Heat Recovery in Greenhouses: A practical solution

D.R. Rousse[1], D.Y. Martin[2], R. Thériault[1], F. Léveillée[3], R. Boily[1]
[1] *Université Laval, Cité Universitaire, Québec, Qc, Canada, G1K 7P4*
[2] *Ministère de l'Agriculture, des Pêcheries et de l'Alimentation du Québec*
[3] *Centre d'Information et de Développement Expérimental en Serriculture (CIDES)*

Abstract - In recent years, passive infiltration of air into greenhouses has been reduced drastically. However, very low air exchange rates can lead to abnormally high levels of humidity which can damage harvests. Hence, farmers have to ventilate. In an effort aimed at reducing heating costs related to ventilation, a heat exchanger was designed to be used as a dehumidifier. A CDN$ 2 000 air-to-air multi-pipe counterflow heat exchanger unit was installed in a greenhouse used for the experimental cultivation of hydroponic tomatoes and cucumbers during the winter of 1996. The first series of tests, carried out between March to May 1996 in a 576 m^3 enclosure, demonstrated that average efficiencies of η=84% and η=78% were obtainable with air volumetric exchanges rates of 0.5 and 0.9 change per hour, respectively. Latent heat was found to play a major role in the overall heat transfer, contributing about 40% of the total energy exchanged in some situations. The unit made of plastic is durable and rot and rust resistant. Its efficiency, mainly because of its very low level of compactness, was found to be very good in the presence of frost and ice. A commercial implementation is now considered as well as experiments in broiler houses.

Keywords - Greenhouse, latent heat recovery, plastic heat exchanger, payback analysis.

Author to whom further correspondence should be addressed:
Daniel R. Rousse
Department of Mechanical Engineering,
Université Laval, Cité Universitaire,
Québec, Qc, G1K 7P4
CANADA

1. INTRODUCTION

1.1 Context

In recent years, passive infiltration of air into greenhouses has been reduced from three or more air changes per hour to less than one half [1]. The reduction of air infiltration into greenhouses leads to significant reductions in heating costs. However, this may be achieved to the detriment of the crops being grown. Very low air exchange rates can lead to abnormally high levels of humidity both during the daytime and at night.

The characterization of the influences of humidity on plant response has not yet been thoroughly investigated unlike those of light, temperature, and carbon dioxide [2]. This may be, in part, due to the difficulty in measuring and controlling humidity in large enclosures and to relate the humidity measurements to the transpiration rates of the crops [3]. Nevertheless, an afternoon above 95% RH may kill or damage a whole harvest. Furthermore, even when the crops are producing at high levels of humidity without any damage, their production rate is much lower than in a controlled environment.

To avoid excessively high humidity levels, venting and heating often remains the only solution to the farmer and this may annihilate the gains achieved by the reduction of infiltration. Traditional heating and ventilation systems result in an inefficient and expensive use of energy, especially during winter in cold regions of the world.

1.2 Economics in cold regions

The *Syndicat des Producteurs en Serres du Québec* (SPSQ) [4] lists the problem of humidity control in greenhouses as a top priority for this industry. Table 1 [5] indicates the average annual energy requirement per unit area and its corresponding unit cost of operation, for a greenhouse located in Quebec (Canada), as a function of its dehumidification strategy.

In the Table, the first row corresponds to unit heating costs when dehumidification is due to exfiltration of moist air only (balanced by infiltration of cold air), while most of the vapour condenses on the roof and the walls of the greenhouse. This situation is mostly found in old installations where passive infiltration is important. The second row shows figures for a situation where a whole change of air is made in the greenhouse in an hour. The last results presented in the third row of Table 1 pertain to the situation where the farmer ventilates to maintain an adequate level of humidity all the time.

Table 1 shows that in cold climates: (1) about 13% to 18% of the heating costs of a standard greenhouse are due to humidity management; (2) proportional ventilation is about 2.5 to 4.7 CDN$/m^2 per year more expensive than no ventilation. The difference in energy consumption is relatively small due to the dynamics of the greenhouse:

- In summer, late spring and early fall, there is no need to heat the facilities as the direct sunlight is sufficient to maintain a temperature of about 20^{o}C and above. In these periods there is a need for ventilation to avoid high temperatures but this has no effect on the heating costs.
- In winter, most of the water content is removed from the moist air as condensation occurs on the boundaries of the greenhouse. In this period, as the activity of the crop is low due to poor direct sunlight, the water production of the crop may be balanced by condensation on the walls and no ventilation is needed in some cases.
- The critical periods of the year for water transpiration by the plants are spring and fall where there is enough sun to permit a very active growth and not enough condensation on the walls. In these critical periods, ventilation is needed. Yet another reason for the small overall difference in energy requirements with and

without ventilation is that crops transpire mostly in the daytime thus delaying the ventilation need to the afternoon and evening, only.

The results presented by Johnstone and Ben Abdallah [6] confirm those presented in [5] as they also estimated the annual average unit ventilation cost at about 3CDN$/m^2. This results in about 2.6 MCDN$ in heating costs due to ventilation for the 88 hectares of crops being grown in Quebec only [7]. When ornamental plants are considered, this represents another 134 hectares. Globally, in Quebec only, about 5MCDN$ are being spent each year to handle humidity in greenhouses.

1.3 Possible solutions

To decrease the energy consumption of this sector of activities, heat exchangers have been considered. Jorgenson and May [8] indicate that the thermal efficiencies of commercially available air-air heat exchangers used in Canadian agriculture are currently about 40%. Because they are both expensive and subject to freezing at subzero temperatures, these models failed to meet the needs of Quebec's producers. However, Lepoitevin et al. [9] and Brundrett et al. [1] have demonstrated that it is possible to design and build heat exchangers with efficiencies in the order of 70 to 80%, which are particularly suitable for the dehumidification of greenhouses. It has been shown that these exchangers perform adequately under freezing conditions. Despite their efficiencies, these exchangers have not been retained for commercial purposes probably because of their fragility. Lepoitevin et al. [9] used mainly plywood and polyethylene films to create rectangular parallel channels while Brundrett et al. [1] employed seven thin polyethylene tubes to carry the ventilation air within a larger polyethylene envelope in a counterflow arrangement.

1.4 A practical solution

The aim of the research presented in this paper was therefore to find a way for fruits and vegetables producers in Quebec (Canada) to lower their operating costs by recovering some of the heat lost from their ventilation systems. The main restriction of paramount importance was cost. Based on discussions with the *Union des Producteurs Agricoles du Québec* (UPA) it was found that the overall cost of the unit should be less than CDN$2000 for a typical small greenhouse of about 600 m^3 (or about 220 m^2). The other design restrictions had to do with: (1) the ability of the producers to assemble, operate, repair, and maintain the unit; (2) the corrosion and rottenness resistance of the material; and (3) the overall operating efficiency with or without frost.

This paper presents a five tube-in-shell counterflow heat exchanger prototype used as a recovery system that involves high efficiencies, low costs, durability, user-friendliness, and a payback period that should be less than 3 years. The core of the exchanger is made of corrugated plastic (drainage) tubing involving integral annular fins. The prototype was located along the side wall of a 596 m^3 greenhouse. The simple unit can be understood and build by any trained person familiar with drainage tubing technology. Since the first prototype has been constructed, others were designed and built for other applications pertaining to agriculture.

The remainder of the paper presents a detailed description of the prototype followed by a section that introduces the numerical design tool used in the conception of the unit. Then, results are presented and other applications are discussed. The synthesis of the first prototype results are discussed in the concluding section.

2. DESCRIPTION OF THE PROTOTYPE

After a feasibility study, it was decided to build a multi-tube counter-flow heat exchanger. In view of the restrictions formulated in the introduction, corrugated and flexible thermoplastic drainage tubing [10] was selected to serve as the core of the multi-tube exchanger, four thermoplastic tubes 76 mm I.D. wrapped around a central 101 mm I.D. tube were used. The external kernel or shell of the exchanger that carries the warm and moist air was a tube 305 mm I.D. with a corrugated outer surface (361 mm O.D.) and a smooth inside surface to permit ease of assembly [11], see Fig. 1.

Due to the unlimited amount of space available within greenhouses and because the major part of the exchanger could be buried, compactness [12] was not a critical parameter here. As a result the heat transfer area density of the first prototype was about 27 m^2/m^3. The first exchanger prototype was 24.3 m long and involved only about 66.9 m^2 of direct exchange area. In the calculation of the exchange area, the effects of the corrugations have been taken into account. This yields about 100% increase over smooth tubes. Figure 2 shows a longitudinal cross-section of the 101 mm I.D. The surface increase for the 76 mm tube is the same.

Figure 3(a) shows the warm end of the unit: the four gray tubes are carrying the warm moist air which is injected in the external shell. Figure 3(b) shows the cold end of the prototype. It can be seen in this figure that the ventilator is built into the plenum and that the tubes are isolated to prevent condensation in the greenhouse.

The size of the prototype is justified by the requirement to operate at subzero temperatures for which accumulation of ice should not significantly increase the pressure drop and decrease the overall efficiency. In addition to having a low area density, the unit has been designed to permit a maximum volumetric exchange rate of one volume per hour in a 576 m^3 greenhouse located at the *Institut des Technologies Agro-alimentaires de St-Hyacinthe*, Québec. The greenhouse is part of a larger complex involving several units. It is entirely covered by polyethylene films on the top and on its sides.

3. NUMERICAL DESIGN TOOL

3.1 Original predictions.

Brundrett et al. [1] proposed a simple model to design heat exchangers to be used as dehumidifiers in greenhouses. These researchers validated their model with respect to results obtained from two prototypes. The prototypes involved two air streams separated by a polyethylene film on which condensation occurred as the warm and moist stream reached its dew point. In [1], the comparison between experimental and predicted performance is reported to be excellent. In that study [1], the discrepancies are believed to be due to heat transfer to the outer shell of the exchanger which is neglected in the model.

Brundrett’s model is reported here for the sake of clarity. It is noted that the resistance of the polyethylene tubes used in this study [1] is small (hence neglected) compared to the film resistance. For the dry portion of the prototype, the overall heat transfer coefficient is:

$$U_{dry} = h_i \times h_{dry,o} \,/\, \left(h_i + h_{dry,o}\right) \tag{1}$$

while the wet zone overall heat transfer coeficient is:

$$U_{wet} = h_i \times h_{dry,o} \times h_{wet,o} \,/\, \left(h_i \times h_{dry,o} + h_{wet,o}(h_i + h_{dry,o})\right) \tag{2}$$

where:

$$h_i = 0.00053018 \; Re_{D_i}^{0,8} \,/\, D_i \tag{3}$$

$$h_{dry,o} = 0.0003175 \; Re_{D_{ho}}^{0,77} \,/\, D_{ho} \tag{4}$$

and

$$h_{wet,o} = 0.725\ \left[g\rho_l(\rho_l - \rho_v)k_l^3 h'_{fg} / \left(N\mu_l (T_{sat} - T_w) D_o \right) \right]^{1/4} \tag{5}$$

where

$$h'_{fg} = h_{fg} + 3/8 c_{p,l}(T_{sat} - T_w) \tag{6}$$

A debatable or arguable feature of the model proposed by Brundrett et al. [1] is that in the wet zone of the prototypes, Eq.(2), the authors add the resistance due to condensation to those due to external and internal convection found by standard correlations for dry and smooth surfaces [13]. This is equivalent to neglecting condensation as this resistance is very low compared to the other two. However, the mass fraction of air in the warm stream is significant. Thus, a gas phase resistance for the superheated water vapour will act in series with the film resistance and therefore Eq. (2) may not be as inadequate as it could be assumed initially. This may explain the exponent *0.77* in Eq. (4).

Nevertheless, based on the model of Brundrett et al. [1], a one-dimensional basic numerical design tool was developed and implemented to allow for the design of the above-described prototype. This was later found to lead to overall experimental results that were in fair agreement with the predictions, but the latter could be improved slightly with an improved model. Some modifications (heat transfer correlations) and improvements (pressure drop estimates) were incorporated in the original model (similar to that of Brundrett et al. [1]). The improved design tool was found to provide better predictions when compared to the experimental results. Subsequent units were designed based on the improved model. The next subsection presents the modifications incorporated into the original tool.

3.2 Improved predictions

The proposed approach is still considering a one dimensional simulation of the exchanges that includes the latent heat exchanges along the axis of the exchanger. The correlation that was used for the internal and external surfaces of the five tubes that constitute the core of the unit is the acknowledged relation proposed by Gnielinski [14,15] with the entrance correction factor derived by Hausen [16,17]. For the internal Nusselt number this yields:

$$Nu_i = \frac{(f/8)\left(Re_{D_i} - 1000\right)\mathrm{Pr}}{1 + 12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3} - 1\right)} \left[1 + \left(\frac{D_i}{L} \right)^{2/3} \right] \tag{7}$$

where Re_{Di} is the Reynolds number, based upon the tube diameter D_i, Pr is the Prandtl number, and f is the friction factor given by:

$$\frac{1}{\sqrt{f}} = -0.86 \ln\left(\frac{\varepsilon / D}{3{,}7} + \frac{2{,}51}{Re_D \sqrt{f}} \right) \tag{8}$$

For corrugated drainage tubes, there are no data available to quantify the relative roughness, ε/D. Hence, after a series of pressure drop measurements, ε was approximated to an average of 0.001m. The diameter D was the internal diameter D_i for the cold side and the hydraulic diameter D_{ho} for the warm side of the exchanger. The inner diameter D_i of the corrugated tubes was taken as the minimum internal diameter and the outer diameter D_o of these same tubes was taken as the maximum external diameter. The correlation employed here [14,15] is for straight tubes, that is with no corrugations. This effect will be investigated later.

For the external Nusselt number, based on the external hydraulic diameter, this gives:

$$Nu_{dry,o} = \frac{(f/8)\left(Re_{D_{h,o}} - 1000\right)\mathrm{Pr}}{1 + 12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3} - 1\right)} \left[1 + \left(\frac{D_{h,o}}{L} \right)^{2/3} \right] \tag{9}$$

where Re_{Dho} is the Reynolds number, based upon the tube hydraulic diameter D_{ho}. The hydraulic diameter is calculated with reference to the shell and five tubes perimeters. The Nusselt relation, Eq.(5), does not account for the air flowing on the wetted surface. Shekriladse and Gomelauri [18] proposed a correlation for the condensation of a pure vapour that accounts for flow velocity in the direction normal to the tubes while Honda and Fuji [19] successfully tested that correlation for fluid flow aligned with the tubes. This correlation is:

$$Nu_{wet,o} = 0.64 Re_{D_{h,o}}^{1/2} \sqrt{1 + \left(1 + 1.69 \frac{g h'_{fg} \mu_l D_{h,o}}{U_\infty^2 k_l (T_{sat} - T_s)}\right)^{1/2}} \qquad (10)$$

The presence of even a small quantity of non-condensable gas in the condensing vapour has a profound influence on the resistance to heat transfer in the region of the liquid-vapour interface [20]. In agreement with what was proposed by Brundrett et al., the interfacial resistance was assumed to be that of dry air.

Thus, the thermal resistance of the tube wall can be predicted with the following expression:

$$R_T = \frac{1}{\pi D_i L h_i} + \frac{R''_{,i}}{\pi D_i L} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{R''_{,o}}{\pi D_o L} + \frac{1}{\pi D_o L h_{,o}} + \frac{1}{\pi D_o L h_{dry,o}} \qquad (11)$$

where $R''_{f,i}$ and $R''_{f,o}$ are the thermal resistances due to fouling. The fouling resistances are initially negligible. However, due to the corrugations on the internal and external surfaces of the tube, these resistances are expected to become significant as dust will accumulate on the tubes. The resistance due to conduction is evaluated with a thickness of 1 mm and a thermal conductivity of 0.15 W/mK.

The outer shell was assumed to be adiabatic. The predictions then have to include the specifications of the psychrometric properties of the hot air, with wet and dry bulb air temperatures and absolute pressure being required. The prediction model thus determines where the warm fluid will experience condensation of moisture by dropping below its dew point temperature. The calculation of the overall exchanger is then divided into two sections: the first where heat transfer occurs exclusively by sensible transfer and the second where heat transfer involves latent as well as sensible heat.

To complete the model, two more basic relations are provided here: one for the overall heat transfer and the other for the efficiency of the unit. The effectiveness of the unit, based on enthalpy variations, was also calculated but neglecting the enthalpy increase due to the fans. To accurately account for the heat transfer between the two fluids with phase change, an enthalpy based approach was introduced in the prediction model. The overall heat transfer between the hot and cold fluids is given by:

$$q = \dot{m}_o \left(i_{o,inlet} - i_{o,outlet}\right) = \dot{m}_i \left(i_{i,outlet} - i_{i,inlet}\right) \qquad (12)$$

An iterative procedure is employed in the two sections until a balance is obtained in the calculation of the heat transfer with Eq.(12) and that with UA LMTD [13].
The contribution of latent heat to the total heat transfer was estimated with:

$$L = \left[1 - \frac{c_p (T_{o,inlet} - T_{o,outlet})}{i_{o,inlet} - i_{i,inlet}}\right] * 100 \qquad (13)$$

The efficiency is defined as:

$$\eta = \frac{T_{i,outlet} - T_{i,inlet}}{T_{o,inlet} - T_{i,inlet}} \qquad (14)$$

where subscript *i* refers to the stream inside the tubes and subscript *o* refers to that outside the tubes or into the kernel. The definition of the effectiveness based on enthalpy differences enabled evaluation of the ratio of the actual exchange to the maximum possible exchange, while that of the efficiency is the farmer's interest. The difference between the two occurs when the water vapour in the warm stream, *o*, condenses on the tubes walls. As the mass fraction of air into the warm and humid stream is high the effectiveness is about equal to η most of the time.

The water content on the warm side at the five stations in the exchanger was evaluated with the RH and temperature measurements and relations providing the saturation pressure extracted from the program PLUS.EXE proposed by Albright [21] and reproduced below .

For humid air below zero Celsius, the saturation pressure is evaluated with :

$$P_{sat} = \exp\begin{pmatrix} -\dfrac{5800}{T} + 1.391493 - 4.864024E\text{-}02*T \\ +4.1764768E\text{-}05*T^2 \;\; -1.4452093E\text{-}08*T^3 \\ +\; 6.5459673*\log(T) \end{pmatrix} \tag{15}$$

while for humid air above zero Celsius, the saturation pressure is evaluated with :

$$P_{sat} = \exp\begin{pmatrix} -\dfrac{5674.5359}{T} + 6.3925247\text{-}9.677843E\text{-}03*T \\ +6.22157E\text{-}07*T^2 + 2.0747825E\text{-}09*T^3 \\ -9.484024E\text{-}13*T^4 + 4.1635013*\log(T) \end{pmatrix} \tag{16}$$

Finally, the enthalpy on the cold and warm sides was evaluated as:

$$i_{mix} = c_{pa}(T-273.15) + W(i_{gv} + c_{pv}(T-273.15)) \tag{17}$$

3.3 Pressure Drops Considerations

A single tube and shell design requires considerably more length and implies large pressure drops to achieve an acceptable level of effectiveness. This is one of the reasons which motivated the multiple tubes design. Moreover, farmers possessing greenhouses usually employ fans of about 0.25 to 0.75 HP to ventilate their facilities and therefore the objective was to allow farmers use their own fans in their exchanger. The Darcy-Weisbach equation [13] was used to estimate the pressure drop within the tubes and in the shell with the friction factor given by Eq.(8).

The actual work, intended to improve the original model presented here, accounts for the exhaustive review recently presented by Rifert [22] on condensation on horizontal tubes. In the improved model, the heat transfer coefficient estimate will also be based on the work by Amadek and Webb [23], which gives detailed calculation of h_i along the fin height and tube perimeter with the assumption that the pressure gradient is constant.

4. EXPERIMENTAL RESULTS

A selection of experimental results is presented here for an evaluation period ranging from March 21st to May 21st 1996. All data cannot be presented herein and the interested reader should contact the first author for further details. Two operating conditions were used: 0.5 and 0.9 air change per hour. 1 chg/h was the original target but the pressure drops were such that the available Delhi 410/610 (0.25 HP) fans did not provide enough power. The rate of 0.9 air change per hour was found to be the maximum possible given the fan power and the pressure drop.

A Campbell Scientific data acquisition system (21X) was used in conjunction with 45 thermocouples distributed along five measurement locations. Figure 1 shows the location of the nine thermocouples at each measurements locations. Temperature was also monitored at different locations within the greenhouse. Thermocouples were calibrated in an isothermal bath and the reference temperature was measured with an RTD probe. Three humidity probes model 207 from Campbell Scientific where also employed, one located at the exit of the cold fluid, one in the plenum of warm moist air near the exit, and the third one at the center of the greenhouse to monitor and control the global humidity level inside. Two pressure transducers installed into the plenums permitted to monitor the pressure drops and ensure an adequate operating regime of the fans. An electronic tipping trough, shown in the lower right corner of Fig. 3(a), was installed at the warm end of the unit to recover the condensate and establish mass balances. The prototype was tilted about 1^{o} towards the warm end. A general purpose air velocity transducer (Omega FMA-900, 1.5% acc. FS) was used to measure the flow rates on both sides of the exchanger.

Results presented herein for temperature are average temperatures of the cold and warm streams, see Fig.1, at 5 axial locations along the prototype. Standard deviations were about 0.1^{o}C in each case.

4.1 Global Results

In this section overall results are provided for the period extending from March 21st to May 21st. At a rate of $\dot{Q}$=0.5 air change per hour, the average efficiency based on temperature for the whole period of investigation was about : η =84% with a 5% standard deviation. For the results obtained with $\dot{Q}$=0.9 air change per hour, the average efficiency decreased to η=78% with a 3.5% standard deviation.

The experimental results carried out over the two months period indicate that for $T_{i,inlet}$ varying between 1 and 3°C with RH varying between 63% and 70%, the contribution of the latent heat to the overall heat transfer fell within a 39 to 43% range.

The amount of water that condenses on the walls is calculated based on the variation of the absolute water content of the warm moist fluid along the exchanger. A typical rate of condensation is about 1680 ml/h. The maximum condensation rate was found to reach 3200 ml/h when the external temperature was -10^{o}C and the internal temperature 20^{o}C with 85% RH.

The maximum power used by the Delhi fans was 637 W, and the rate of heat gained by the cold fluid varied from 874 W at $T_{i,inlet}$ = 14°C to 3 089 W at $T_{i,inlet}$ = -10°C. This indicates a variation in the COP such that: 1.4 < COP < 4.8. The maximum COP was found to reach 6.9 for very cold winter operation (this was recorded later on a sunny day in January 1997).

4.2 Efficiency Results

Figures 4, 5, and 6 indicate the temperature distribution within the exchanger for three typical days of operation. The first day was March 26th, 1996 when the volumetric flow rate of warm fluid, $\dot{v}_h$, was 0.099 m^3/s and that of the cold fluid, $\dot{v}_c$, was 0.079 m^3/s. The profile presented in Figure 4 is typical of what was observed when the prototype operated at 0.5 air change per hour. For this case, the relative humidity at the warm exit of the cold stream was 15.7% while it was almost completely saturated at 93.5% at the cold exit of the warm stream. The efficiency, based on equation (8), was 89%. The heat recovery was excellent: 1948W. And at that time of the day, provided that the fans needed 355W, the COP was 5.51.

Figure 5 shows results for April 5th , 1996 when $\dot{v}_h$ was 0.148 m^3/s and $\dot{v}_c$ was 0.141 m^3/s. Similar trends can be observed. For this second case, the relative humidity at the warm exit of the cold stream was 18.9% and the efficiency was 81%. 2856W were recovered while 637W were used: the COP was 4.48.

Finally, Figure 6 shows the temperature distribution on April 30th, 1996 for similar flow rates. For this last case, the relative humidity at the warm exit of the cold stream was very high at 72.4%. In this case however, the inlet air on the cold side was at 14.4^{o}C which is a very high temperature for which the use of the exchanger is not strategically required. Still, the efficiency was 74.6%. The COP was nevertheless 1.39.

Examination of the results in Figures 4, 5, and 6 show an appreciable difference in the temperature distributions. When the inlet temperature on the cold side was low (Figures 4 and 5), the temperature distribution was nonlinear while for a higher inlet temperature (Figure 6), it was nearly linear. These differences were mainly caused by the exchange of latent heat as soon as the warm air reached the dew point: for the first two days, the dew point temperature occurs closer to the entrance of the warm stream. An interesting consequence of this is that the more the air is cold outside, the better the efficiency and COP will be for a given activity level of the crop. As heat transfer occurs from the greenhouse through the external shell of the exchanger, temperature profiles should be affected by such transfer. But in this context, this amount of heat is small compared to the exchange between the hot and cold streams.

Figure 7 reports the variation of the exchanger efficiency for March 26th1996. In this figure, it is shown that the average efficiency for March 26th was 82% and this accounts for the dynamics of the greenhouse that is: (1) the transpiration by the plants in the daytime; (2) the incoming feeding water in the volume; and (3) the condensation occurring on the surfaces. The interesting result here is that the efficiency is maximum when the crop needs it the most. In the daytime, the sun is responsible for high growth rates and therefore high transpiration rates. This in turn raises the ambient relative humidity. As a result, the dew point in the warm air stream of the exchanger is reached near the entrance and, consequently, this increases the amount of latent heat transferred between the hot and cold streams because condensation occurs on a wider surface. Physically, for given inside and outside temperatures, the more humidity there is inside, the better performance of the unit.

4.3 Psychrometric Results

The relative humidity was also monitored to assess the ability of the unit to fulfil the needs of the plants. It is worth noting that 0.9 air chg/h is not enough to maintain an adequate level of humidity in the complex all year long: it should be adequate about 80% of the time. But for this design, only general characteristics were to be obtained. The test was carried out in the critical period of growth for a greenhouse in Québec. As a result, it was expected that the humidity level would be very high in this period even under operation: traditional ventilation had to be used as a complement.

Figure 8 shows the relative humidity distribution for March 26th 1996. The results for the humidity in the greenhouse (diamonds) show a first peak early in the morning: March 26th was sunny and the plants were active early. The humidity had to be lowered with standard ventilation as the unit was not able to deliver a sufficient flow rate to evacuate a sufficient amount of moisture. A second peak appears at about t = 900 min, that is when the sun sets. At that time, the greenhouse had to be closed as the external temperature became too low to maintain an adequate temperature level inside. The interesting part of the curve is that the unit was able to lower the humidity level rapidly after sunset. In brief, a bigger unit would have been needed only in the morning for that day. The inlet stream humidity results (squares) show the period in the day when it stopped: the unit operated almost continuously. The last results (crosses) show that air was saturated in the warm stream except when additional ventilation was used. In these conditions, the humidity level in the greenhouse was below 75%.

Figure 9 presents typical results obtained for a period ranging from April 5th to April 9th 1996. This sequence demonstrates the performance of the prototype as a dehumidifier over an extended period. At that time, about 300 mature plants of tomato and cucumber were growing. During this period, the exchanger was operated continuously with a RH threshold of

75%. The transpiration cycle of the plants can be interpreted as follows. The photosynthesis activities diminish after sunset. As shown in the figure, the relative humidity then reaches peak lows of about 79 to 82%. The high peaks occur at about noon with maximum relative humidity of about 90 to 91%. On an average, the relative humidity was about 85% in the greenhouse.

Again, it is shown in Figure 9 that the prototype is too small to permit a total compensation for the needs of the plants: the threshold of 75%RH is never reached. This was predicted as the capacity of the exchanger is about 5 times lower than the maximum greenhouse requirement. However, these results are interesting as they permit one to compare the humidity management using the undersized unit with traditional ventilation techniques. Here, the cycles never reach 100% relative humidity which would sometimes be nearly the case with manual ventilation. This indicates that although two to five air changes/h may be needed in critical periods, the smaller unit of about one air change/h can nevertheless permit preventing relative humidity to shoot above 91%. Results from Figure 8 and 9 were used in the design of a second generation of pre-commercial units that are now undergoing a more thorough experimental testing procedure.

Knowing both incoming and outgoing volumetric flow rates in conjunction with their relative humidities and temperatures, a mass balance can be performed for water vapor in the greenhouse. Figure 10 shows the amount of water evaporated by the plants that is not condensed per period of 10 minutes. The figure indicates that a minimal transpiration rate of 0.65 liter per 10 minutes period was monitored in the unit between April 5th and April 9th while the maximum rate achieved was 1.10 liter per 10 minutes period. These rates correspond to about 0.31 liter/day-plant and 0.52 liter/day-plant. The average curve indicates that the maximum activity occurs in the daytime with about 40% more transpiration than at night.

The results presented in Figure 10 permit to design new units according to the number of full grown plants to be planted in a facility.

4.4 Pressure Drop Results

Figure 11 indicates the relative pressure drops in the exchanger for 0.9 chg/h that is when $\dot{v}_h$ was 0.148 m^3/s and $\dot{v}_c$ was 0.141 m^3/s. Figure 11 indicates that the pressure drop within the inner tubes is much higher than that in the external shell. The pressure drop for the cold fluid is higher than its warmer counterpart, and this is in agreement with the roughness differences. This was used as a limitation when other designs were proposed. Here, the Delhi 0.25 HP fan is just slightly too small to provide the desired 1 air chg/h flow rate.

4.5 Ongoing Experiments

The interesting results obtained with the first prototype convinced the *Centre d'Information et de Développement en Serriculture* (CIDES) to support the project and to elaborate a development program for new units. Several issues are discussed here without any details to avoid this paper to become overly lengthy.

Further experiments are now being carried out in a laboratory to obtain a suitable correlation for the average and local Nusselt numbers on the warm side of a double wall counter-flow exchanger with a view to improving the numerical predictions.

The original prototype was also used during the fall of 1996 and winter of 1997, and it was found that frost did not significantly influence the overall operating efficiency. Ice was only found near the warm air outlet (cold air inlet) at thicknesses of less than 5 mm even though saturated warm air was in contact with the cold tube carrying air at -22°C. This ice was detected in the last meter of the unit which reduced the performance somewhat due to additional flow and thermal resistances in this small section. This suggests that the next prototype might profit from a higher area density.

In winter 1998, two exchangers were installed in a commercial greenhouse complex near Victoriaville, Quebec. These units permitted one to control the global ventilation requirements in terms of humidity, 80% of the time. It is worth mentioning that when the normal solar irradiation reaches 300 W/m^2 for an outside temperature of –5^{o}C, the temperature in the complex rises above 22^{o}C and then natural ventilation is used via openings in the roofs. Results permit one to estimate the payback period for this complex to be about 3 years. Average efficiencies of η=65% were obtained with a corresponding air volumetric exchanges rate of $\dot{Q}$=1.3 change per hour in a 30m x 26m greenhouse complex which is nearly 6 times bigger than that pertaining to the first prototype.

Fall 1997 saw a second application for this type of exchanger and a new prototype, with an improved design, installed in a broiler house to demonstrate that such units could sustain operation in dusty environments. The successful demonstration convinced the *Fédération des producteurs de Volailles du Québec* to launch a full scale test in broiler houses. In broiler houses, the ventilation need increases from zero to 0.48 l/s per chicken in the spring/fall seasons within the 40 days growth period. At the end of this period, each of the 1.8 kg chickens will dissipate 7.3 W in heat but this is insufficient in winter to maintain the ambient room temperature at 21.1^{o}C [24]. As continuous ventilation is required to evacuate both ammonia and water vapor [25], the installation of a heat exchanger is highly desirable. Economically, due to the operating regime, this represents about $18 000 per year in ventilation costs only, for a 240 000 chicken facility [26].

Another test in broiler houses is also underway at the experimental farm of the Québec Government in Deschambault, Québec where chicken are grown in four separate but identical test compartments. Two of these small compartments, involving about 200 chickens, are equipped with exchangers having a 86 l/s capacity. The use of exchangers is shown to reduce the fossil fuel consumption significantly.

CONCLUSION

A prototype air-air counter-flow multi-tube heat exchanger has been designed and built to meet the specific greenhouse requirements of operating in a cold climate. The uncompact design involving plastic components was retained so as to meet the following requirements:

- low cost, CDN$ < 2000 (3 year pay-back period);
- ease of assembly, maintenance, repair, and operation;
- corrosion and rottenness resistance;
- satisfactory operating efficiency when frost present.

The prototype was designed during the fall of 1995 using a basic numerical tool. Drainage tubing were retained as they readily permitted one to meet the design requirements. One of the goals was to convince producers that such a simple design could spare them a substantial part of their yearly heating costs. The unit was assembled and calibrated in a greenhouse used for the experimental cultivation of hydroponic tomatoes and cucumbers during the winter of 1996. The first series of tests, carried out between March to May 1996, demonstrated that average efficiencies of η=84% and η=78% were obtainable with air volumetric exchanges rates of 0.5 and 0.9 change per hour, respectively, in a 576m^3 greenhouse. Latent heat was found to play a major role in the overall heat transfer, contributing about 40% of the total energy exchanged in some situations.

In conclusion, with sufficient exchange area, simple heat exchangers can be economically used as dehumidifiers in several applications. The encouraging results presented and mentioned here demonstrate that yet other applications could be found for heat exchangers in sustainable development strategies.

ACKNOWLEDGEMENTS

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NOMENCLATURE

COP	Coefficient of performance
c_p	Specific heat, J/kg
f	Friction factor
g	Gravitational acceleration, m/s^2
h	Heat transfer coefficient, W/m^{2}K
h_{fg}	Latent heat of vaporization, J/kg
i	Specific enthalpy, J/kg
k	Thermal conductivity, W/mK
$\dot{m}$	Mass flow rate, kg/s
p	Pressure, Pa
A	Surface area, m^2
D	Diameter, m
L	Length, m
L	Contribution of latent heat, %
N	Number of tubes along a vertical line
Nu	Nusselt number, hD/k
P	Pressure, Pa
q	Heat transfer rate, W
$\dot{Q}$	Air change/ hour
Re	Reynolds number
R''_f	Resistance due to fouling, m^{2}K/W
R_T	Thermal resistance, K/W
T	Temperature, K
U	Overall heat transfer coefficient,
$\dot{v}$	Volumetric flow rate, m^3/s
V	Velocity, m/s
W	Humidity ratio
ε	Roughness, m
η	Efficiency
μ	Dynamic viscosity, Ns/m^2
ρ	Density, kg/m^3

Subscripts

b	Bulk
c	Cold side
dry	When there is no condensation
h	Hot side
h	Hydraulic diameter
i	Internal
l	Liquid phase
mix	Air-vapor mixture
o	External
sat	At saturation
v	Vapor phase
w	Wall
wet	When condensation occurs

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Table 1: Energy requirements and costs as a function of the ventilation strategy in greenhouses.

Dehumidification Strategy	Energy Requirement (MJ/m^2)	Cost* ($/m^2)			Difference With/without ($/m^2)		
		Gas	Oil	Electricity	Gas	Oil	Electricity
None	1672	13.44	15.47	25.54	-	-	-
1 vol/h	1883	15.14	17.43	28.77	1.73	1.96	3.23
Proportional	1980	15.92	18.32	30.25	2.51	2.85	4.71

*Cost estimates based on: 37.3MJ/L@0.24$/L and 80% efficiency for natural gas
38.9MJ/L@0.27$/L and 75% efficiency for oil no. 2
3.6MJ/kW-h@0.055$/kW-h for electricity

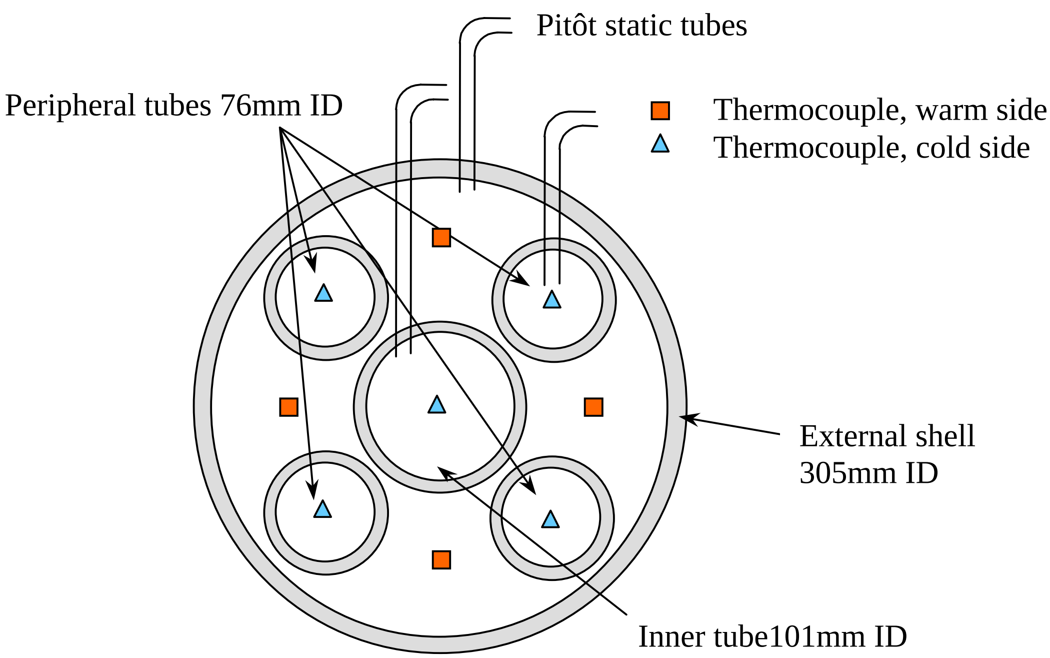

Figure 1: Schematic of the prototype cross-section

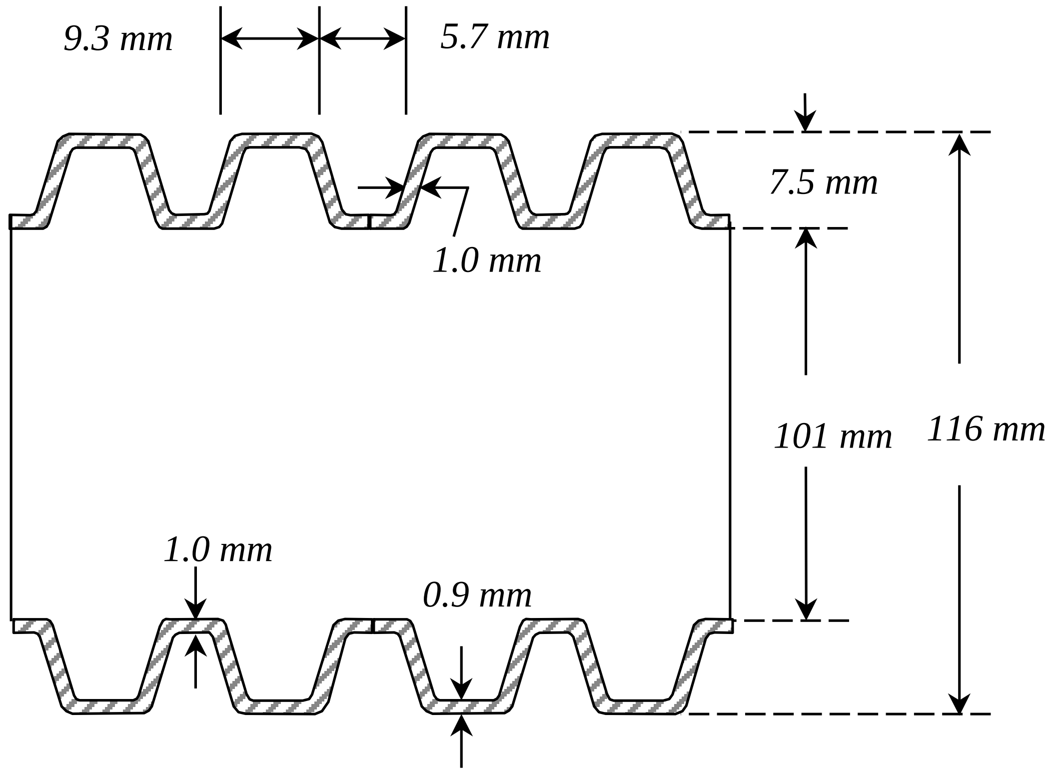

Figure 2: longitudinal cross-section and geometrical details of a 101mm I.D. tube

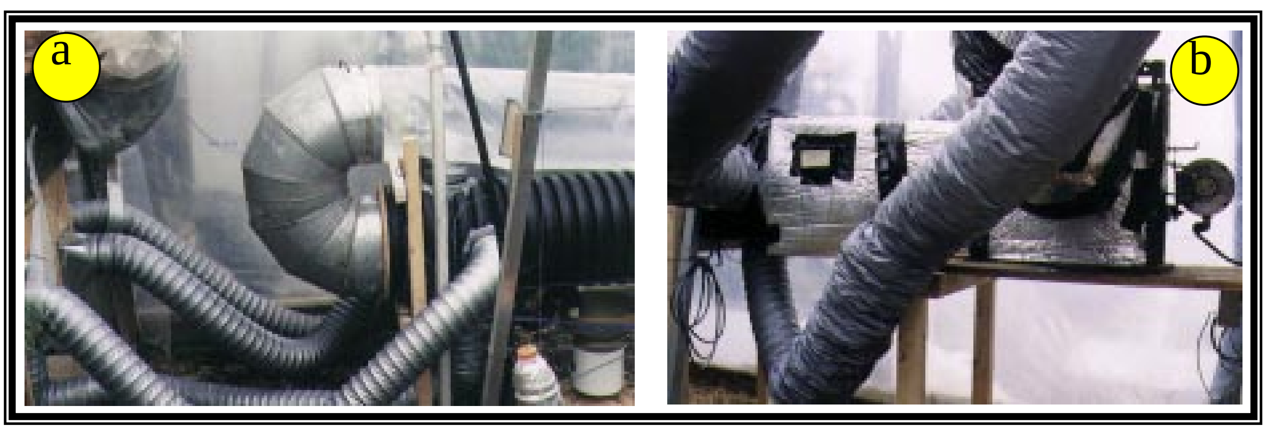

Figure 3: (a) The warm end of the unit; (b) The cold end of the unit

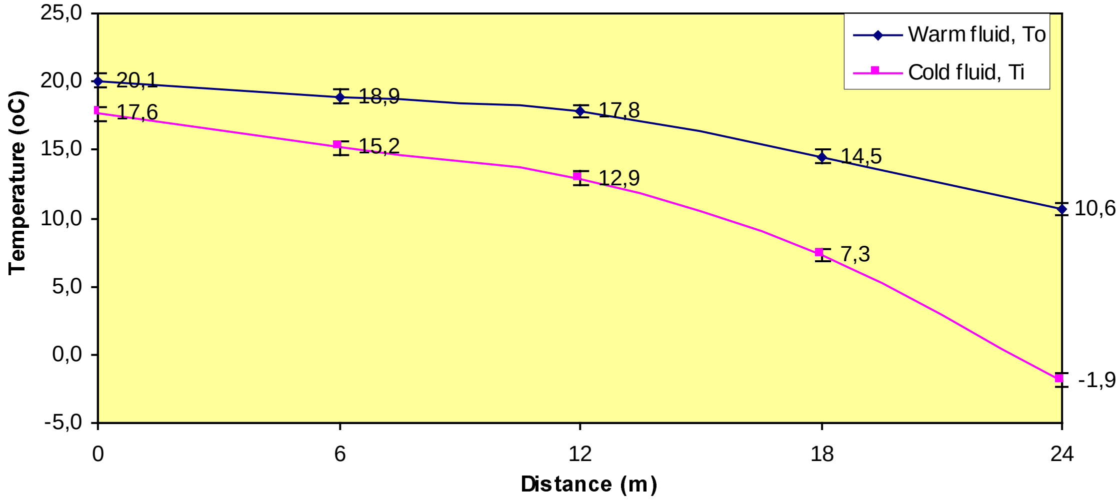

Figure 4: Temperature distribution. March 26th, 1996: 8h10, 0.5 air chg / h

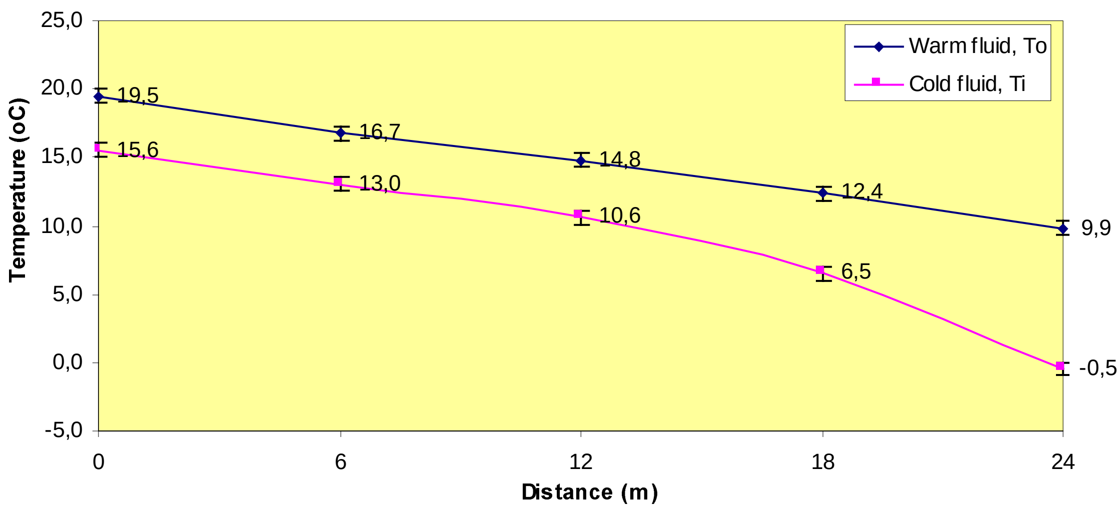

Figure 5: Temperature distribution. April 5th, 1996: 4h50, 0.9 air chg / h

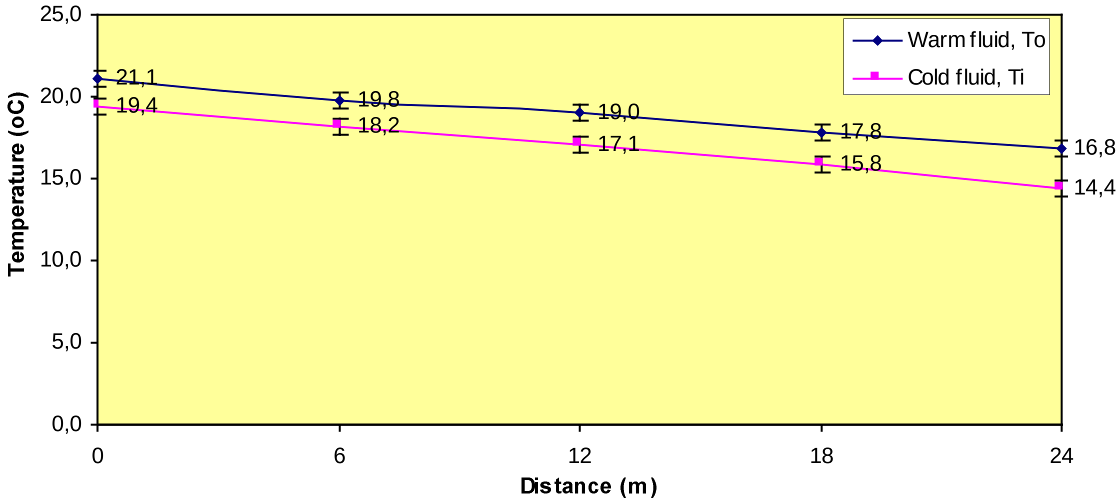

Figure 6: Temperature distribution. April 30th, 1996: 18h10, 0.9 air chg / h

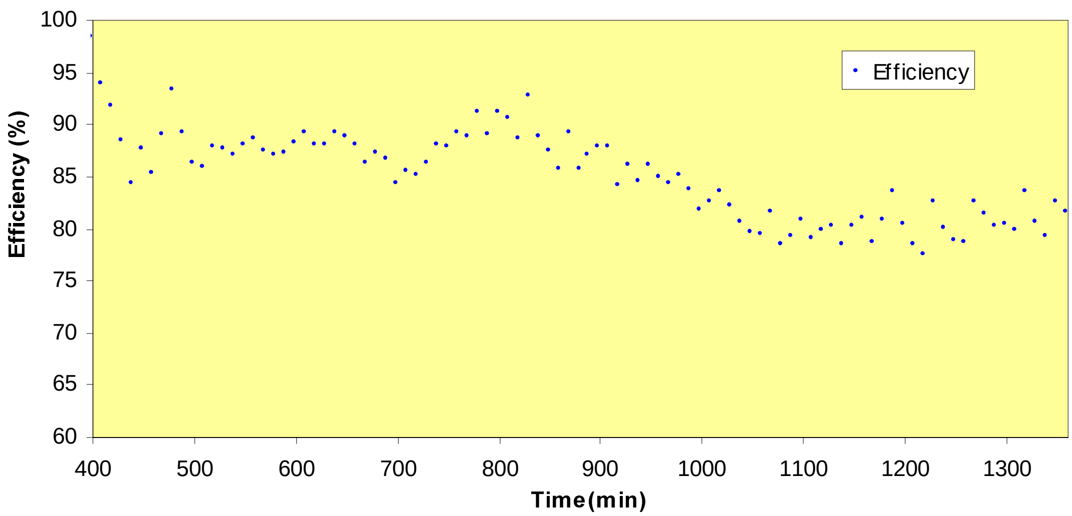

Figure 7: Variation of efficiency with time. March 26th, 1996: 0.5 air chg / h

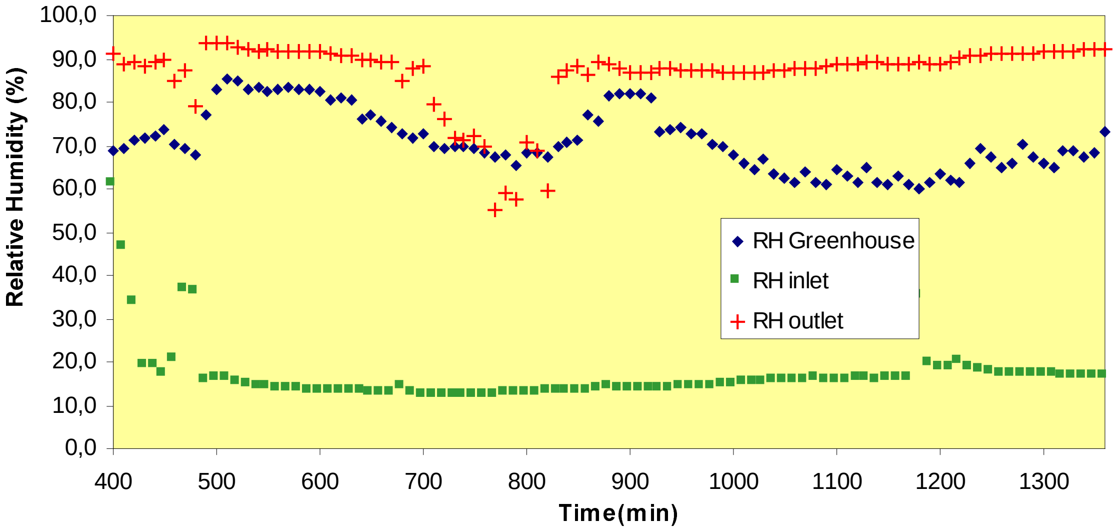

Figure 8: Relative humidity distribution in the greenhouse on March 26th 1996.

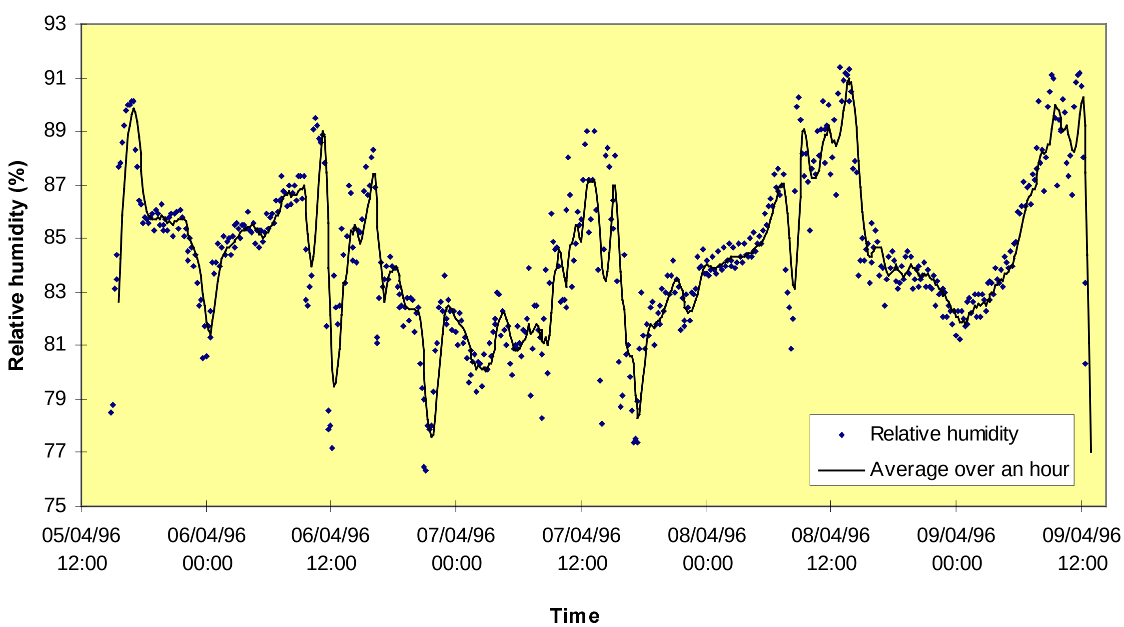

Figure 9: Relative humidity distribution in the greenhouse between April 5th and 9th 1996.

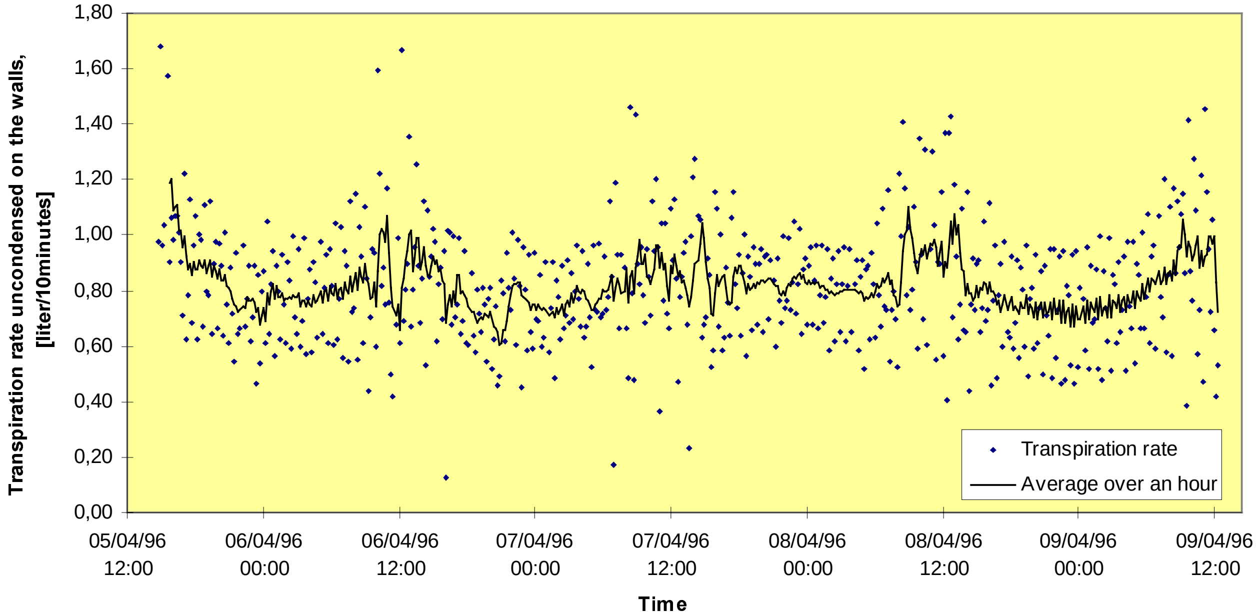

Figure 10: Rate of transpiration (not condensed on the walls) between April 5th and 9th 1996

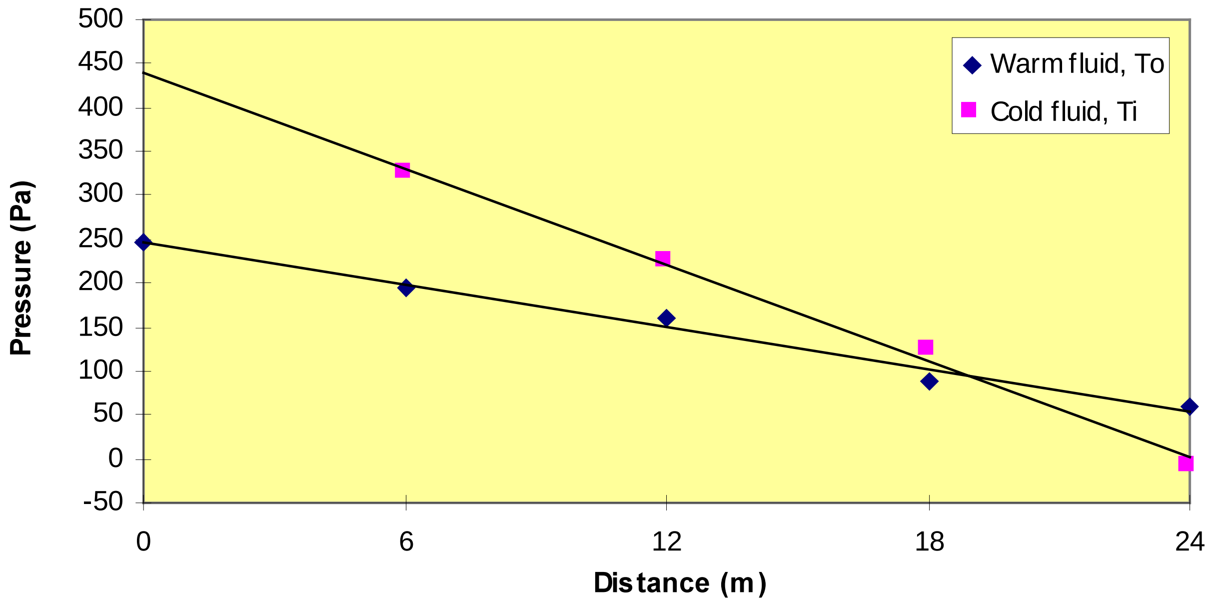

Figure 11: Pressure drops along the unit on the cold and warm sides.