned below the crop not causing any light reduction and distributed throughout the greenhouse. With a computational fluid dynamics (CFD) program the optimal geometry and dimensions of the components in the system are determined. A separate model for condensation has been created because this is not included in the CFD-program. A prototype based on the best design is built and tested. The calculations and experiments are in good agreement. The system has a capacity to remove 50 to 70 ml of condensate per hour per metre length of system during humid greenhouse conditions (>80% relative humidity). Predictions show that 4 to 7 % energy can be saved in a conventional greenhouse. For modern well-insulated greenhouses these savings are much higher.

1. Introduction

Humidity is a key factor in greenhouse climate. It tends to be high due to crop evaporation. In conventional greenhouses with a single thermal cover the humidity level is limited by condensation on the relatively cold cover and by ventilation through air leaks. Modern greenhouses are better insulated and more airtight thereby saving energy. However this induces higher humidity levels. Crops exposed to high humidity levels have a higher risk of developing fungal diseases and physiological disorders (Bakker,1991). Therefore humidity control is essential in these greenhouses.

Common practice to remove moisture is to simultaneously ventilate and heat, losing not only vapour and thereby latent heat, but also sensible heat. This adds a large fraction to the heating costs of the Dutch glasshouse industry. As an alternative, several dehumidifying systems have been developed and tested (Boulard *et al.*, 1989; Chasseriaux, 1987; Isetti *et al.*, 1997). The main problems of these systems concern high energy consumption due to forced air circulation, large humidity gradients in the greenhouse, a too low capacity, high air velocities in the greenhouse and large size of the installation with high light interception.

A new type of a dehumidifying air conditioner has to be developed that eliminates most of these problems.

In this paper the layout of the final system and its operation is described. Results from computational fluid dynamics calculations that are used in a condensation model are discussed. Experimental results with a one metre long prototype are followed by an overall discussion.

2. Layout of the system

After studying some possible geometry's by CFD modelling the design according to figure 1 proved to perform best. The dehumidifier has three separate units, a cold section, a warm section, and a sensible heat recovery section. The cold and warm section are formed by pipes (D+F) which are in thermal contact with the plates (E+G). The heat recovery unit (I) in between these two sections recovers sensible heat from the incoming flow (A) to the returning flow (B).

The air enters the configuration at A and passes the heat recovery unit (I) before it enters the cold section of the system (D+E). Here the precooled air is cooled and dehumidified. The condensed water is removed through a hole (J). The chilled air flows back (B) recovering sensible heat from the incoming flow (A). The air is then heated in the warm section of the system (F+G). The flow is forced through the system by natural convection due to the acting temperature differences. The system is best powered by a heat pump. In this way the latent and sensible heat removed from the air in the cold section is pumped to the warm section and heats the greenhouse. So the gained latent heat and the energy to drive the heat pump are used in the greenhouse to cover (part of) its heat demand. This will save a considerable amount of energy (Saye *et al.*, 1999).

The installation is small and can be installed below the crop in a decentral way preventing large temperature and humidity gradients.

3. Results

3.1. Computational fluid dynamics calculations

The system performance is affected by geometry, shape, and dimensions. This is not predicted by conventional means. Therefore the performance is studied by calculating the airflow and temperature distribution (figure 2) using computational fluid dynamics (CFD). A commercial CFD-program FLUENT is used. The condensation of water vapour is not included in this program. Density differences drive the air circulation and the effect of decreasing water vapour content on the air density is small compared to that of the effect of temperature differences (approximately 7 times smaller around 20°C calculated from tables (Weast, 1974) and the saturation curve (Ham, 1984)). So excluding condensation has a small effect on the total mass flow. In the modelling only a slice parallel to the hot and cold plate containing half a plate and half the space in between the plates is represented because of the geometrical symmetry.

The dimensions of the design are firstly varied for a set of conditions (warm and cold pipe temperature of 50°C and 5°C, a surrounding temperature of 20°C, a plate thickness of 1 mm and a pipe diameter of 1 cm). The airflow and the amount of heat exchanged are optimal at a width of the cold and hot plate (0: b and d) of 5 cm, a length of the heat exchanger (c) of 10 cm, a height (e) of 10 cm and a distance between the plates (a) of 0.9 cm. A prototype with these dimensions is built and tested.

3.2. Condensation model

Cooling a gas including condensation of water vapour requires that the relatively large enthalpy of phase change be removed, resulting in large heat transfer rates. At a surface below dewpoint temperature condensation will appear. The amount of condensed vapour is calculated using partial vapour pressure, surface temperature, the energy and mass balances and equations describing the heat and mass transfer. Analytical equations (VDI, 1988) are solved discretely in the gas flow direction. The airflow through the system is derived from CFD-calculations. The heat transfer coefficient is based on the Nusselt number for a forced flow passing a plate. The mass transfer coefficient is calculated from the heat transfer coefficient and the Lewis number. For a cold section temperature of 10°C and a warm section of 50 °C, 0 shows the temperature and relative humidity in the system as a function of the position (0 m is the inlet, 0.4 m is the outlet) for a range of initial relative humidity inlet flows at 21.5°C. In the first section of the system, the heat recovering unit (0-0.1 m), condensation already occurs at a 100 % initial relative humidity. For an initial condition of 60 and 80 % RH condensation only occurs in the cold section. So then the temperature profiles coincide as can be seen in figure 3. For an initial relative humidity of 40 % no condensation occurs, while the temperature of the cold section (10°C) is above the dewpoint temperature. The amount of condensation per metre of system for the given conditions is calculated at : 40% 0 ml/hour, 60% 11.3 ml/hour, 80% 33.2 ml/hour and 100% 56.4 ml/hour. For a cold section temperature of 6.4°C and warm section at 50°C the calculated values are : 1.3, 23.9, 46.8, and 71.7 ml/hour.

3.3. Prototype in greenhouse

A one-meter long prototype of the dehumidifier (0) is built and placed in a greenhouse. Warm and cold water with fixed temperatures is pumped through the warm and cold section of prototype. The amount of condensate dripping out is measured as a function of time with a tipbucket rainmeter giving an impulse every 10.98 ml. Temperature and relative humidity are monitored in time during the experiment by a dry and wet bulb thermometer positioned between the full grown cucumber crop at a height of 3 metre.

In 0 results are depicted for a 5.5 days long period. The top figure shows the temperature differences between the cold and the warm section and the amount of condensate removed in ml/hour. The bottom figure shows the temperature and relative humidity. On the first one and half day (0-1.5) the cold and warm water temperatures were set at 9.8 °C and 50 °C respectively. During the second period (1.5-4.5) the cold water temperature is lowered to 6.4 °C. For the third period (4.5-5.5) the warm temperature is raised to 64 °C. For high humidity (around 90%) periods the amount of condensate collected is approximately 50 ml/hour in the first period, and 65 ml/hour during the other two periods. The average humidity in the greenhouse for the third period is low at nighttime compared to the other periods, explaining the same level of condensate removal as in the second period where the warm plate temperature is lower. At 2.6 and 3.7 days a dip in the temperature difference between the cold and warm section occurs caused by a lack of cooling capacity during high greenhouse temperatures.

4. Discussion

The experiment proves the system functions in a greenhouse, though all the benefits mentioned in the introduction can not be proven because the experiment was performed with a small prototype so greenhouse conditions were not affected. The calculations with the condensation model including the CFD-calculations underestimate the amount of condensate removed from the greenhouse air with 10% compared to the experimental results. For humid conditions (90 % RH) one meter of system removes 44.8 ml/hour according to the calculations and about 50 ml/hour during the experiment for a cold temperature of 10°C and a warm temperature of 50°C. For a cold section of 6.4 °C, prediction by the model is 59.3 ml/hour and 65 ml/hour is measured in the experiments. So there is satisfying agreement between modelling and experiment.

These preliminary results can be used to estimate the energy efficiency of the system for a whole year. First predictions show that 4 to 7 % energy can be saved in a conventional greenhouse. For modern well-insulated greenhouses these savings are much higher while it prevents ventilation to dehumidify the air.

Acknowledgement

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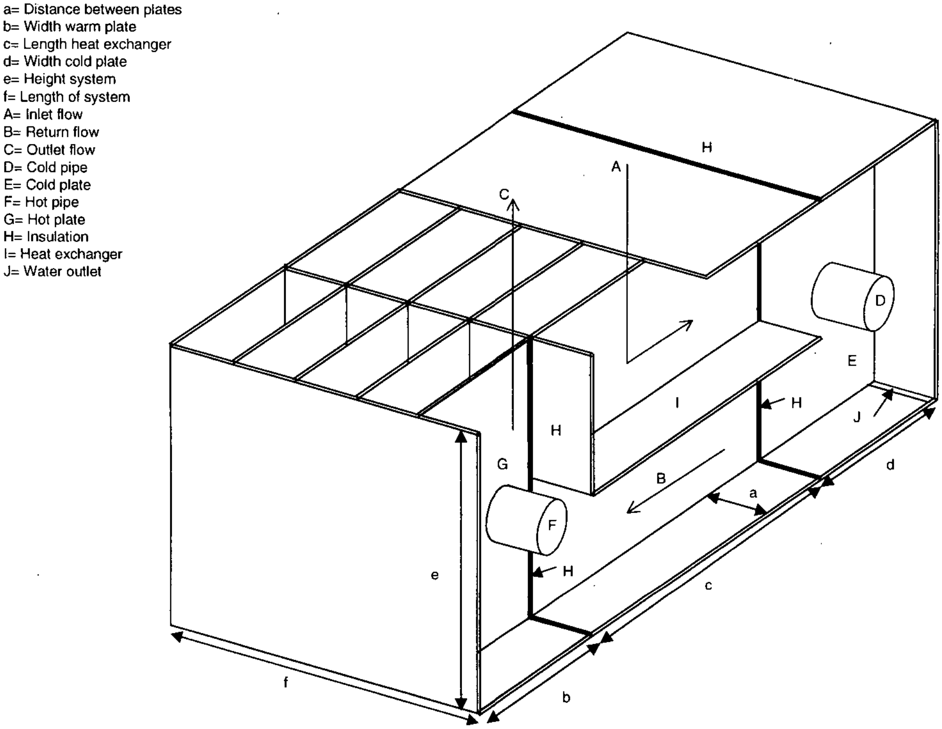
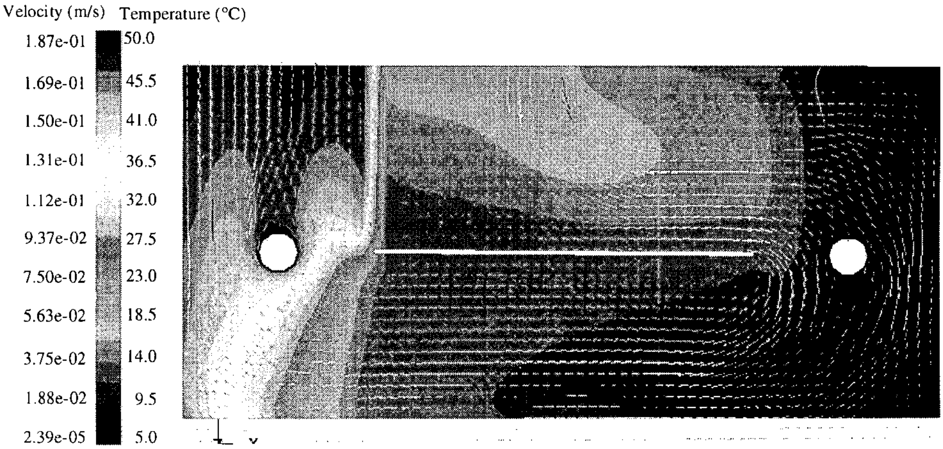
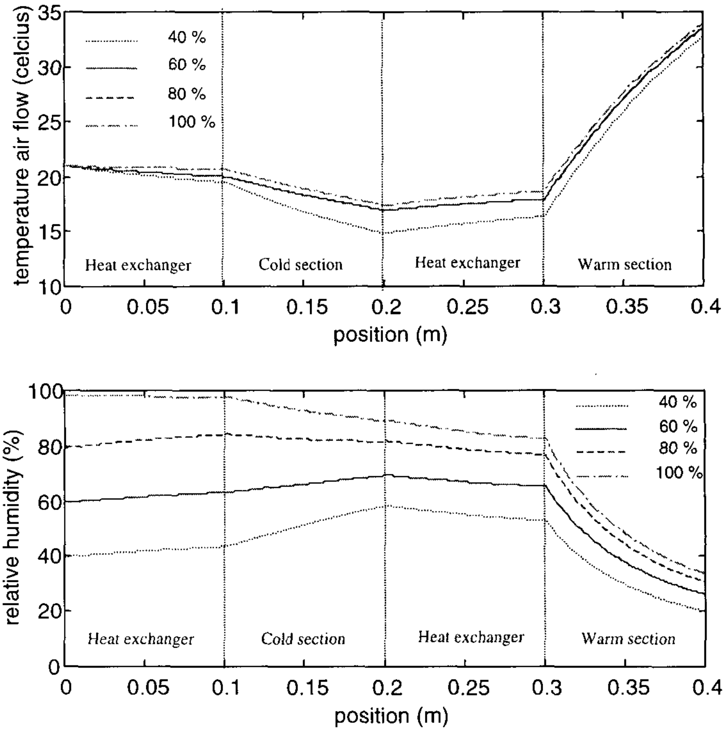
Figures



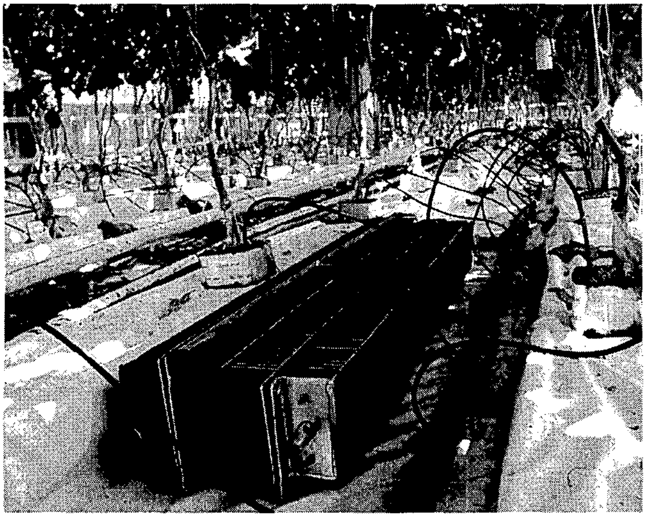
1. Schematic representation of the dehumidifier



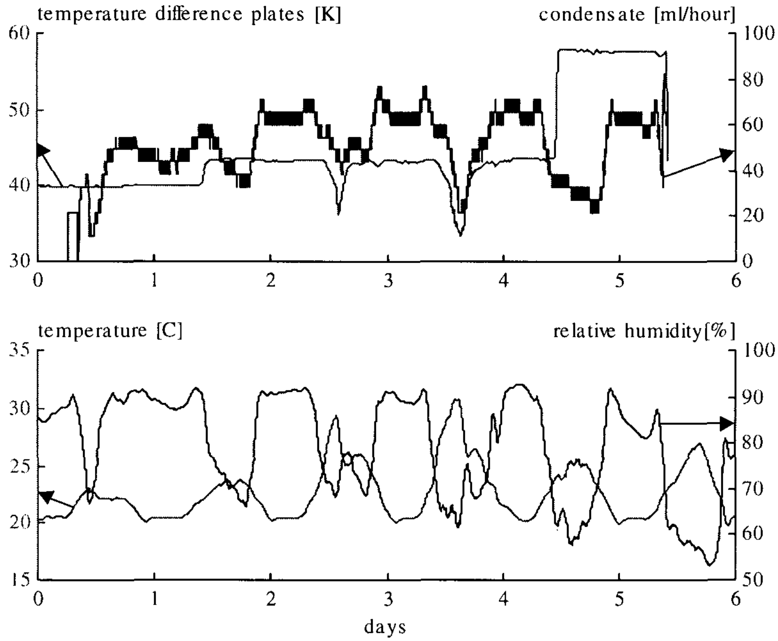
2. Velocity and temperature profiles in the system resulting from a CFD-calculation.



3. The temperature of the airflow (top-figure) and the relative humidity (bottom-figure) as a function of the position in the system calculated using the condensation model for different initial relative humidity airflow (40,60,80 and 100 %) at 21.5°C. The cold section is 10°C and the warm section is 50°C.



4. One-meter long prototype placed in a greenhouse with cucumbers.



5. In the top figure the temperature difference between the warm and cold section is indicated together with the condensate removal. In the bottom figure the temperature and relative humidity in the greenhouse are shown as a function of time.