COMPARATIVE ANALYSIS OF THREE HEATING CIRCUITS FOR BASEBOARD CONVECTORS

Paulo Roberto Wander
University of Caxias do Sul – Department of Mechanical Engineering
Rua Francisco Getulio Vargas, 1130 – 95001-970 – Caxias do Sul - RS – Brazil
E-mail: prwander@ucs.br

Carlos Roberto Altafini
University of Caxias do Sul – Department of Mechanical Engineering
Rua Francisco Getulio Vargas, 1130 – 95001-970 – Caxias do Sul - RS – Brazil
E-mail: craltafi@ucs.br

Abstract *– This paper describes a comparative analysis of three different heating circuits applied to baseboard convectors with aim at assessing the assembly configuration that brings out the highest heat exchange rate. The heating systems studied are constituted of convectors installed near the floor of the room. The systems operate with water flowing inside the tubes, impelled by a centrifugal pump, while the air flows through the outside finned area by free convection. Volumetric flow and inlet and outlet temperatures of the water heater were measured to obtain data to the workbench study. Basically, temperature and flow sensors were used in the tests: PT-100 sensors with accuracy of 0.1%, i.e., that represents an uncertainty nearly 0.14°C; and digital flowmeter with accuracy of 0.5%, or an uncertainty nearly 0.5 liters/min. Water mean temperature at convector inlet was 69.7°C, and the volumetric flow measurements showed a linear relation between this parameter and the angular speed of the pump. Three configurations with two convectors each were experimentally studied and were realized in an air conditioned room, whose temperature was maintained between 18.9° and 20.0°C. Each convector is 2.380 mm long, 115 mm high, and 80 mm deep, and is composed by two tubes of cooper with external diameter of 22.23 mm, and rectangular fins, 11 mm spaced. The configuration identified as A, i.e., with the four tubes of the two convectors working in series, presented a greater heat exchange rate, overcoming in 25% that of the configuration B (two tubes working in series and these in parallel with other two), and 35% that of the configuration C (two convectors working in parallel). The results showed that the heat exchange rate presents a tendency of increasing with volumetric flow, although not in the same proportions. This fact occurred due to the presence of a dominant external convective thermal resistance, 1/heAe, relatively to the other resistances. Because of the comparative aspect of this assessment, the conduction resistance and the effectiveness of the extended surfaces weren't considered in the calculus.*

Keywords: *baseboard convectors, heat transfer, dominant thermal resistance.*

1. Introduction

Baseboard convectors are used in winter in order to achieve thermal comfort in the ambient, usually residences, offices and vehicles. It's a very simple device that uses one or two tubes and fins at the outside to increase the area of heat transfer and drive the heated air upwards. In the south of Brazil, the need for heating in the winter is increasing mainly for public areas. There are some types of heating systems offered like the one studied here. With this type, some suppliers offer different circuit configurations and each one claims to have the best one. The purpose of this study is to compare the circuits to help users to make their choice.

According to Recknagel and Sprenger (1972) there are some requirements that a heating system must attend:

- the perceived temperature in the room (the average between air and wall temperatures) must be as uniform as possible while the heating system is in operation. This temperature should not be less than 20°C nor greater than 23°C, with a variation of ± 1°C;
- the system must have a possibility of regulation, with a low inertia system, therefore, showing a quick answer;
- the heating system should not contaminate the ambient air nor produce dust, toxic gases or vapors, nor also generate excessive noise or air currents;
- the heaters should be easy to clean; and
- the heating system must be economical, considering installation, maintenance and operation costs.

A heating system using hot water is able to fulfill many of these requirements. The system is composed by a pump and a source of heated water which passes through the heat exchanger heating the ambient air. With the aid of the pump, internal heat transfer coefficient is increased, but usually the outside air flows by natural convection and, therefore, doesn't have a good heat transfer coefficient. Since the water temperature is relatively high, radiative heat transfer plays an important role in the heating capacity.

2. Heat Transfer in Heating Systems

As already mentioned, baseboard convectors are systems used to warm up the ambient air having a tube in which passes the hot water, and fins, in the form of a vertical plate, at the outside that transfer heat to the air. The heated air flows upwards and is replaced by cool air from the ambient at the bottom of the heater. In a similar device reported by Pettersson and Stënstrom (2000), the energy balance for a heated tube deals with radiation, conduction and convection following Eq. (1):

$$\dot{m}_{water} \cdot \frac{d}{dx}\left(C_{p,water} \cdot T_f\right) = \dot{Q}_{rad} + \frac{\left(T_f - T_\infty\right)}{R_{tot}} \tag{1}$$

where: $\dot{m}_{water}$ is the mass flow of water [kg.s^{-1}], $C_{p,water}$ is the specific heat of water [J.kg^{-1}.K^{-1}], T_f is the fluid temperature [K], T_∞ is the ambient temperature [K], Q_{rad} is the heat transferred by radiation [W], and R_{tot} is the total thermal resistance (conduction and convection) [m^2.K.W^{-1}].

In the system studied, steady state was considered and the water temperature didn't change very much, so the specific heat can be considered constant, simplifying the differential on the left side of equation Eq. (1).

2.1. Natural Convection

According to Incropera and Dewitt (1998), free convection in vertical plates is related to the development of a boundary layer over the heated plate. Considering the plate immersed in a fluid, with the surface temperature (T_{surf}) greater than the fluid temperature (T_∞), fluid density near the plate is smaller than that of the fluid far from the plate. So, buoyancy establishes a boundary layer of free convection in which the hot fluid flows upwards and drags the fluid from the vicinity. The velocity profile obtained is different from that associated to boundary layers in forced convection.

Still based on Incropera and Dewitt (1998), considering the empirical correlation found to simple geometric forms immersed in a fluid (external flow), Eq. (2) may be used to determine the average natural convection heat transfer coefficient, *h*:

$$Nu_L = h.L/k_f = C.Ra_L \tag{2}$$

where: Nu_L is the mean Nusselt number for the surface, *h* is the mean convection coefficient [W.m^{-2}.K^{-1}], *L* is a characteristic length [m], k_f is the thermal conductivity [W.m^{-1}.K^{-1}], *C* is the thermal capacity [W.K^{-1}] and Ra_L is the Rayleigh number.

Churchill and Chu (1973) *apud* Incropera and Dewitt (1998), recommend a correlation able to be applied for natural convection in vertical plates according to Eq. (3):

$$Nu_L = \left\{0{,}825 + \frac{0{,}387 \cdot Ra_L^{1/6}}{[1+(0{,}492/Pr)^{9/16}]^{8/27}}\right\}^2 \tag{3}$$

where: *Pr* is the Prandtl number.

The Rayleigh number, Ra_L, is determined from Eq. (4), where the properties must be determined at the mean temperature between the surface and the ambient. From Eq. (4) it can be seen that the characteristic length, *L*, has the greatest influence in the heat transfer coefficient and even a small change in its value represent a high influence in the amount of heat transfered.

$$Ra_L = Gr_L \cdot Pr = \frac{g \cdot \beta \cdot \left(T_{surf} - T_\infty\right) \cdot L^3}{\nu \cdot \alpha} \tag{4}$$

where: Gr_L is the Grashoff number, *g* is the gravity acceleration [m.s^{-2}], β is the thermal expansion coefficient (air is considered as an ideal gas) [K^{-1}], T_{surf} is the surface temperature [K], ν is the kinematic viscosity [m^2.s^{-1}] and α is the thermal diffusivity [m^2.s^{-1}].

The work of Stevanovic et alli (2005) for the steam condensation in a non-vented pipe insulated at the outer surface shows an external heat transfer coefficient of 5 W.m^{-2}.K^{-1} for a tube external temperature of 40^{o}C. Also Mendes and Philippi (2005) in a study of heat and moisture transfer through porous walls used an external heat transfer coefficient of 5 W.m^{-2}.K^{-1} for a wall external temperature of 35^{o}C. Since the temperature of the tube is near 70^{o}C in the baseboard convector, an external heat transfer coefficient of 6 W.m^{-2}.K^{-1} was adopted.

2.2. Radiation

The heat transferred through radiation is strongly influenced by surface temperature. Water temperature in a baseboard convector is around 70°C, and so the radiation heat transfer cannot be neglected. According to IPT (2003), the emissive power of a black body (E) is determined by Eq. (5):

$$E = \sigma.T^4 \tag{5}$$

where: E is the emissive power [W], σ is the Stephan-Boltzmann constant, equal to 5.67 x 10^{-8}, [$W.m^{-2}.K^{-4}$] and T is the black body temperature [K].

Another important concept related to radiation is the shape factor, defined as the fraction of emissive power that leaves one surface and reaches other surface. It depends only of the geometry and the relative position of the surfaces.

When dealing with real surfaces an emissivity factor must be used in the emissive power equation. All these definitions are joined in Eq. (6):

$$Q_{rad} = A.F_{1-2}.\varepsilon.\sigma.(T_{surf}^4 - T_\infty^4) \tag{6}$$

where: A is the surface area [m^2], F_{1-2} is the shape factor between surfaces 1 and 2 and ε is the emissivity factor.

For the baseboard convector, the shape factor, F_{1-2}, was estimated as 0.5 and if the external surface is made of aluminum, its emissivity can be considered as 0.5.

2.3. Global Heat Transfer Coefficient

The analysis of the global heat transfer coefficient for a given heat exchanger is very useful, since it allows understanding if there is a dominant thermal resistance in that system configuration. The amount of heat transferred in a heat exchanger is defined by an expression like Eq. (7):

$$Q = U.A.LMTD \tag{7}$$

where: U is the global heat transfer coefficient [$W.m^{-2}.K^{-1}$], A is the surface area [m^2] and $LMTD$ is the logaritmic mean temperature difference [K].

The $LMTD$ can be calculated as in Eq. (8):

$$LMTD = \frac{(T_e - T_\infty) - (T_s - T_\infty)}{ln\left[\frac{T_e - T_\infty}{T_s - T_\infty}\right]} \tag{8}$$

And finally, the global heat transfer coefficient is the inverse of total thermal resistance, found in Eq. (9):

$$R_{tot} = \frac{1}{h_e \cdot A_e} + \frac{ln(D_e/D_i)}{2 \cdot \pi \cdot k_t \cdot L_t} + \frac{1}{h_i \cdot A_i} \tag{9}$$

where: h_e is the external heat transfer coefficient [$W.m^{-2}.K^{-1}$], h_i is the internal heat transfer coefficient [$W.m^{-2}.K^{-1}$], A_e is the external heat transfer area [m^2], A_i is the internal heat transfer area [m^2], D_e is the tube external diameter [m], D_i is the tube external diameter [m], k_t is the tube thermal conductivity [$W.m^{-1}.K^{-1}$] and L_t is the tube length [m].

When there is a finned area, the fin efficiency must be introduced to account for the fact that, according to ASHRAE (1997), the temperature difference between the fin tip and the surrounding fluid is greater at the root than at the tip, causing a corresponding variation in the heat flux. Although this is an important factor to be considered, when the external thermal resistance is low and the fin material conductivity is high, fin efficiency charts show results near unity.

3. Material and Methods

The baseboard convectors studied are made of two copper tubes one over the other with external diameter of 22.23 mm (7/8"), thickness of 0.8 mm and rectangular fins with dimensions of 115 x 80 mm and a space of 11 mm between them. There were two sets of convectors 2380 mm long each, resulting in an internal area (A_i) of 0.617 m^2 and an external area (A_e) of 7.2868 m^2.

Three different circuits were tested to compare its performance. They were called configuration A, B and C. Circuit A has the two convectors and the two tubes in series as showed in Fig. 1.

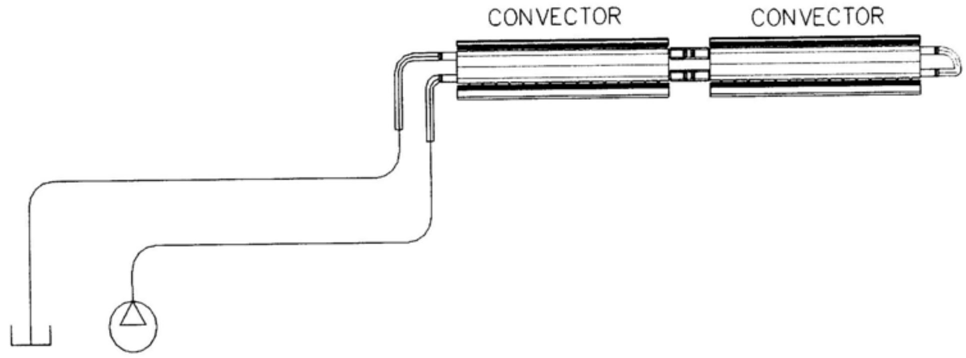

Figure 1 – Schematic of Baseboard Convectors with Circuit A

Circuit B has the two convectors in series but the two tubes in parallel and is showed in Fig. 2.

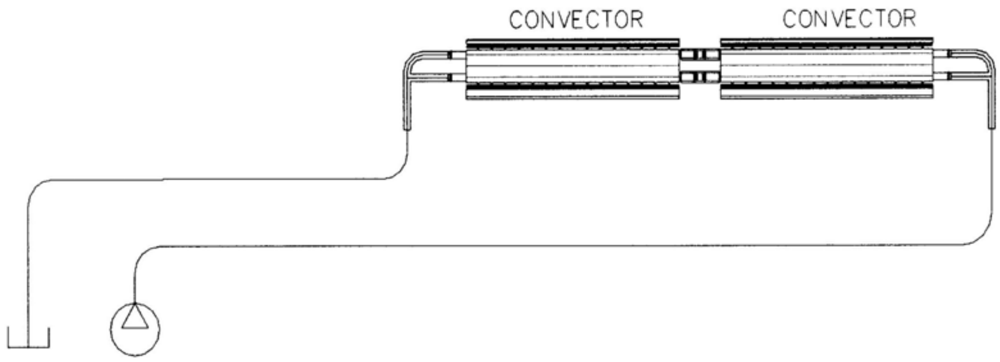

Figure 2 – Schematic of Baseboard Convectors with Circuit B

Circuit C has the convectors in parallel and the tubes in series as can be seen at Fig. 3.

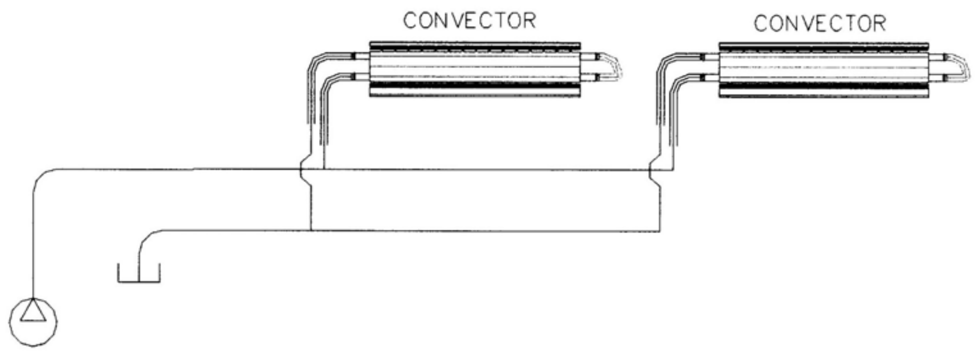

Figure 3 – Schematic of Baseboard Convectors with Circuit C

3.1. Flow Measurement

The water flow through the convectors was done by a centrifugal pump with variable speed. To measure convector performance at different water flows, the electric motor speed of the pump was varied by a frequency inverter, and the flow measured by means of a turbine type flowmeter with an axial rotor that gives the instantaneous value of the water flow in liters/min. The turbine flowmeter was mounted in a special assembly to ensure the correct straight length before and after the rotor and a bypass tube to protect it against sudden changes of the water flow that might damage the sensor. The accuracy of the flowmeter is 0.5% of full scale. The results obtained are shown in Fig. 4.

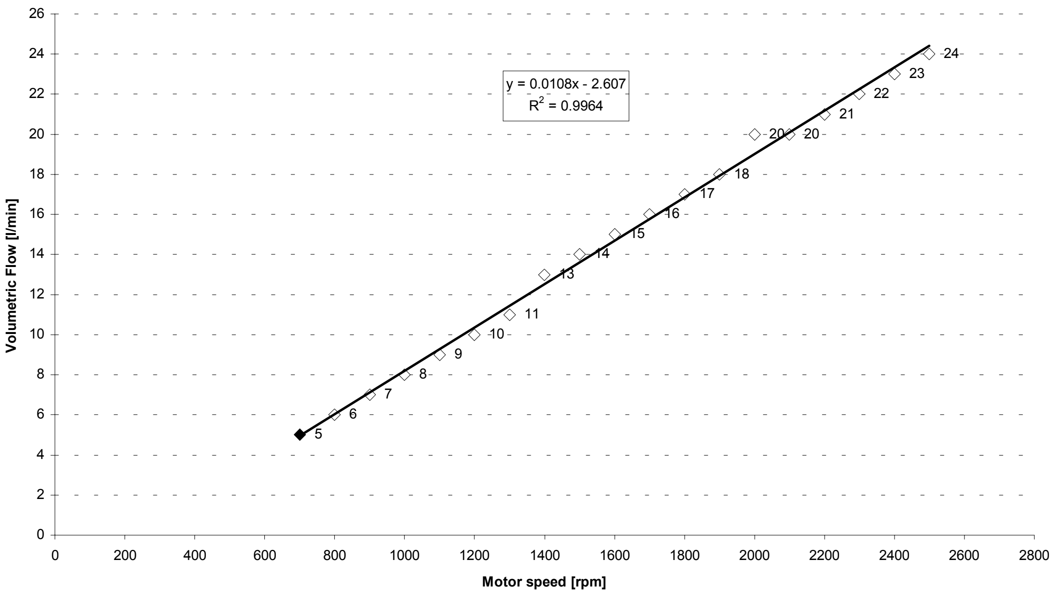

Figure 4 – Volumetric water flow x Motor Speed

At Fig. 4 it can be seen that there is a linear relationship between the volumetric flow and motor speed. Although this type of installation isn't common in practice, it's the best choice to control the water flow rate through the system, instead of using a valve at its outlet, because the pump operates with better efficiency.

3.2. Temperature Measurement

After establishing the operating parameters for the baseboard convectors, the inlet and outlet temperatures were measured to calculate the convector heat exchange rate. The temperature was measured using platinum resistance temperature (PT-100) devices connected to a data acquisition system that stores and organizes the values in a table form. PT-100's accuracy is 0.1% which means an uncertainty of 0.14°C. The temperature sensors were fixed inside the tubes in order to assure that the temperature was correctly measured. The same assembly was used for all circuits.

With the flow and the temperatures measured, the heat exchange capacity of the baseboard convectors was calculated using the energy balance equation, as can be seen in Eq. 10, which is similar to Eq. 1.

$$Q_{bc} = m_{water}.C_p.(T_e - T_s) \tag{10}$$

Since the water temperatures varied in a small range (4°C), water density and specific heat were considered constant. Using the formulation presented at Popiel and Wojtkowiak (1998) for thermophysical properties of liquid water, a value of 979.443115 kg.m^{-3} and 4.1887797 kJ.kg^{-1}.K^{-1} for respectively density and specific heat was considered. Therefore, the volumetric flow and the temperature differences measured were multiplied by 4102671.42 J.m^{-3}.K^{-1} to get the total heat transferred from water. Using the temperatures measured, the *LMTD* was calculated and the product *U.A* was determined, which is the inverse of total thermal resistance, R_{tot}. The results were compared to see which circuit gives the best performance and has the smallest thermal resistance.

4. Results and Discussions

Each circuit was tested during 90 minutes after stabilization and the entering and leaving water temperatures were recorded in the data acquisition system. The average values for flow and temperatures were used to calculate heating capacity and this is presented at Table 1, 2 and 3 for circuits A, B and C respectively.

Table 1 – Heating Capacity for Circuit A

Flow [l/min]	Te [°C]	Ts [°C]	Tavg [K]	Tamb [°C]	Qbc [W]	Qlin [W/m]
12	69.93	66.57	341.40	19.92	2757.0	579.2
14	69.41	66.49	341.10	19.73	2795.3	587.2
16	69.69	67.13	341.56	19.25	2800.8	588.4
18	69.63	67.33	341.63	19.36	2830.8	594.7

Table 2 – Heating Capacity for Circuit B

Flow [l/min]	Te [°C]	Ts [°C]	Tavg [K]	Tamb [°C]	Qbc [W]	Qlin [W/m]
12	69.46	66.70	341.23	18.89	2264.7	475.8
14	69.81	67.55	341.83	20.03	2163.5	454.5
16	69.56	67.40	341.63	19.40	2363.1	496.5
18	69.77	68.04	342.06	19.68	2129.3	447.3
20	69.48	67.77	341.78	19.34	2338.5	491.3
22	69.68	68.11	342.05	19.32	2361.8	496.2

Table 3 – Heating Capacity for Circuit C

Flow [l/min]	Te [°C]	Ts [°C]	Tavg [K]	Tamb [°C]	Qbc [W]	Qlin [W/m]
12	69.88	67.46	341.82	18.96	1985.7	417.2
14	69.21	67.12	341.32	19.18	2000.7	420.3
16	69.13	67.21	341.32	19.62	2100.6	441.3
18	69.75	67.96	342.01	19.38	2203.1	462.8
20	69.39	67.67	341.68	19.60	2352.2	494.2
22	69.78	68.30	342.19	19.33	2226.4	467.7

From the results obtained it can be seen that circuit A has the best configuration of all. Its capacity is 25.6% greater than that of circuit B and 35.1% greater than the capacity of circuit C using an average of the difference for each water flow. The main difference from circuit A to the others is that the entire flow of the water passes through one tube, while in circuits B and C it is divided. Therefore, the water velocity is much higher at circuit A, and also the pressure drop, increasing the internal heat transfer coefficient.

An important consideration that must be made is concerned to uncertainty. Since there is a small water temperature difference, combining the accuracies of PT100 sensors and turbine flowmeter gives a heat transfer uncertainty of 7.85%, 10.18% and 10.81% respectively for circuits A, B and C.

At Fig. 5, the variation of the capacity with flow for each circuit is showed. There is a slight increase of capacity with the increase of flow, although it isn't meaningful (an increase of 50% in the flow, increases only 2.7% the capacity at circuit A).

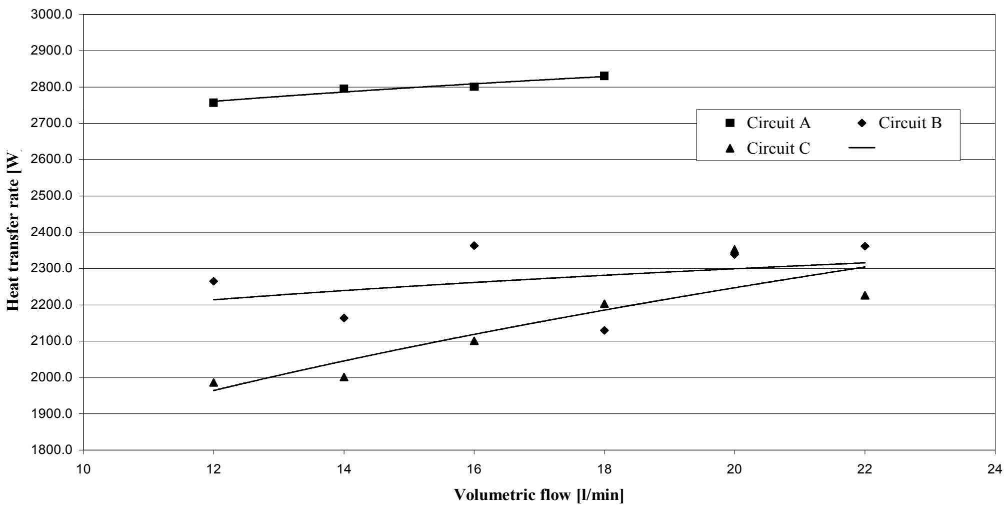

Figure 5 – Variation of Heat Transfer rate with Volumetric Flow for the 3 circuits

The capacity difference from circuit B to circuit C is around 6%, but considering the uncertainty mentioned before, this difference can't be assured. Also, the behavior difference of circuits B and C may be linked to this uncertainty. Nevertheless the higher heat transfer rate of circuit A can be assured since it's greater than the uncertainty, although the absolute values may differ slightly.

When calculating the radiation heat transfer is possible to verify, by the results at Tables 1, 2 and 3, that it may not change very much from one circuit to the other, since the water and ambient temperatures didn't change very much during the tests. A simple estimation gives an amount of 650 W transferred by radiation, what means 23, 29 and 31% of the average total heat transfer for circuits A, B and C, respectively.

Discounting the radiation heat transfer from the total and calculating the *LMTD*, is possible to determine the product *U.A* for each circuit. The results are: 44.2 W/K for circuit A, 32.9 W/K for circuit B and 30.3 W/K for circuit C. As mentioned before, the internal convective heat transfer coefficient is different from circuit A to circuits B and C because its velocity is the double. The average internal convective heat transfer coefficient was estimated varying from 3690 to 5100 $W.m^{-2}.K^{-1}$ for circuit A, and from 2120 to 3440 $W.m^{-2}.K^{-1}$ for circuits B and C. Since the external convective thermal resistance is much smaller (around 6 $W.m^{-2}.K^{-1}$) and don't change so much as the internal coefficient, the difference in the overall heat transfer coefficient from one circuit to the other isn't so high.

Since this type of device isn't common in Brazilian market, it wasn't found similar tests to compare. Also is important to notice that since the geometry is complex, there are other factors influencing the value of the thermal resistance which are not addressed here.

5. References

ASHRAE Handbook, 1997, "Fundamentals", Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

ASHRAE Handbook, 1996, "HVAC Systems and Equipment", Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers.

Incropera, F.P. and Dewitt, D.P., 1998, "Fundamentos de Transferência de Calor e de Massa", Rio de Janeiro: Ed. Livros Técnicos e Científicos.

IPT – Instituto de Pesquisas Tecnológicas, 2003, "Trocas de calor entre chamas e superfícies", Handbook of the Industrial Combustion course, Chapter 7, São Paulo.

Mendes, N.; Philippi, P.C., 2005, "A method for predicting heat and moisture transfer through multilayered walls based on temperature and moisture content gradients", International Journal of Heat and Mass Transfer, Vol 48, pp 37–51.

Pettersson, M. and Stenström, S., 2000, "Modelling of an electric IR heater at transient and steady state conditions. Part I: Model and Validation", International Journal of Heat and Mass Transfer, Vol 43, pp 1209-1222.

Popiel, C.O. and Wojtkowiak, J., 1998, "Simple Formulas for Thermophysical Properties of Liquid Water for Heat Transfer Calculations (from 0^oC to 150^oC)", Journal of Heat Transfer Engineering, Vol 19, No 3.

Recknagel and Sprenger, 1972, "Manual de Calefacción y Climatización", Madri: Ed. Blume, 1^{st} spanish edition of 57^{th} german edition.

Stevanovic, V.D.; Stosic, Z.V.; Stoll, U., 2005, "Condensation induced non-condensables accumulation in a non-vented vertical pipe", International Journal of Heat and Mass Transfer, Vol 48, pp 83–103.

6. Responsibility notice

The authors are the only responsible for the printed material included in this paper.