EXPERIMENTAL CHARACTERIZATION OF SENSIBLE AND LATENT HEAT EXCHANGES IN A COUNTERFLOW AIR-TO-AIR HEAT EXCHANGER

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ABSTRACT
Cold weather in Eastern Canada has incited local farmers to improve air-tightness in greenhouses in order to reduce heating costs. Low air exchange rates however, engender an increase of the inside humidity, which affects the health, growing rate and quality of the crops. The use of air-to-air heat exchangers constitutes an interesting low cost solution for greenhouse dehumidification, allowing the recovery of part of the energy used for heating while improving air quality at the same time. Tube-shell and polytube-shell counterflow heat exchangers requiring access to large areas to be effective are well-suited configurations for greenhouses where space is normally available. Cold air (from the exterior) flows in one direction inside the core as is heated by the warm (and humid) air (from the inside) flowing in the opposite direction through the shell side. The purpose of this work is to characterise the heat transfer in the cold (sensible) and warm (sensible and latent) sides of a simplified tube-shell experimental heat exchanger having a core composed of one high-density polyethylene (HDPE) corrugated pipe and an external Plexiglas shell, under close to typical *in situ* internal conditions.

1. INTRODUCTION

This study has been conducted as a part of a Project sponsored by two Canadian organisms: the MAPAQ[1] and the CIDES[2], to find a solution to the humidity excess problem in greenhouses [1]. Different studies have demonstrated that it is possible to effectively build low cost heat exchangers to be used in greenhouse dehumidification [2, 3]. Based on this investigations, the design and construction of a first prototype has been accomplished [4, 5] whose core was formed by 1 central 101 mm I.D. tube wrapped around 4 tubes of 76 mm I.D. Such tubes, made from corrugated plastic, are typically used for drainage.

Having very encouraging results from a first sequence of tests, a second generation of prototypes with the core composed of 45 tubes of 50 mm I.D., has been constructed and tested. This time, in addition to greenhouses, a new agricultural environment has been studied: hen-houses. A third generation of prototypes composed of 150 tubes of 38 mm I.D. with baffles it's presently under investigation.

Summarizing, the prototypes are still evolving, for this reason it is of paramount importance to count on a reliable designing tool. The computer program used until now for heat transfer simulations is based on empirical correlations that only approaches the actual phenomena.

For the continuation of the research, the derivation of more suitable convective heat transfer correlations is rrequired for both sides of the prototypes. To complete this task, the thermal performance of the heat exchangers has to be more accurately described.

[1] Ministère de l'Agriculture, des Pêcheries et de l'Alimentation du Québec
[2] Centre d'Information et de Développement Expérimental en Serriculture

2. CONFIGURATION OF THE PROTOTYPES

The configuration of the prototypes is schematized in Figure 1. As it can be seen from this graphic, cold air T_c flows inside the tubes and it is heated as it passes through the heat exchanger, from x_1 to x_2. Heat transfer is transferred without phase change inside the tubes since the internal tube wall temperature remains always above the inside air's saturation temperature.

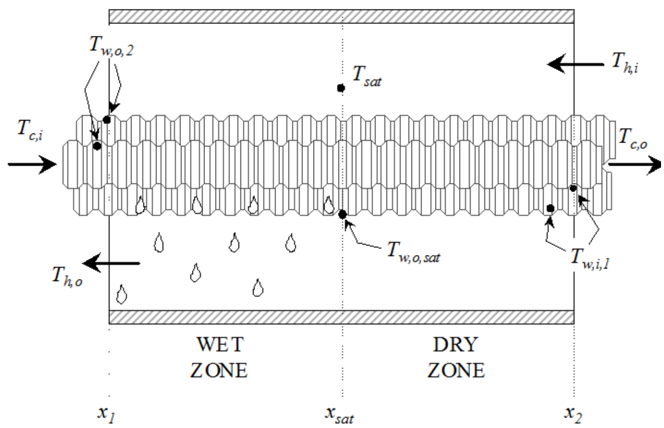

Figure 1. Thermal configuration of the prototypes.

On the contrary, the hot air T_h moving through the shell-side, *i.e.* outside the tubes, has normally a high humidity level. The external tube wall temperature T_w can be below warm air saturation temperature T_{sat} and a phase change takes place. For typical conditions, condensation is produced on x_1, corresponding to the cold air entrance (and warm air exit), where the temperature gradient is at its highest level. Conversely, there is less or no phase change at all in x_2, where the temperature gradient is lower.

There is a point between this two limits denoted as x_{sat} in Figure 1, where the external wall temperature equals the saturation temperature: $T_{w,o}=T_{sat}$. The heat exchanger can then be divided in two sections: a *dry zone* and a *wet zone*.

The total energy transfer in the wet zone is the result of the combination of two contributions: *sensible heat*, resulting from a temperature gradient; and *latent heat*, which results from a phase change.

With the purpose of derive suitable correlations to characterize both sides of the prototypes, a simplified tube-shell experimental heat exchanger was constructed and tested as discussed next.

3. EXPERIMENTAL APPARATUS

The different parts of the experimental apparatus together with the air circulation circuits are shown in Figure 2.

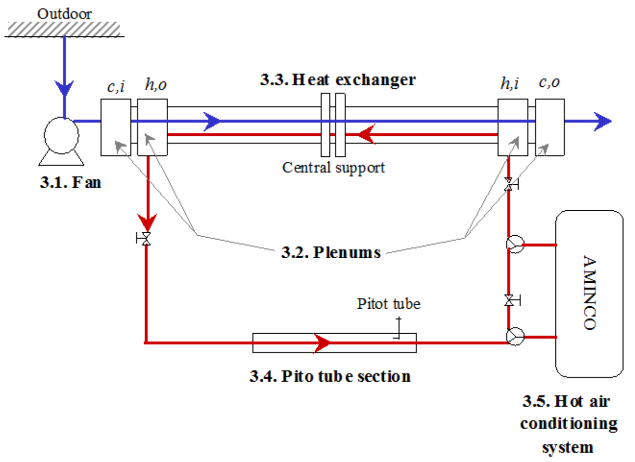

Figure 2. Experimental apparatus.

3.1 Fan

Cold air from outside the building is supplied to the heat exchanger with help of a fan. Air is first passed through a plenum to stabilize the flow.

3.2 Plenums

Two plenums were installed to stabilize the flows, one in cold air input (*c,i*), the other at the hot air input (*w,i*). Two more plenums were located on the exit of both flows, (*c,o*) and (*w,i*), before entering the Pitot tube section where air velocity was measured.

3.3 Heat exchanger

To reduce the interference of the probes with both flows and wall measurements, only one corrugated plastic 101 mm I.D. tube, similar to those used on the heat exchanger prototypes, has been chosen to form the core. The external shell has been constructed with a 30 cm O.D. acrylic transparent tube to allow visual inspection throughout the tests.

3.4 Pitot tube section

A Pitot tube has been incorporated to the setup in conformity with the American Society for Testing and Materials standard [7] concerning the measurement of the average velocity in a duct. The dynamic pressure was measured by an pressure transducer (OMEGA PX 653).

3.5 Hot air conditioning system

To simulate the conditions normally present inside greenhouses, hot and humid air was provided to the heat exchanger by a temperature and humidity-controlled cabinet (AMINCO-AIRE) in which different atmospheric conditions can be reproduced.

3.6 Instrumentation

Thirteen temperature monitoring stations has been installed at every 25 cm in the heat exchanger to determine the temperature profiles. Relative humidity has been recorded in each and every plenum with a humidity probe (VAISALA HMD7OU/Y). Data acquisition has been carried out using relay multiplexers (Campbell Scientific: AM416 and AM32).

Table 1 presents the different parameters of interest as well as the operation limits, the measured errors and the number of measurements corresponding to each parameter.

Table 1. Experimental parameters and related errors.

Parameter	Limits	Error	Measurements
T_{ext}	-20 a 10°C	-	1
T_f	-11,5 a 10°C	±0,30°C	39
T_c	20 a 26°C	±0,34°C	39
$T_{p,c}$	-1,0 a 15°C	±0,32°C	34
ϕ_c	30 - 90%	±0,0035 %	6
$\dot{V}_c$	30 - 75 L/s	±0,819 L/s	1
TOTAL			**120**

4. COLD SIDE

Heat exchangers analysis requires the knowledge of the individual thermal resistances involved on the global heat transfer. The average heat transfer coefficients $\bar{h}$, are required to calculate the convective resistances in both sides of the heat exchanger. For the cold side, the procedure to follow resumes in obtaining the heat transfer coefficients directly from Newton's Law of Cooling, and then to compare the results with the predictions of the proper empiric correlations.

4.1 Evaluation of the heat transfer coefficient

Local heat transfer coefficients can be obtained directly from Newton's Law of Cooling, as follows:

$$h_c = \frac{q''_{w,c}}{\left(T_{w,c} - T_c\right)} \tag{1}$$

The heat flux $q''_{w,c}$, can be calculated from an energy balance using First Law of Thermodynamics:

$$q''_{w,c} = \frac{\rho_c V_c c_{P,c}}{\pi D_c}\left(\frac{dT_c}{dx}\right) \tag{2}$$

The slope dT_c/dx can be evaluated by least square regression, using cold side temperature $T_c(x)$ experimental data [3].

Since wall temperature is measured only outside the tube, an expression linking outside and inside wall temperatures, $T_{w,c}$ and $T_{w,h}$, can be derived from Fourier's Law of Conduction [7]:

$$T_{w,c} = T_{w,h} - \frac{r_c q''_{w,c}}{k_w} \ln\left(\frac{r_h}{r_c}\right) \tag{3}$$

4.2 Empiric correlations

Table 2 shows some of the most common empirical correlations for both smooth and rough tubes.

Table 2. Empirical correlations for internal turbulent flow.

	Author	Correlation
Smooth tubes	Colburn, 1933	$\overline{Nu} = 0{,}023 Re^{4/5}\,\mathrm{Pr}^{1/3}$
	Sieder y Tate, 1936	$\overline{Nu} = 0{,}027 Re^{4/5}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_p}\right)^{0{,}14}$
	Gnielinski 1976	$\overline{Nu} = \frac{\left(f/2\right)\left(Re - 10^3\right)\mathrm{Pr}}{1 + 12{,}7\sqrt{f/2}\left(\mathrm{Pr}^{2/3} - 1\right)}$
Rough	Dipprey y Sabersky, 1963	$\overline{St} = \frac{f/8}{1 + \left(f/8\right)^{1/2}\left[5{,}19 Re_k^{0{,}2}\,\mathrm{Pr}^{0{,}44} - 8{,}48\right]}$
	Bhatti et Shah,1988	$\overline{St} = \frac{f/8}{1 + \left(f/8\right)^{1/2}\left[4{,}5 Re_k^{0{,}2}\,\mathrm{Pr}^{0{,}5} - 8{,}48\right]}$

A previous analysis from the results [3] validated the assumption of fully developed flow in the experimental apparatus. This allows to determine the average heat transfer coefficients by integration of the local values obtained from equation (1).

Figure 3 shows the experimental average heat transfer coefficients ("x") and the predictions (continuous lines) estimated from the correlations of Table 2.

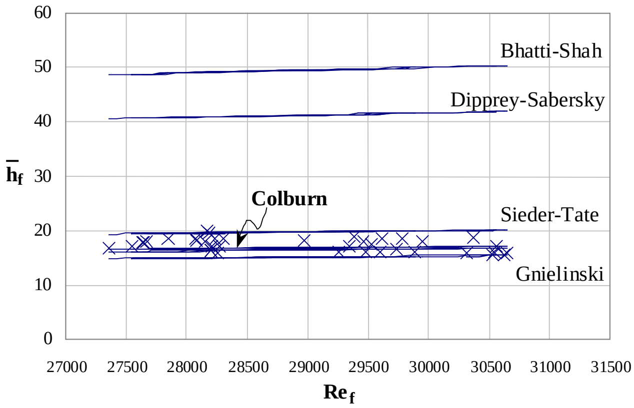

Figure 3. Experimental ("x") vs predicted (continuous lines) heat transfer coefficients.

As can be seen from this graphic, the predictions obtained from empirical correlations typically used for smooth tubes are closer to the experimental results than those obtained from rough tubes correlations. This is in agreement with the observations of Sibley and Raghavan [8], which used the standard absolute deviation to compare their result with the predictions of Dittus-Boelter equation and Sieder-Tate equation for smooth tubes, and Reynolds equation for rough tubes.

The average absolute deviation or AAD, can be calculated from the following relation :

$$AAD = \frac{\sum_{i=1}^{n}\left(h_{emp} - h_{\exp}\right)}{n} \tag{4}$$

where h_{emp} is the heat transfer coefficient calculated from empirical correlations, h_{exp} is the experimental heat transfer coefficient and n is the total number of tests.

Table 3 shows a comparison of AAD values the empirical correlations of Table 2. The average diameter $D_c=(D_{max}+D_{min})/2$, and a correction factor was employed to calculate the total heat exchange surface as described in [3].

Table 3. Experimental vs empirical AAD results.

Author	*AAD*
Colburn	0,49
Sieder-Tate	2,39
Gnielinski	2,19
Dipprey-Sabersky	23,87
Bhatti-Shah	32,16

As it can be seen from this table, in the case of a 101 mm I.D. corrugated plastic tube, the best results are obtained with Colburn's equation. Hence, it can be concluded that, The net impact of using 101 mm I.D. corrugated tubes is to increase the heat transfer area. The area for a 101 mm I.D. corrugated tube is almost twice the area for a non-corrugated tube of equivalent diameter.

It should be pointed out though, that this situation could be different for corrugated HDPE tubes of smaller size that are currently investigated for polytube-shell configurations, e.g. 76, 50 and 38 mm.

5. HOT SIDE

Heat transfer characterization of the shell side is a much more complicated problem than the tube side presented above, mainly because of the partial condensation occurring in the external wall. In this case, the procedure to follow involves the derivation of a suitable correlation from the experimental data.

There are two correlation methods suitable for latent heat quantification, these are: the Threlkeld Method [9] and the Wilson Plot Technique [10]. Threlked Method needs the determination of the point where phase change begins, that is, where the wall temperature equals the saturation temperature of the hot air. This is not always an easy task in practical applications, particularly in tube banks. However, the simplified single-tube core configuration used herein allows to apply this technique.

Alternatively, the Wilson Plot Technique is an interesting method that are suitable for complex configurations such as tube banks. However, the mathematical forms of the solution to characterize the tube and shell sides of the heat exchanger must be known, which unfortunately is not the case for the hot side of our heat exchanger.

Both methods have been implemented to the present work. Solutions were obtained first from the experimental data using Threlkeld Method, which allows developing an experimental correlation for the hot side to be used in a modified Wilson Plot Technique.

5.1 Threlkeld Method

Dry heat exchanger

Dry heat exchanger analysis is well documented in the literature [8-10]. The procedure begins with the well-known overall heat transfer resistance expressed as a sum of individual resistances: $R_{overall}=R_c+R_{eq}+R_h$ where R_{eq} represents the sum of the non-convective resistances, as wall conduction and fouling, which are generally obtained independently.

In terms of the convective heat transfer coefficients and neglecting any thermal resistance besides wall conduction R_w, for the equivalent resistance R_{eq}, the overall heat transfer resistance can be estimated as follows:

$$\frac{1}{UA_h}=\frac{1}{\bar{h}_c A_c}+\frac{r_c \ln(r_h/r_c)}{A_c k_w}+\frac{1}{\bar{h}_h A_h} \quad (5)$$

It has been stated that for corrugated plastic tubes, Colburn equation can be used in heat transfer calculations. The cold side convective resistance R_f, is then estimated.

The overall heat transfer coefficient U, can be determined from the equation:

$$q=UA_h\Delta T_{LM} \quad (6)$$

where the heat transfer rate q can be estimated from an energy balance in the cold side of the heat exchanger.

The logarithmic mean temperature difference ΔT_{LM}, for a countercurrent heat exchanger can be obtained from the following expression:

$$\Delta T_{LM}=\frac{(T_{h,i}-T_{c,o})-(T_{h,o}-T_{c,i})}{\ln\left(\dfrac{T_{h,i}-T_{c,o}}{T_{h,o}-T_{c,i}}\right)} \quad (7)$$

The average hot side heat transfer coefficient can be derived from equations (5) to (7):

$$\bar{h}_h=A_h\left(\frac{1}{UA_h}-R_w-R_c\right)^{-1} \quad (8)$$

Wet heat exchanger

In presence of a phase change, Threlkeld proposes a relation similar to equation (5), using an overall enthalpy resistance: $R_{i,total}=R_{i,c}+R_{i,eq}+R_{i,h}$, which allows to take into account the contribution of the latent heat in the total heat transfer.

In terms of the transfer coefficients and neglecting enthalpy resistances any other than wall conduction:

$$\frac{1}{U_i A_h}=\frac{b_c}{\bar{h}_c A_c}+\frac{b_w r_c \ln(r_h/r_c)}{A_c k_w}+\frac{1}{\bar{h}_M A_h} \quad (9)$$

There are three important points to note from equation (9): (1) U_i is the overall enthalpy heat transfer coefficient which is analogue to U for sensible heat transfer. However, the units of U_i and U are different; (2) for temperature intervals of approximately 5°C, humid air saturation enthalpy can be consider to vary linearly: $i=a+bT$. Threlkeld propose to use a fictitious enthalpy i', to calculate the constants b_c and b_w appearing in equation (9), using dry bulb temperature and assuming that the air flow is saturated; and (3) since latent heat contributes to the overall heat transfer in a condensing heat exchanger, the mass transfer coefficient $\bar{h}_M$, appearing in equation (9), is used for overall enthalpy coefficient U_i, instead of a heat transfer coefficient $\bar{h}_h$, which accounts only for sensible heat transfer.

The total heat transfer on a wet heat exchanger can be calculated from:

$$q=U_i A_h \Delta i_{LM} \quad (10)$$

where Δi_{LM} is the logarithmic mean enthalpy difference, which, for a counterflow heat exchanger, can be estimated as:

$$\Delta T_{LM}=\frac{(i_{h,i}-i'_{c,o})-(i_{h,o}-i'_{c,i})}{\ln\left(\dfrac{i_{h,i}-i'_{c,o}}{i_{h,o}-i'_{c,i}}\right)} \quad (11)$$

The mass transfer coefficient can be calculated as:

$$\bar{h}_M=A_h\left[\frac{1}{U_i A_h}-\frac{b_c}{\bar{h}_c A_c}-\frac{b_w r_c \ln(r_h/r_c)}{A_c k_w}\right]^{-1} \quad (12)$$

Sensible heat exchanges on a wet heat exchanger can be calculated from equations (5) to (8).

Data correlation

The Colburn analogy stress underlines the existent similarities between heat and mass transfer phenomena:

$$j=St\,Pr^{2/3}=St_M\,Sc^{2/3} \quad (13)$$

The j factor which appears in this relation is in fact a variation to the heat transfer coefficient as can be obtained from the Stanton number definition: $St=Nu/RePr=h/u\rho c_P$.

Adopting the traditional form of the forced convection heat transfer solution: $Nu=CRe^m Pr^{1/3}$, combined with equation (13), the Colburn heat transfer factor j_T can be calculated as:

$$j_T=CRe^{m-1} \quad (14)$$

In a similar way, from the mass Stanton number definition: $St_M=Sh/ReSc$, and assuming a solution of the form: $Sh=C_M Re^n Sc^{1/3}$, the masse Colburn factor j_M is calculate as:

$$j_M=C_M\,Re^{n-1} \quad (15)$$

A relation relating the heat and mass transfer coefficients can be derived from Colburn's analogy:

$$h_M=\frac{h_c}{\rho c_{P,c} Le^{2/3}} \quad (16)$$

where Le is the Lewis number: $Le=Sc/Pr$, which for the case of humid air at atmospheric pressure $Le^{2/3}=(0{,}6/0{,}7)^{2/3}\sim 1$.

In normal operation conditions, the heat exchangers present a dry zone and a wet zone. Hence, the complete heat transfer solution consists first, in the determination of the start point of the condensation x_{sat}, then, the heat exchanger is divided in two zones with different thermal characteristics : a dry zone, where heat transfer is the result of sensible heat transmission, and a wet zone, where the total energy transmission is the result of sensible and latent heat transfer.

The heat exchange surface used in the calculations must be estimated according to the zone of the heat exchanger, that is: $A_h=A_{dry}=\pi D(L-x_{sat})$ for the dry zone, and $A_h=A_{wet}=\pi D(x_{sat})$ for the wet zone.

The complete solution requires then of three correlations: one for the sensible heat on the dry zone, one for the sensible heat on the wet zone and one for the latent heat on the wet zone. Representative results for ϕ_c =50% are :

$$j_{T,\sec}=0{,}5538Re_c^{-0{,}4992} \qquad R= 0{,}9273$$

$$j_{T,hum}=1{,}1567Re_c^{-0{,}5813} \qquad R= 0{,}9214$$

$$j_M = 1{,}544 \times 10^{-3} Re_c^{-0{,}6112} \qquad R= 0{,}7488$$

Similar results, were obtained for relative humidity ranging form 35 to 80 % and are available in [3].

5.2 Data correlation as a function of the relative humidity

The application of Threlkeld Method is complex, especially for polytube configurations, due to the fact that a wet and dry zones have to be idntified. Nevertheless, the procedure presented above allowed to highlight the importance of the relative humidity. It is therefore suggested that this parameter should be included into the heat transfer solution, using an experimental correlation of the form :

$$\overline{Nu}_c = C Re_c^m \phi_c^n \tag{17}$$

Constants C, m y n are obtained following the procedure described by Holman [11]. Figure 4 presents the resulting correlation.

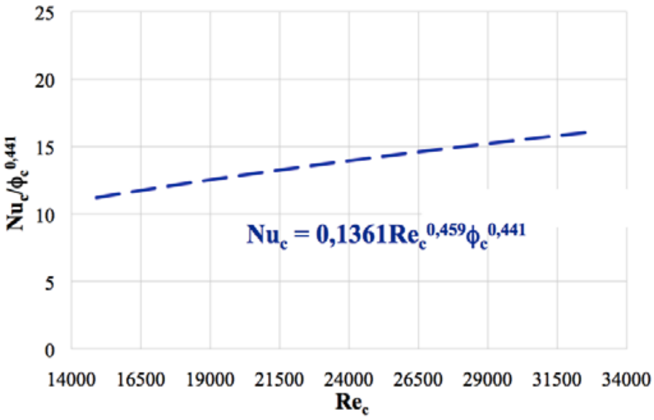

Figure 4. Experimental data correlation as a relative humidity function.

$$\overline{Nu}_c = 0{,}1361 Re_c^{0{,}459} \phi_c^{0{,}441} \qquad R=0{,}8859 \tag{18}$$

This equation is proposed to evaluate heat transfer coefficients in a corrugated tube-shell counterflow air-to-air heat exchanger with partial condensation.

Constants C, m y n in equation (17) heed to be adjusted for different configurations. For tube banks for example, wall temperature is particularly difficult to evaluate without disturbing somehow one of the two flows. A simplified method can be implemented by the way of the Wilson Plot Technique.

5.3 Wilson Plot Technique

In 1915, Wilson proposed a condenser analysis method for the cases where wall temperature was not available or where it was difficult to obtain. Briggs and Young [10], among many other researchers, proposed some modifications to the original Wilson Plot Technique to consider a wider variety of situations.

The modified technique start from the well-known global heat transfer equation for a heat exchanger:

$$\frac{1}{UA_c} = \frac{1}{\overline{h}_f A_f} + R_{eq} + \frac{1}{\overline{h}_c A_c} \tag{19}$$

where R_{eq} is the sum of all the non-convective thermal resistances participating in the system (material conduction, fouling, etc), which generally may be obtained by independent methods.

Equation (19) can be modified to obtain a linear relationship. This can be made multiplying both sides of the equation by the product $h_f A_f$:

$$\underbrace{\left(\frac{1}{UA_c} - R_{eq}\right)\overline{h}_f A_f}_{y} = \underbrace{\frac{1}{C_f}}_{a} + \underbrace{\frac{1}{C_c}}_{b}\underbrace{\left(\frac{\overline{h}_f A_f}{\overline{h}_c A_c}\right)}_{x} \tag{20}$$

It has been stated that Colburn correlation properly characterizes the internal heat transfer in corrugated plastic tubes. Colburn's correlation for the cold side, and equation (18), derived from the experimental data for the warm side, and can be used in equation (20),:

$$\left(\frac{1}{UA_c} - R_{eq}\right)\left(0{,}023 Re_c^{4/5}\,\mathrm{Pr}^{1/3}\right)\frac{A_f k_f}{D_f} = \tag{21}$$

$$\frac{1}{C_f} + \frac{1}{C_c}\frac{k_f}{k_c}\left(\frac{0{,}023 Re_c^{4/5}\,\mathrm{Pr}^{1/3}}{0{,}1361 Re_c^{0{,}459}\phi_c^{0{,}441}}\right)$$

This arrangement takes into account flow and temperature variations in both sides of the heat exchanger without the need of the wall temperature. The problem is then reduced to the calculation of the numeric values of the constants C_f, C_c y m, which varies with the configuration and the operating conditions of each particular case.

6. CONCLUSIONS

An experimental setup has been designed and constructed with the purpose of characterizing both sides of a polytube-shell counter-flow air-to-air heat exchangers for agricultural building dehumidification. Corrugated HDPE pipes are used to compose the core, for which the number of tubes and their dimensions are still under development. For this experimental study, a single corrugated tube (101 mm I. D) is used for the core, surrounded by an acrylic shell (300 mm I.D.).

It was determined that the internal flow for the 101 mm I.D. corrugated tube tubes was best characterized by empirical correlations for smooth tubes, with Colburn correlation showing the best results. The main effect of the corrugations is to increase the exchange surface. The situation could be different for tubes with smaller diameters (e.g. 76, 50 and 38 mm I.D.), for which the corrugations might further promote turbulence in the cold side.

The warm side case is more complicated since latent heat must also be considered. Experimental data was correlated as a function of Reynolds number and the relative humidity ϕ_c. Furthermore, in order to take into consideration the case of polytube core heat exchangers, a modified Wilson Plot Technique was proposed, which not longer require the determination of the wall temperature.

7. NOMENCLATURE

A	surface, m^2
C	heat capacity ,W/K
c_P	specific heat at constant pressure, J/kgK
D	diameter, m
f	Moody or Darcy friction factor = $-2\Delta PD/\rho u^2$
h	convection heat transfer coefficient, W/m^2K
k	thermal conductivity, W/mK
L	length, m
Nu	Nusselt number = hD/k
q''	heat flux density, W/m^2
r	radio, m
Pr	Prandtl number = $\nu/\alpha = c_P\mu/k$
R	thermal resistance, K/W
Re	Reynolds number = $uD\rho/\mu$
T	temperature, K, ^oC
U	overall heat transfer coefficient, W/m^2K
$\dot{V}$	volume flow, m^3/s

Greek symbols

ϕ	relative humidity
μ	dynamic viscosity , kg/sm
ν	cinematic viscosity, m^2/s
ρ	density, kg/m^3

Subindex

c	cold
i	enter
h	pared
sat	saturation
o	exit
w	wall

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