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AN APPROXIMATE METHOD TO EVALUATE THE VENTILATION WITH A NATURAL CONVECTION SOLAR AIR HEATER

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طريقة تقريبية لتقييم التهوية بإستخدام سخان الهواء الشمسى بالحمل الطبيعى

خلاصه - استعرض هذا البحث نموزج رياضى مبسط لإستنتاج معدل السريان بالحمل الطبيعى فى سخان الهواء الشمسى من النوع ذات القناة المفتوحة . هذا النوع من السخانات يمكن إستخدامه بكفاءة فى نظام التهوية السلبية بالطاقة الشمسية ، وذلك بتوصيلة من أسفل بالفراغ المراد تهويتة . و قد تم إستخدام القيم المتوسطة على مقطع السريان لسرعة الهواء و درجات الحرارة و الضغط فى إستنتاج المعادلات الرياضية المكونة للنظام . و قد تم حل هذه المعادلات عدديا باستخدام ظروف حرارية حدودية مشابهة تماما لنظام حقيقى ، وكذلك بإستخدام مجموعة لابعدية مناسبة . و قد تم عمل عرض بيانى لمقارنة درجات حرارة الهواء المستنتجة بواسطة النموزج الرياضى و المسجلة من تجارب سابقه على نظام مماثل .

ABSTRACT - In the present paper, a simple mathematical model has been developed for the natural convection flow, in the channel-type solar air heater. This type of heaters can be used in a passive solar ventilation system by connecting it from the bottom to the space to be ventilated. The cross-sectionally averaged values of the flow velocity, temperature and pressure are used in the derivation of the governing equations. These equations are numerically solved with thermal boundary conditions similar to that of an actual system, and using appropriate dimensionless groups. A comparison between the temperature variations predicted from this model, and that recorded on a typical system is given in graphical form.

Introduction

The problem of natural convection flow between two parallel flat plates has been investigated by many researchers. This problem is subjected to both experimental study [1-3], and theoretical analysis [4,5]. However, there are several parameters that affect the free convection flow between two parallel plates. The effect of spacing between plates (or the aspect ratio) has taken a great interest in the majority of reported data. The influence of channel inclination angle is also investigated [3]. The problem has been studied for a wide range of Prandtl, Grashof numbers, and thermal boundary conditions. Outdoor experimental data on a solar air heater are also available [6].

The classical way to deal with the problem theoretically; namely the solution of continuity, momentum and energy equations in two dimensional form has been followed by most investigators. However, the method of solution and the selection of inherent thermal boundary conditions differ from one to another. The simplifying assumptions depend to some extent on the specific field of application. However, it is more convenience for the solar energy system designer to deal with easier models, even if some approximations are involved. This is because of the large number of design and operating parameters that exist in any solar energy system design procedure.

Theoretical model

This is a problem of free convection flow between two inclined parallel plates, the upper is uniformly heated and the other is insulated. The assumptions made in this flow situation are that the flow is laminar, steady, and fully developed. The model is also based on the Boussinesq approximation, in which the fluid properties are considered constant except for the body force term. The following linear variation of the air density ρ, with temperature T, is considered,

$$\rho = \rho_o \left[1 - \beta(T - T_o) \right]$$

Where β is the coefficient of thermal expansion of the air, and ρ_o is the air density at a reference temperature T_o.

Referring to the geometry and coordinate system sketched in Fig. 1, the control volume at a distance x from the channel inlet is considered. The control volume has a length dx, width D and a unit breadth. The continuity equation at x, can be written in differential form as,

$$\rho \frac{du}{dx} + u \frac{d\rho}{dx} = 0 \qquad (1)$$

where u is the mean velocity of air at x,
and ρ is the air density at this section.

The momentum equation for the control volume is given by,

$$\rho D u \frac{du}{dx} = - D \frac{dp}{dx} + \rho g D \sin \phi - 2 \tau_w$$

where ϕ is the channel tilt angle,
p is the pressure,
and τ_w is the wall shear stress.

The term on the LHS of the above equation represents the inertia force, while those on the RHS are the pressure, gravity and frictional forces respectively.
Results of previous investigations of the closed and open natural convection loops indicate that the wall shear stress can be

expressed as [8],

$\tau_v = \rho\, u^2\, f/8$

where the friction factor $f = 24 / Re$
The characteristic length in the above Reynolds number Re, is the channel length.
Substituting for τ_v , the momentum equation will be reduced to,

$$u \frac{du}{dx} = - \frac{1}{\rho} \frac{dp}{dx} + g \sin\phi - 6\, \nu\, u\, (D/l) / D^2 \qquad (2)$$

where ν is the kinematic viscosity of air.
and l is the channel length.
The pressure variation in the transverse direction is assumed to be zero.

If the viscous dissipation and heat conduction in the axial direction are assumed negligible, compared to the thermal convection, the energy equation becomes,

$$C_p\, \rho\, D\, u \frac{dT}{dx} = h\, (T_p - T) \qquad (3)$$

where C_p is the air specific heat at constant pressure,
T_p is the absorber plate temperature,
and h is the film heat transfer coefficient.
Correlations for h are available [9], but the plate temperature in the above equation introduces some complications.

However, the use of solar collector analysis was found more helpful in the solution procedure. Referring to the energy balance equation of the solar air heater [10], the heat gain per unit collector area is $F_R[q-U_L(T - T_a)]$. Therefore, the energy equation for the given control volume can be written as,

$$C_p\, \rho\, D\, u \frac{dT}{dx} = F_R\, [\, q - U_L(T - T_a)\,] \qquad (4)$$

where F_R is the collector heat removal factor,
U_L is the collector overall heat loss coefficient,
and T_a is the ambient air temperature.
The velocities and temperatures appearing in the above equations are cross-sectionally averaged values.
The following dimensionless quantities are often used :

$$X = \frac{x}{D\, Gr} \qquad U = \frac{D\, u}{\nu\, Gr}$$

$$\theta = \frac{(T - T_i)\, k}{q\, D} \qquad P = \frac{(p_a - p)\, D^2}{\rho\, \nu^2\, Gr^2}$$

$$Gr = \frac{g\,\beta\,q\,D^4}{k\,\nu^2} \qquad Pr = \frac{Cp\,\nu\,\rho}{k}$$

where, q is the constant heat flux absorbed by the upper plate. Equations 1, 2 and 4 can be converted to the following dimensionless forms :

$$\frac{dU}{dX} - \frac{D\,\rho_o\beta\,q}{\rho\,k}\,U\,\frac{d\theta}{dX} = 0 \tag{5}$$

$$\frac{dP}{dX} = -\,U\,\frac{dU}{dX} + \theta\,(\rho_o/\rho)\,\sin\phi - 6\,(D/l)\,U \tag{6}$$

$$\frac{P_r\,q}{F_R}\,U\,\frac{d\theta}{dX} = q - U_L\,[q\,D\,\theta/k - (T_a - T_i)] \tag{7}$$

To solve these three equations, a set of boundary conditions must be specified :

for $X = 0$ $\quad \theta = 0$, $U = U_o$ and $P = 0$

for $X = L$ $\quad P = 0$

where L is the channel dimensionless length = $l/(D\,Gr)$
The air enters the channel at a temperature T_i, with a uniform velocity U_o. In free convection problems, the inlet velocity U_o is not known, and must be determined as a part of the solution process. The forward marching, implicit finite difference method with iteration was used to solve these equations. If L is divided into N equal differential segments ΔX, the corresponding finite difference equations will be :

$$U_{i+1} = U_i + \frac{D\,\rho_o\beta\,q}{\rho\,k}\,U_i\,(\theta_{i+1} - \theta_i) \tag{8}$$

$$P_{i+1} = P_i - U_i\,(U_{i+1} - U_i) + \Delta X\,\theta_i\,(\rho_o/\rho)\,\mathrm{Sin}\,\phi - 6\,(D/l)\,U_i \tag{9}$$

$$\frac{Pr\,q}{F_R}\,U_i\,(\theta_{i+1} - \theta_i) = \{\,q - U_L\,[q\,D\,\theta_i/k - (T_a - T_i)]\,\}\,\Delta X \tag{10}$$

with the following boundary conditions :

$\theta_1 = 0$, $\quad U_1 = U_o$ $\quad$ and $\quad P_1 = P_{N+1} = 0$

The solution is obtained by first selecting a value of U_o. Then by means of a marching procedure, the variables U, θ and P for each step beginning at step i+1 are obtained using the values at the previous step i. The procedure is continued until the pressure P returns to zero (or just positive) at the end of the channel.

Results and Discussion

In order to solve the above equations, the involved design and operating parameters should be predetermined. However, the following values are chosen for a typical real system [7]:

$U_L = 6\ W/m^2$ $F_R = 0.7$

$\phi = 30^\circ$ $D = 0.01\ m$

The air properties (k, ν and Pr) are taken at 40°C, which is an average temperature for a typical solar air heater. The solution is obtained for N = 31, which is a good compromise between the computation time and accuracy of results.

Results are obtained for Gr = 16200 and 32400, corresponding to solar radiation intensities H = 400 and 800 W/m^2 respectively. It is to be noted that $q = (\alpha\tau)$ H, where $(\alpha\tau)$ is the effective transmissivity-absorptivity product of the solar air heater and is assumed to be 0.8. The air velocity, temperature and pressure are plotted in Fig. 2 in dimensionless forms against X/L, for the two values of Grashof number. It is clear that U and θ increases linearly with X, while P decreases to a minimum value at about X/L = 0.5, and then increases to zero again at X/L= 1. Since the values of P and P_{min} are negative, P/P_{min} is positive. It is clear that U decreases markedly with Grashof number, but P has almost no effect with it. The variation of θ with Grashof number is very small as shown in Fig. 2. The effect of channel aspect ratio L/D, on the air flow rate and maximum dimensionless temperature (at X/L = 1) is displayed in Fig.3. As expected, both values increase with aspect ratio.

Theoretical results obtained from this model has been compared with the experimental data collected by the writer and others [7]. All the system dimensions and actual meteorological conditions during the experiments are employed in the model. The solar radiation intensity on the tilted collector plane, and the ambient air temperature for one experiment are shown in Fig. 4. These data are used to predict the air temperature through the channel. Figure 5, shows a comparison between the predicted and measured air temperature variations with time at x=0, 0.5 and 1 m. As expected, the predicted temperature is rather lower than the measured,specially at lower values of x. This is because each predicted temperature is an average value over the cross section, while the measurements are performed at the channel centre line. Besides, at smaller values of x, the flow may be still developing, with more transverse variation of temperature. The air temperature distribution in the flow direction at different hours is given in Fig. 6. The same trend is observed for both the theoretical and experimental data. However, the agreement between both results may be improved by using more accurate design parameters, that are experimentally predetermined.

Conclusions

A simple mathematical model has been developed for the natural convection flow, in the channel-type solar air heater that can be used in a passive solar ventilation system. The governing equations are numerically solved by the forward - marching finite difference method, assuming thermal boundary conditions which are similar to that of an actual system. The variations of the dimensionless forms of air velocity, temperature and pressure with the dimensionless distance from the channel bottom, at different values of Grashof number are obtained. The theoretical results obtained from this model have been compared with the experimental data collected on a typical system. The theoretical results have shown little underestimation with respect to the measured data. A satisfactory agreement would be obtained, if the involved design parameters are accurate, or experimentally evaluated. However, the simplicity of this model is encouraging to recommend it for quick and approximate evaluation of the solar passive ventilation in an actual system.

Nomenclature

C_p Specific heat of air at constant pressure, J/kg K.
D Spacing between plates, m.
F_R Collector heat removal factor.
H Total solar radiation intensity on a tilted plane, W/m^2.
f Friction coefficient.
Gr Grashof number.
g Gravitational acceleration, m/s^2.
h Film heat transfer coefficient, W/m^{2}K.
k Air thermal conductivity, W/m K.
L Channel dimensionless length.
l Channel length, m.
p Pressure, N/m^2.
P Dimensionless pressure.
Pr Prandtl number.
q Heat flux, W/m^2.
Re Reynolds number.
T Air temperature, K.
U Dimensionless velocity.
$\alpha\tau$ Effective transmissivity-absorptivity product.
β Coefficient of thermal expansion of air, K^{-1}.
θ Dimensionless temperature.

ϕ Channel tilt angle, degrees.

ν Kinematic viscosity, m^2/s.

References

1. W. Aung, L. S. Fletcher and V. Sernas : Developing laminar free convection between vertical flat plates with asymmetric heating, Journal of Heat and Mass Transfer, Vol.15,pp. 2293-2308, (1972).
2. E. M. Sparrow and L. F. A. Azevedo :Vertical-channel Natural Convection Spanning Between the Fully-developed Limit and the Single-plate boundary-layer Limit, Int. Journal of Heat and Mass Transfer, Vol. 28, No. 10, 1847-1857, (1985).
3. L. F. A. Azevedo and E. M. Sparrow : Natural convection in open-ended Inclined channels, Journal of Heat Transfer, Vol. 107, pp. 893-901, (1985).
4. A. Moutsoglou: An Inverse Convection Problem,Journal of Heat Transfer. Vol. 111, February (1989).
5. S. Ramanathan and R. Kumar : Correlations for natural convection between heated vertical plates. Journal of Heat Transfer, vol. 113, pp. 97-107, (1991).
6. I. C. Macedo and C.A.C. Altemani: Experimental evaluation of natural convection solar heaters, Solar Energy, Vol. 20. pp. 367-369, (1978).
7. M. S. Abdel Salam, M. A. Rayan, H. E. Gad and A. E. Kabeel : Passive Solar Ventilation Using an Air Heater.
8. A. Chen : Design of Plane-type Bidirectional Thermal Diode, Journal of Solar Engineering, Vol. 110, November (1988).
9. W. M. Kays : convective Heat and Mass Transfer, McGraw-Hill Book Co., New York, (1972).
10. F. Kreith and J. F. Kreider:Principles of Solar Engineering, McGraw-Hill Book Co., New York, (1978).

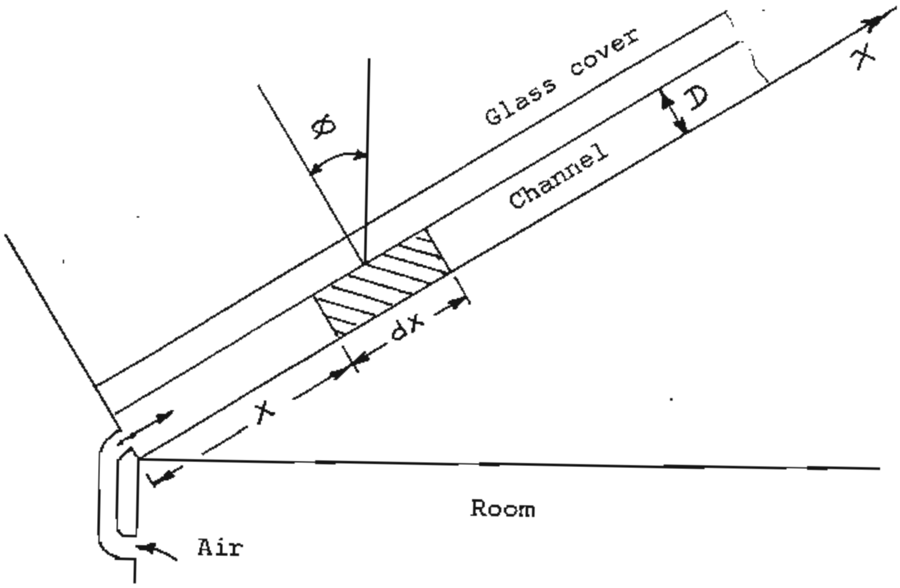

Fig. 1. Geometery and coordinate system.

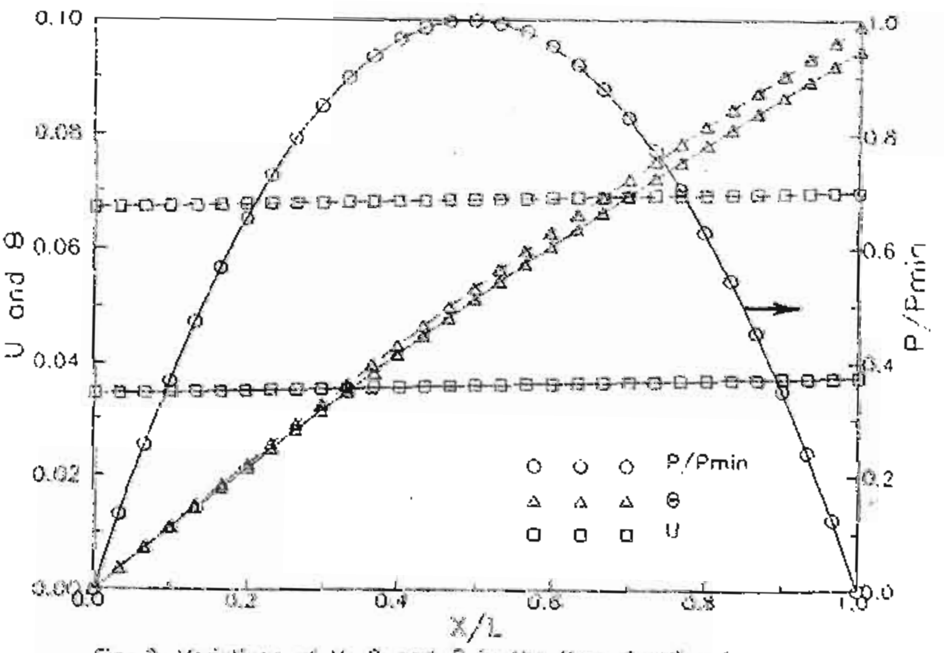

Fig. 2. Variations of U, Θ and P in the flow direction for
- - - - - Gr = 13200
——— Gr = 26400

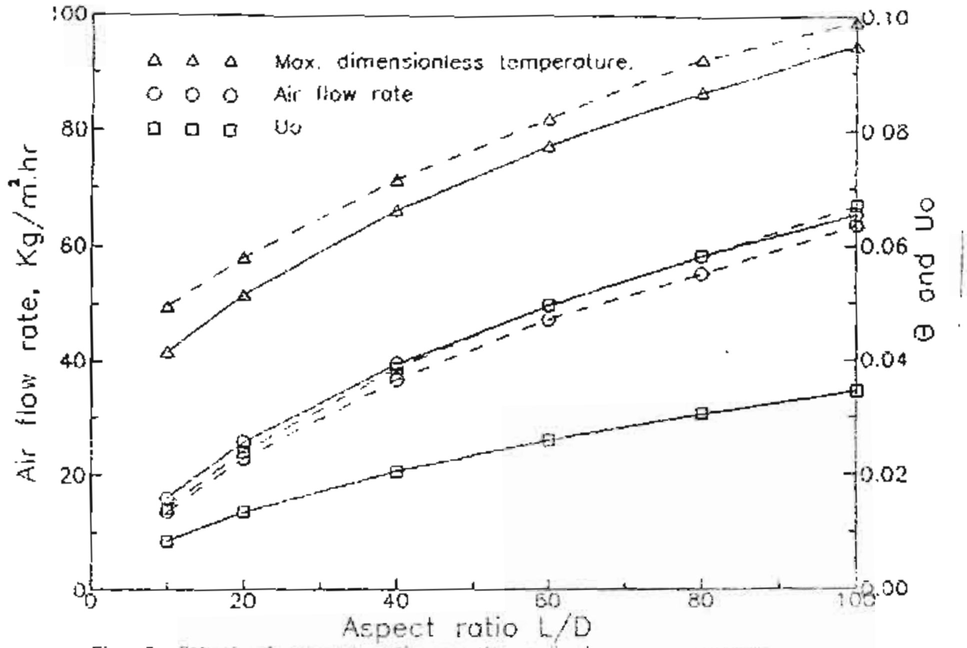

Fig. 3. Effect of aspect ratio on the collector performance

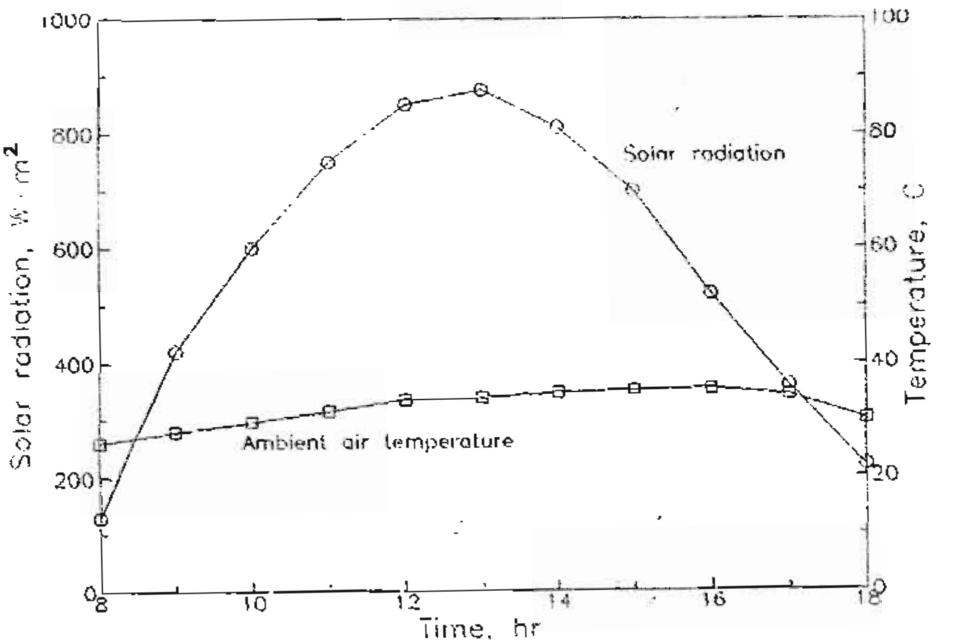

Fig. 4. Actual environmental conditions recorded during the experiment.

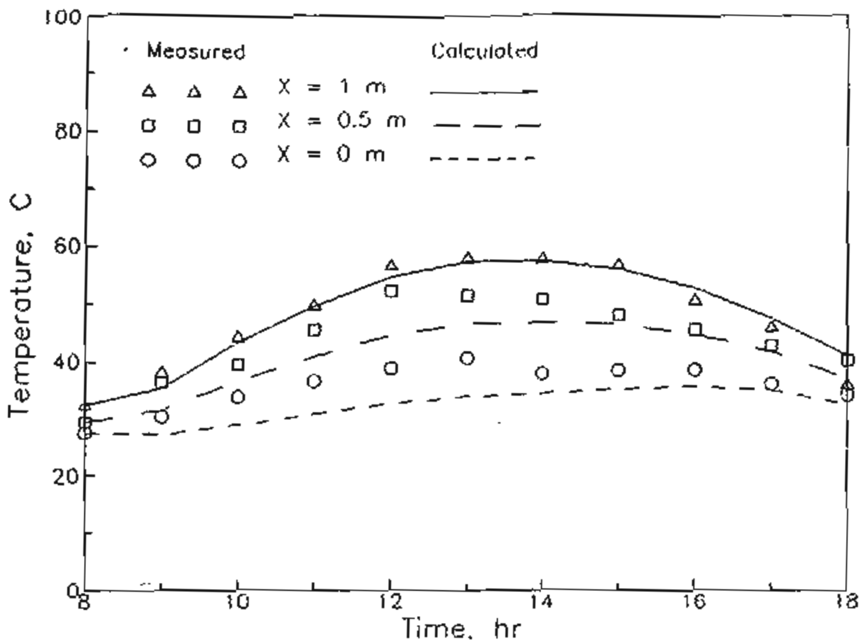

Fig. 5. Temperature variations with time at different locations.

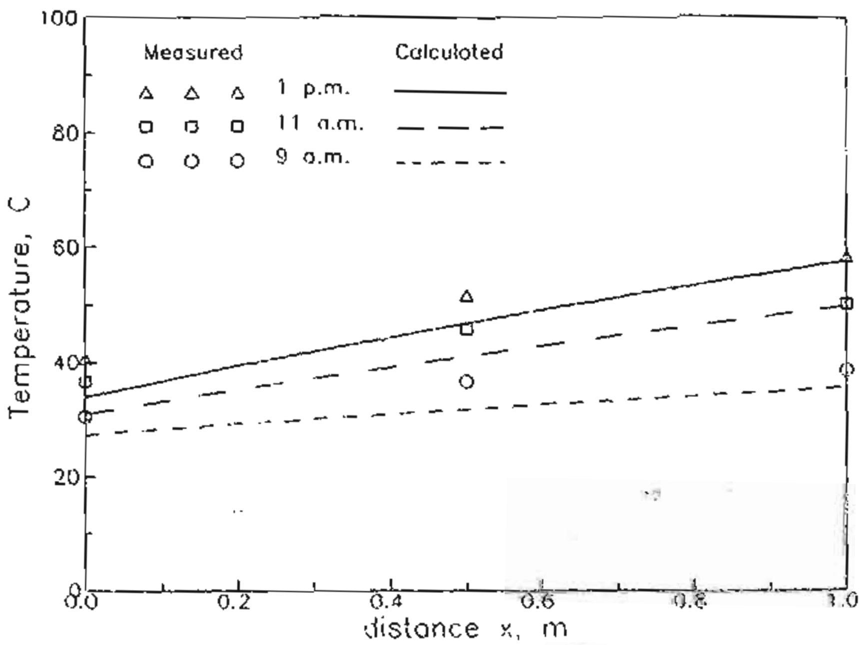

Fig. 6. Air temperature distribution in the flow direction.