A NUMERICAL AND EXPERIMENTAL INVESTIGATION OF A FLAT PLATE COLLECTOR

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ABSTRACT

Radiative cooling systems by means of nocturnal longwave radiation have captured the attention of today's specialists involved in the construction building process. While most of the systems incorporate a flat-plate radiator that utilizes water, the present study accounts for the air as a heat transfer medium. A numerical model has been adapted for studying the performance of the radiative system under various working conditions. In parallel, a simplified flat-plate radiator has been built and tested at the Civil Engineering Department at Instituto Superior Técnico at Lisbon, Portugal, in order to validate the results numerically obtained. The experimental and numerical results agree well with each other.

KEYWORDS

Nocturnal longwave radiation, radiative cooling systems, numerical model.

INTRODUCTION

Reducing the cooling loads in buildings by means of natural cooling techniques is one of the major concerns of today's specialists involved in the construction building process. Among the natural cooling techniques, night radiation is an available process whose performance depends strongly on the local climate conditions. It is well known the fact that objects exposed to sky during clear nights reach temperatures lower that the ambient. They are in fact dissipating infrared radiation to the sky, a process governed by the intensity of downward atmospheric radiation. Using this concept with a radiator exposed to the sky during the night one can lower the temperature of a fluid that flows through it. The performance of a radiative cooling system improves with low ambient temperature and with clear weather conditions. In addition, a polyethylene cover for the radiator may be used to reduce the thermal convective gain from the surroundings without preventing the long wave radiation losses to the sky. Fluids, whose temperature has been lowered in such a way, could then be used for applications in building energy demand and comfort. In related literature, considerable research has been carried out to study the phenomenon of radiative cooling. Some of the works attempted to predict the effective sky temperature while some others focused on the performance of radiative cooling systems. Among most recent reviews on radiative cooling we mention here those performed by Santamouris et al (1996) and Givoni (1994).

The present study is a part of a more general one that is trying to evaluate the thermal behavior of a residential room building located in Lisbon, where an air based radiative cooling system is used. The whole study follows the general patterns suggested by Givoni (1982). The potential for radiative cooling in Lisbon has been evaluated first. At this moment, the performance of the radiative cooling system is studied. To conduct the study, a radiative cooling panel has been built. At the same time, a numerical model has been adapted for predicting the temperature of the air at the radiative panel outlet. This paper, presents some preliminary results from the numerical and experimental investigations.

NUMERICAL APPROACH

The flow radiative panel may schematically be represented in Figure 1. Air moves at a constant flow rate, and convection heat transfer occurs on the inner surface of the radiative plate. The model assumes steady-state conditions and neglects back and edges heat transfer effects.

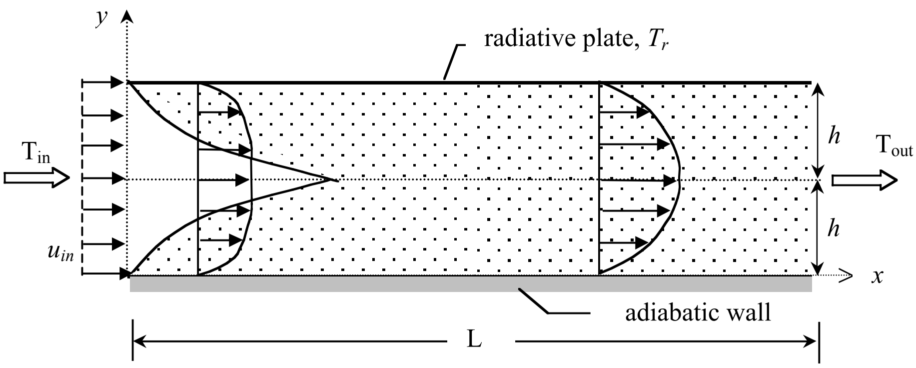

Figure 1: A schematic of the radiative system

In order to predict the temperature of the air at the panel outlet, a CFD computer code [1] has been modified. The program solves the transport equations for momentum and enthalpy, and accounts for turbulence using the well-known *k-ε* model, where the two additional transport equations (for kinetic energy and dissipation rate) are solved to deduce the eddy viscosity μ_t. These equations can be written in a generalized form for the dimensionless variable Φ, as in Versteeg et al (1995):

$$\frac{\partial}{\partial x}(\rho u \phi)+\frac{\partial}{\partial y}(\rho v \phi)=\frac{\partial}{\partial x}\left(\Gamma_\phi \frac{\partial \phi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_\phi \frac{\partial \phi}{\partial y}\right)+S_\phi \tag{1}$$

The convection and diffusion terms for all the transport equations are identical with Γ_Φ representing the diffusion coefficient for scalar variables and the effective viscosity μ_e.

The boundary conditions are stated as follows:

- At inlet ($x = 0$): $\dot{m} = 0.031$ kg/s; $v = 0$; $T_{in} = T_a$ (ambient air temperature)

[1] TEACH-2E, by Gosman et al. (1976)

- At the top wall ($y = 2h$): $u = 0$; $v = 0$; T_r = constant
- At bottom wall ($y = 0$): $u = 0$; $v = 0$; $\frac{\partial T}{\partial y} = 0$ (adiabatic wall).
- At outlet ($x = L$): $p = 0$; $\frac{\partial}{\partial x}[u, v, T] = 0$

At the topside of the radiator there is also a heat transfer associated with radiation and convection. However, these quantities have not been introduced as boundary conditions. The temperature of the radiative plate has been taken as constant and calculated experimentally.

To solve the non-linear second order partial differential equations on form given by Eqn. 1., they are discretized on a 40 x 20 grid according to a control volume based finite difference method. This divided the computational domain into 38 x 18 control volumes. The grid has been conceived in such a way as to allow us to introduce the value of the distance y from the wall to the first node of computational domain as desired. This approach is supposed to give us a little freedom when dealing with values of local Reynolds's number y^+ near the barrier of 11.63 (where the linear velocity profile in the viscous sublayer meets the logarithmic velocity profile in the inertial sublayer). For the velocity components u and v, staggered grids have been used. Primitive variables are used, with the velocities and pressure derived from SIMPLE algorithm developed by Patankar and Spalding (1972). For solving the discretisation equations for all the dependent variables (*Φ)*, the program uses a line-by-line iterative procedure (TDMA).

Various subroutines have been introduced into the model in order to calculate important flow parameters like Nusselt number, heat flux at the radiative plate and pressure drop.

EXPERIMENTAL APPROACH

A simple radiative cooling system has been build and mounted on the terrace roof. The experimental model consists of a 2.0 x 1.0 x 0.05 m (h = 0.025 m, see Figure 1) plywood box whose bottom and sides are thermally insulated with a 15mm polystyrene layer. The topside of the box is a 0.003 m painted stainless steel plate whose emissivity has been determined experimentally. In order to collect data, an automatic weather station was mounted on site. The station was equipped with an anemometer for measuring the wind speed, a sensor for measuring the solar irradiance and an air temperature and a relative humidity sensor. Three copper-constantan thermocouples were placed on the aluminum surface plate for temperature measurements. A similar type of thermocouple has been placed within the moving fluid at approximately 18 cm before the outlet to measure the fluid temperature at the symmetry axis. About 20 cm above the plate, has been mounted a net radiometer. This measured directly the net total radiation between the radiator surface and the sky vault. Air has been injected under the radiative plate at a constant flow rate of $\dot{m}$ = 0.031 *kg/s*.

The station was set-up to take readings every 10 minutes over a period of 15 hours during the day of 20 May 2001. Beginning at 1:22 AM, when it performed the first reading, and up to 7:22 AM, the system worked as a cooling device. After 7:22 AM, a time when the solar irradiance exceeded the radiant loss from the radiative plate, and up to 6:12 PM, when it performed the last reading, the system worked as a solar air collector. It should be pointed out that the set of data recorded during the experimental work corresponds to data for a clear day.

RESULTS

The temperature readings given by the thermocouples placed on the radiative plate were averaged, and the obtained value was used as an input data for the numeric model. The fluid properties have been evaluated at the mean fluid temperature.
According to the given geometry and initial velocity, the following D-based Reynolds number resulted:

$$Re_D = \frac{\rho u_{in} D_h}{\mu} = 3.761 \times 10^3 \tag{2}$$

Since the approximate barrier between laminar and turbulent regime in case of a flow through a straight pipe, according to Bejan (1987) has been established at $Re_D \cong 2300$, the value calculated with Eqn. 3. indicates that this is a turbulent flow. The temperature of the fluid (air) at the outlet has been evaluated numerically, for each reading performed over the 15 hours interval. In other words, the radiative system has been considered as operating under quasi-steady state conditions over each time interval between the readings. Figure 2 shows the numerical predictions versus experimental measured fluid temperature at the exit of the radiative panel.

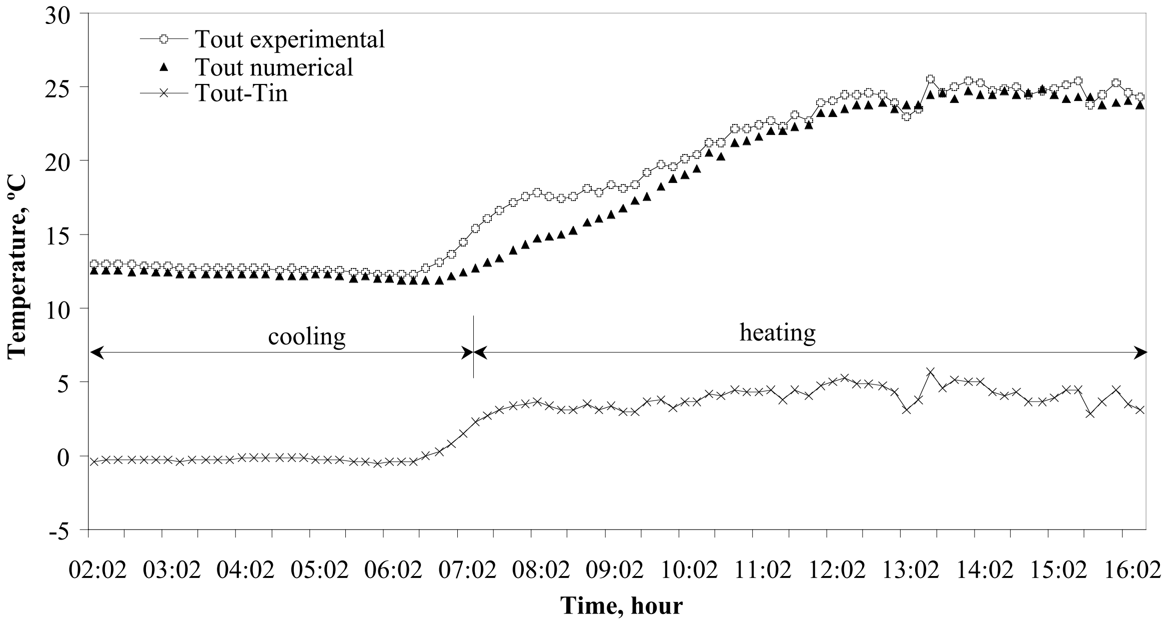

Figure 2: Numerical versus experimental outlet fluid temperature

The curve into the Figure 2 labeled "Tout-Tin" represents the quantity given by the difference between the outlet fluid temperature and the inlet fluid temperature. The comparison shows an acceptable qualitative agreement between the numerical and experimental predictions, with the exception of the time interval ranging between 7AM and 9AM. This time interval corresponds to the period of sunrise, when the direct component of the solar irradiation is mostly horizontal. Even though the thermocouple for measuring the outlet fluid temperature has been positioned inside the duct, the solar radiation "penetrated" the duct horizontally and was sensed by the thermocouple. In this way, the temperature readings recorded over this interval were strongly influenced by a radiative component.

Limiting further our analysis only to the radiative cooling, we may perform another verification. The overall heat balance of the radiative system is written as follows:

$$Q_{net} = R_{net} - Q_c \tag{3}$$

But the cooling power of the radiative system calculated with Eqn. 4. should be equal with the total heat flux lost by the fluid through the radiative plate. In order to see if these two quantities are in good agreement one with another, the convective component in Eqn. 4. has been evaluated with:

$$Q_c = h_c(T_a - T_r) = 1 + 6w^{0.75}(T_a - T_r) \tag{4}$$

where the convective coefficient h_c has been replaced with a formula provided by Clark and Berdahl (1980). Knowing that the net radiant loss of the radiative system is the quantity given by the radiometer, it was then possible to evaluate Q_{net} from Eqn. 3. The quantities of Eqn. 3., and the total flux traversing the radiative plate as calculated numerically are plotted together into Figure 3.

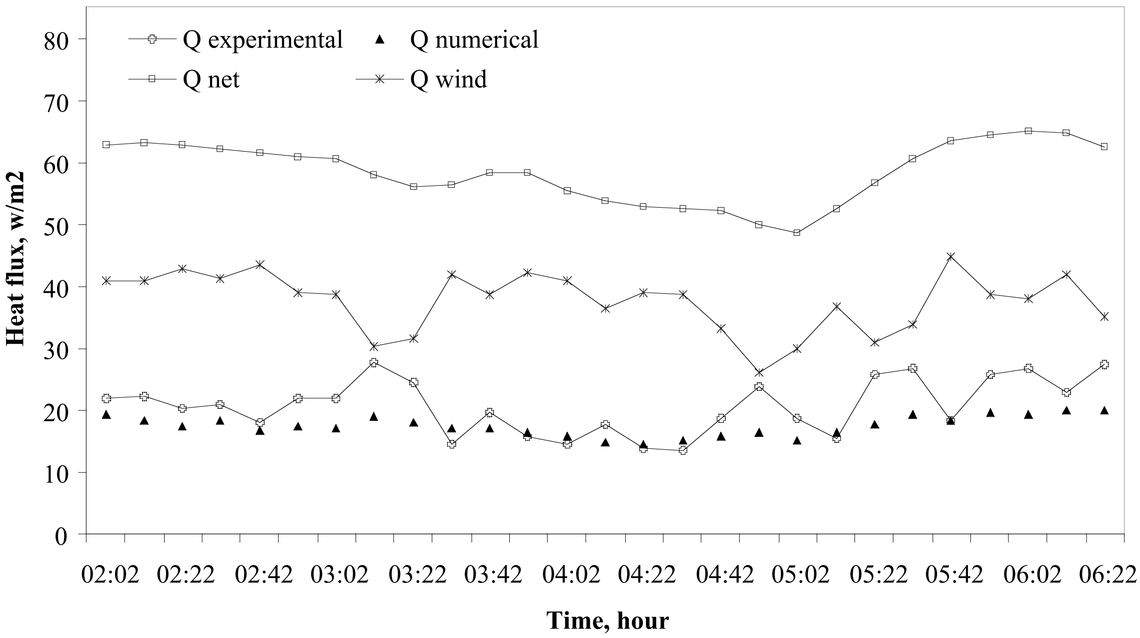

Figure 3: Numerical versus experimental total cooling power

Considering the fact that the convective heat exchange with the ambient air was roughly estimated, the comparison between the experimental and numerical Q_{net} shows relatively good agreement. It may be noticed from Figure 3 the significant proportion of heat gain due to convection. Worth mentioning here is that during the experiments, the average value of wind speed was about 3 m/s.

CONCLUSIONS

A CFD computer code has been adapted for calculating the temperature field within a flat plate radiative system that uses air as a heat exchange medium. The model uses as input data values recorded experimentally. A simple radiative cooling system has been build and tested in order to validate the numerical model. The comparison between the numerical and experimental results shows an acceptable qualitative agreement. On the other hand, the radiative system provided up to 0.5ºC instantaneous reduction in the temperature of the moving fluid and a maximum instantaneous efficiency η of 30%. The large amount of heat gain due to convection suggests the need for considering the application over the radiator of a polyethylene windscreen transparent to longwave radiation. Furthermore, some optimization studies might reveal the configuration for which a greater energy conversion is obtained.

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NOMENCLATURE

D_h	hydraulic diameter
h	half of the flat plate collector height
h_c	convective heat transfer coefficient, radiative plate to ambient air
L	length of the flat plate collector
p	pressure
Q_c	convective heat exchange between the radiative plate and the ambient air
Q_{net}	cooling power of the radiative system
Re_D	Reynolds number based on hydraulic diameter
R_{net}	net radiant loss of the radiative system
S_Φ	source term
T	temperature
T_a	ambient air temperature
T_{in}	inlet fluid temperature
T_{out}	outlet fluid temperature
T_r	radiative plate temperature
u_{in}	inlet fluid velocity
u	velocity component on x axis
v	velocity component on y axis
μ	laminar viscosity
μ_t	turbulent viscosity (eddy viscosity)
μ_e	effective viscosity, $\mu_e = \mu + \mu_t$
ρ	fluid density
η	instantaneous efficiency of the flat plate collector, $\eta = R_{net} / Q_{net}$
w	wind speed