Low exergy heat recovery for sustainable indoor agriculture

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Abstract: With improved greenhouses, farmers have to ventilate. An air-to-air multi-tube counter flow heat exchanger unit was installed in a greenhouse used for the experimental cultivation of hydroponic tomatoes and cucumbers. This 24m long unit involves a 12" O.D. external shell used to exhaust moist air and five inner tubes to bring fresh air inside. The tests, carried out between March and May 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 exchanger could be buried underneath the ground or suspended above the crops. The unit made of plastic is durable, rot and rust resistant, affordable, and is ice and frost compliant. A pre commercial implementation with an improved design is now considered in collaboration with Gaz Metro. This paper presents the original prototype that help in reducing the consumption of natural gas, fuel, bunker, or propane.

Keywords: *Heat exchanger, Latent heat recover, Sensible heat recovery, Plastic.*

Nomenclature

A	*Surface area*	m^2
cp	*specific heat*	$J.kg^{-1}$
f	*friction factor*	m^2
D	*diameter of the tubes*	m
h	*heat transfer coefficient*	$W.m^{-2}$
k	*thermal conductivity*	$Wm^{-1}K^{-1}$
L	*contribution of latent heat*	%
l	*length of the tubes*	m
$\dot{m}$	*mass flow rate*	$kg.s^{-1}$
Nu	*Nusselt number, hD/k*	-
Re	*Reynolds number*	-
T	*temperature*	K
i	*specific enthalpy*	$J.kg^{-1}$

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. To keep sustainable development strategies, this exchanger should be low cost, user friendly, rot and corrosion resistance, efficient even when ice and frost are present, and, obviously, save energy. The purpose of this study is to design, build, and test such an exchanger to be used in greenhouses located in Northern countries.

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. The data for unit costs are updated for 2011.

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	29,14	44,13	35,76	-	-	-
1 vol/h	1883	32,81	49,70	40,28	3,68	5,57	4,51
Proportional	1980	34,50	52,26	42,35	5,37	8,13	6,59

Cost estimates based on: 37.3MJ/m^3@0.48\$/m^3 and 80% efficiency for natural gas
38.9MJ/L@0.54\$/L and 75% efficiency for oil no.2
3.6MJ/kW-h@0.077\$/kW-h for electricity

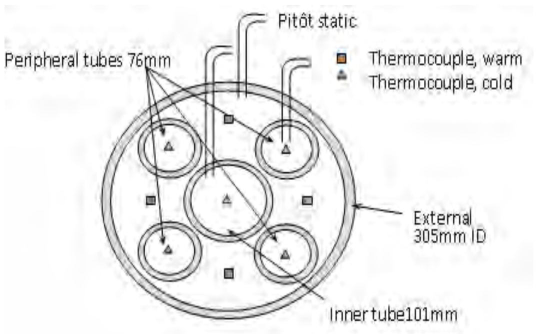
In Table 1, the first row corresponds to unit heating costs when dehumidification is due to exfiltration of moist air only (balanced by infiltration of cold air), while most of the vapour condenses on the roof and the walls of the greenhouse. This situation is mostly found in old installations where 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 5.4 (for natural gas) to 8.1 CDN\$/m^2 (for Oil, indeed electricity is cheaper than oil in Québec) per year more expensive than no ventilation. This is twice as much as in the 1990s for which this cost varied from about 2.5 (for natural gas) to 4.7 CDN\$/m^2 (for electricity). This represents a minimum extra cost of about 800\$/y for a small 144 m^2 unit which results in millions of dollars for the 110 hectares of crops and 134 hectares of ornamental plants being grown in Quebec only. Hence, one of the objectives of the work is to provide an equipment with a low payback period to be used by most farmers. At last, it should be stated that the critical periods for ventilation are fall and spring for which crops are growing and a fast rate and condensation on the walls is not as important as in winter.

2. Methodology

2.1. 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 [6] 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 [7], see Fig. 1.

Due to the unlimited amount of space available within greenhouses and because the major part of the exchanger could be buried or suspended, compactness [8] 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 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. The surface increase for the 76 mm tube is the same. Fig. 2(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. Fig. 2(b) shows the cold end of the prototype.



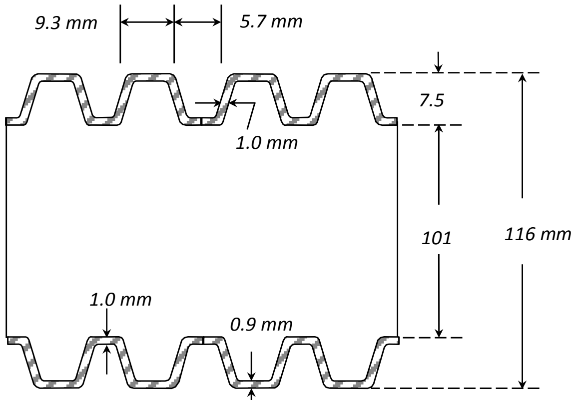


Figure 1: Schematic of the prototype: (left) cross-section; (right) longitudinal cross-section and geometrical details of the 101mm I.D. tube

It can be seen in Fig. 2b that the ventilator is built into the plenum and that the tubes are isolated to prevent condensation in the greenhouse. The overall cost of this prototype, excluding the fans, is much below 2000 CDN$.

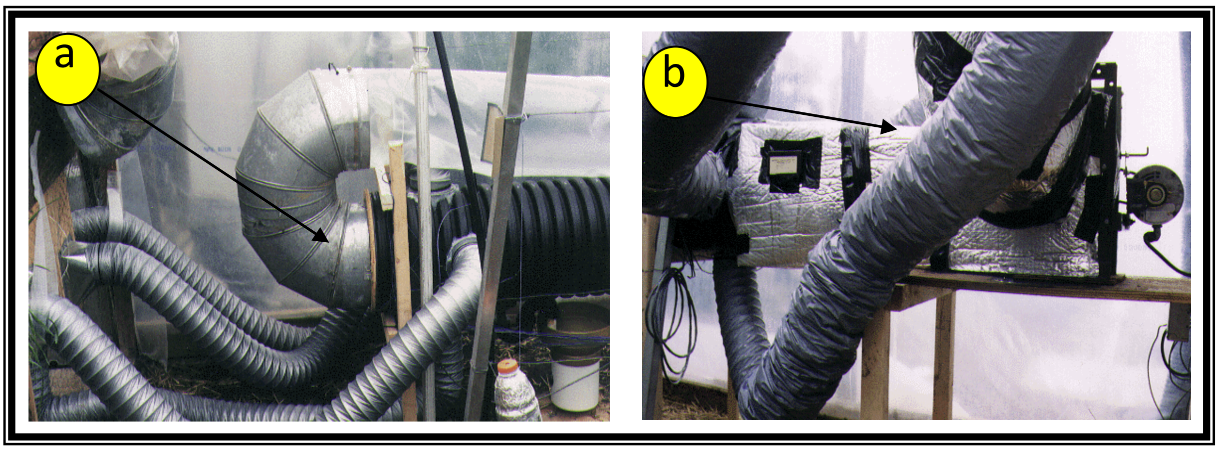


Figure 2: (a) The warm end of the unit; (b) The cold end of the unit

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 original 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.

2.2. Numerical design tool

Brundrett et al. [1] proposed a simple model to design heat exchangers to be used as dehumidifiers in greenhouses. In [1], the authors proposed to carry out energy balances along the axis of the exchanger from one volume to the next. In dry and wet zones, the overall heat transfer coefficient is calculated differently while the external kernel is assumed to be adiabatic. 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. 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. 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 [9,10] with the entrance correction factor derived by Hausen [11,12]. For the internal Nusselt number this yields:

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

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 [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 outer shell was assumed to be adiabatic. The predictions then have to include the specifications of the psychometric 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. The overall heat transfer between the hot and cold fluids is given by:

$$q = \dot{m}_o\left(i_{o,inlet} - i_{o,outlet}\right) = \dot{m}_i\left(i_{i,outlet} - i_{i,inlet}\right) \tag{2}$$

An iterative procedure is employed in the two sections until a balance is obtained in the calculation of the heat transfer with Eq.(2) and that with UA LMTD [8]. The contribution of latent heat to the total heat transfer was estimated with:

$$L = \left[1 - \frac{c_p\,(T_{o,inlet} - T_{o,outlet})}{i_{o,inlet} - i_{i,inlet}}\right]*100 \tag{3}$$

where subscript *i* refers to the stream inside the tubes and subscript *o* refers to that outside the tubes or into the kernel. The efficiency is defined as:

$$\eta = \frac{T_{o,inlet} - T_{i,inlet}}{T_{i,outlet} - T_{i,inlet}}$$

3. Results

3.1. Global results

In this section overall results are provided for the period extending from March 21st to May 21st. Spring is selected as it corresponds to a critical period as the plants are active and condensation rates on the walls very low due to higher temperatures than those found in the winter. 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. To obtain such results, the amount of condensation recovered is measured (to estimate latent heat recovery) as well as the overall temperature differences.

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°C and the internal temperature 20°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 first day was March 26th, when the volumetric flow rate of warm fluid, $\dot{V}_h$, was 0.099 m3/s and that of the cold fluid, $\dot{V}_c$, was 0.079 m3/s. The profile presented in Fig. 3 (a) is typical of what was observed when the prototype operated at 0.5 air change per hour.

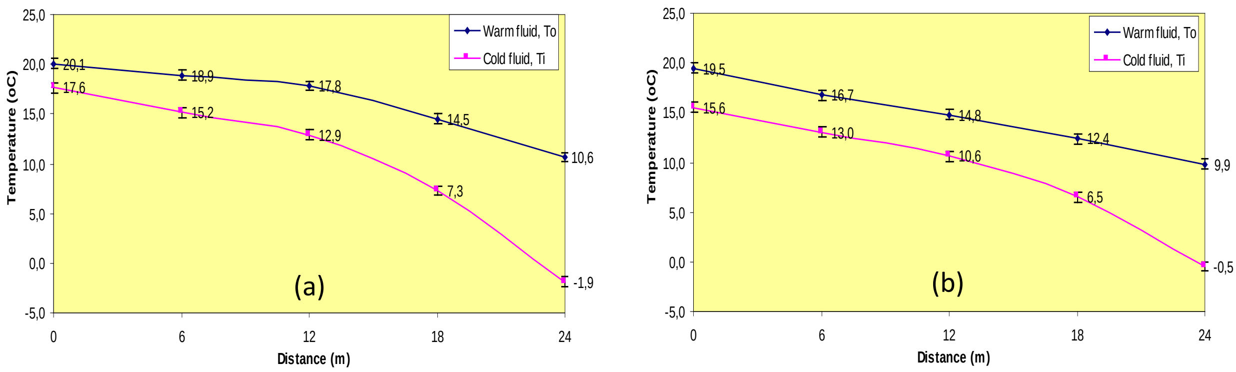


Fig. 3: Temperature distribution. (a) March 26th: 8h10, 0.5 air chg / h; (b) April 5th: 4h50, 0.9 air chg / h

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 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.

Fig. 3(b) shows results for April 5th, 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.

3.2. Psychometrics 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. Fig. 4 shows the relative humidity distribution for March 26th.

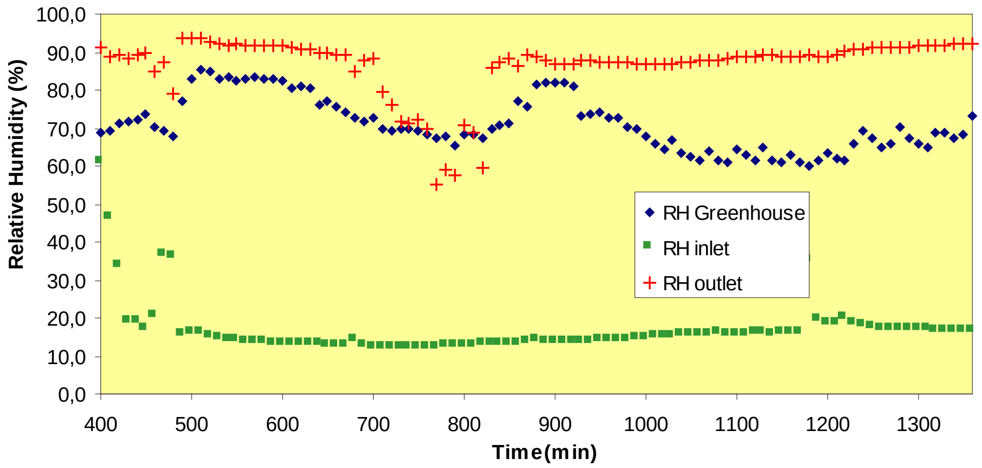


Fig. 4: Relative humidity distribution in the greenhouse on March 26th

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%.

Fig. 5 presents typical results obtained for a period ranging from April 5th to April 9th. 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.

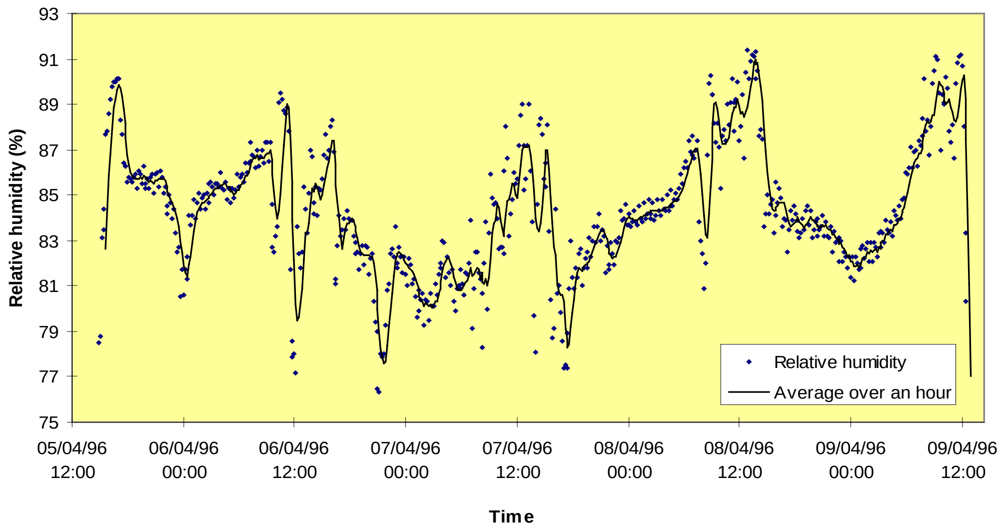


Figure 5: Relative humidity distribution in the greenhouse between April 5th and 9th

Again, it is shown in Fig.5 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 Fig.4 and 5 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.

3.3. Payback period

Here the payback period is estimated with no account for the improvement of the crops growth with adequate level of humidity: the “real” performance of the exchanger should be better. The integrated heat recovery is used to estimate the payback with no account of the fan power as if they were used anyway to extract the moisture from the greenhouse. It has been found that the units were able to recover 9840 kW-h over the whole year which corresponds to a cost of 617$ for gas heating and 935$ for oil heating. As the experimental unit costs 1140$ (calculations carried out for a production and installation of 100 per year), the simple payback period is about 1,5 year (from 1,2 to 1,9 years, without subsidy).

4. 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: (1) low cost, CDN$ < 2000 (1,5 year pay-back period); (2) ease of assembly, maintenance, repair, and operation; (3) corrosion and rottenness resistance; (4) satisfactory operating efficiency when frost present.

The prototype was designed using a basic numerical tool. Drainages 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 winter. The first series of tests, carried out between March to May, 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.

References

[1] E. Brundrett, T.J. Jewett, and R. Quist, Evaluation of polytube heat exchangers for greenhouse ventilation". *Acta-horticultarae*, 148, 1984, pp.49-55.

[2] J.N. Walker and D.J. Cotter, Condensation and resultant humidity in greenhouses durng cold weather, Trans. ASEA, 11(2), 1968, pp.263-266.

[3] D.J.Cotter and R.T. Seay, The effect of circulating air on the environment and tomato growth response in a plastic greehouse, Roc. ASHS, 77, 1961, pp.345-342.

[4] SPSQ., Ékiloserre; Projet d'amélioration de la situation énergétique de l'industrie serricole québécoise, Rapport Final, Syndicat des Producteurs en Serre du Québec, St-Hyacinthe, 1995.

[5] D. DeHalleux, and L. Gauthier, Consommation énergétique due à la déshumidification des serres au Québec, Université Laval, Québec, 1995.

[6] BNQ., Norme 3624-115 (91-08-01); Tubes annelés flexibles et raccords en thermoplastique pour le drainage des sols, Bureau de Normalisation du Québec, Québec, 1991.

[7] BNQ., Norme 3624-120 (90-02-20); Tuyaux annelés à l'intérieur lisse et raccords en plastiques Pe ou PP pour l'évacuation des eaux pluviales, Bureau de Normalisation du Québec, Québec, 1990.

[8] A. Bejan, Convection Heat Transfer, 2nd ed., Wiley, New-York, 1993

[9] V. Gnielinski, New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, Int. Chem. Eng., vol.16, 1976, pp.359-368.

[10] V. Gnielinski, Forced Convection in Ducts, in G.F.Hewitt (ed.), Handbook of Heat Exchanger Design, Begell House, NY, 1952, section 2.5.1-5.

[11] H. Hausen, Darstellung des Wärmeüberganges in Rohren durch verallgemeinerte Potenzbezeihungen, Z. Ver. Dtsch. Ing. Beiheft Verfahrenstech., vol.4, 1943, pp.91-134.

[12] H. Hausen, Heat Transfer in Counterflow, Parallel Flow and Cross Flow, McGraw-Hill, USA, 1983.