Study Of Flow And Heat Transfer In A Vertical Heated Duct As A Passive Solar Ventilation-System

دراسة السريان وانتقال الحرارة فى ممر رأسى كنموذج لنظام التهوية باستخدام الطاقة الشمسية

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خلاصة: فى هذه الدراسة تم تصميم وتنفيذ نموذج معملى لنظام التهوية باستخدام الطاقة الشمسية. هذا النموذج عبارة عن ممر مستطيل المقطع يتم تسخينه كهربيا عن طريق أحد أوجهه فى حين بقية الأوجه معرضة لظروف الجو المحيط. نتيجة للتسخين المنتظم يتولد سريان بالحمل الحر. حسبت قيمة معدل سريان الهواء ورقم نسلت اعتمادا على قياسات درجات الحرارة للهواء الخارج والسطح الساخن. تم دراسة أداء النموذج لقيم مختلفة للتدفق الحرارى ولقيم مختلفة للنسبة البعدية للممر. أختبرت ممرات قيم النسبة البعدية لها تساوى ٠,٠١، ٠,٠٢، ٠,٠٣، ٠,٠٤ لتدفق حرارى يساوى ١٥٠، ٢٥٠، ٣٥٠، ٤٥٠، ٥٥٠، ٦٥٠، ٧٥٠ وات/ متر مربع. فى هذه الدراسة أقترحت علاقتين رياضيتين لحساب قيمة رقم نسلت ومعدل سريان الهواء.

ABSTRACT

An experimental model for passive solar ventilating system is designed and constructed. This model is a rectangular vertical channel electrically heated from one of the two large sides and the other sides are exposed to ambient conditions. Due to uniform heating, a natural convective flow is produced. The induced air flow rate and average Nusselt number are calculated depending on the values of measured temperature of exit air and hot wall. The performance of the model at different values of heat flux as well; at different values of channel space ratio is examined. Channels of space ratio of 0.01, 0.02, 0.03, and 0.04 are tested at values of heat flux of 150, 250, 350, 450, 550, 650, and 750 w/m^2. Two proposed correlations for evaluation of Nusselt number and the mass flow rate are presented here.

1. INTRODUCTION

Solar energy applications are attractive systems to be studied and used because of their simplicity, cost effectiveness and pollution problems of other classical systems. Trombe wall systems [1] and Danvior house systems [2] can be considered as the main types of solar ventilating system [2]. Trombe wall system is a very simple system. It depends on production of natural convective flow in a vertical channel heated from one side. This channel is connected to the heated room through two openings, one at the top of the room and the other is at its bottom. The heated side of the channel is a black painted wall and the other sides are made of transparent glass. On the other hand, Danvior house system is a complicated system. It depends on circulating the heating air in a double layer's solar heater by a fan and supplying it to the heated room. The effect of the inlet and exit losses of the flow within the Trombe wall was studied by J. A. Tichy [3].

Natural convection heat transfer has been investigated for case of bare plate as well as for channels at different conditions either theoretically or experimentally [4-9]. The effect of interplate spacing on natural convection in an open-ended channel bounded by an isothermal and an unheated wall was studied by Sparrow and Azevedo [6] both experimentally and numerically. It was found that the channel heat transfer is particularly sensitive to changes in interplate spacing for narrow channels and at small temperature differences. Free convective laminar heat transfer between the channel surfaces of the Trombe wall has been investigated theoretically by Akbari and Borgers [10]. Several correlations have been developed in [10] to estimate performance characteristics given the channel thickness, height, and surface temperatures.

In the present work, a simple open-ended vertical channel model is selected to study the performance of a solar ventilating systems under prescribed conditions. Through out the experiments air is used as the working medium. The effect of heat flux and of space ratio on the average Nusselt number, mass flow rate of air and its exit temperature are studied.

2. APPARATUS AND EXPERIMENTAL PROCEDURE

The used apparatus is designed such that it satisfies modeling requirements. This model ensures different channel space ratios and different values of heat flux from the heated wall. A schematic sketch of the used channel is shown in fig.(1). The heated wall is 0.8 mm thick galvanized steel plate with height of 1000 mm and width of 290 mm. The plate is heated by an electric heater. This heater consists of fifty stainless steel stripes of 0.5 mm thickness, 10 mm width and 1050 mm length. These stripes are connected to each other in series. The other principal wall of the model; as well the two small sides are made of 6.0 mm thick transparent glass. The unheated principle wall is position-adjusted one to obtain different required space ratio (S/H).

The supplied power to heater is adjusted to the required level by changing the value of the input voltage through manual stepless voltage-regulator. Temperature measurement is carried out by copper-constantan thermocouples with sensitivity of 0.0435 mV/°C accompanied with multimeter of readability of 0.01 mV.

Thirteen thermocouples are solded to the heated wall, five

Thirteen thermocouples are solded to the heated wall, five are fixed to the unheated wall,one to back of insulation, one in the middle point of inlet cross-section and place-adjusted one at exit cross-section. The value of current intensity and potential difference are measured by analog ammeter and voltmeter respectively.

3. RESULTS AND DISCUSSIONS

The temperature of the heated wall is measured at several points in longitudinal and transverse directions. In transverse direction, no temperature variations are exhibited. Figures (2-a)-(2-d) show temperature distribution along the hot wall at different values of heat flux; for channel space ratio of 0.01,0.02,0.03, and 0.04 respectively. It is clear that, the value of temperature, at all measuring points, is proportional to the value of heat flux. In case of high heat flux, it is observed that the temperature distribution curves have a peak point within the first third of the wall length which is physically acceptable. In general, the temperature of the wall, for the same heat flux, is higher in case of smaller space ratios.

Figures (3-a) and (3-d) show the temperature distribution along the unheated wall at different values of heat flux for channel space ratio of 0.01,0.02,0.03, and 0.04. The temperature increases almost linearly in direction of flow for all values of heat flux.

The temperature distribution at the channel exit cross-section is presented in figures (4-a) and (4-d). In general, for all space ratios; the temperature decreases from the hot wall to the unheated one. It is noted that, The value of temperature is higher for smaller space ratios. Examining the above figures, it is clear that the value of heat flux is more effective in case of small space ratios, which is expected for narrow channels.

The mass flow rate is calculated using the energy equation; $q_v = \dot{m}\ C_p\ \Delta T$, where q_v is the net heat flux. The net heat flux (q_v) is equal to the supplied heat minus the heat lost to surroundings from the outer surface of the unheated plate and from the outer surface of the insulation layer of the heated plate ($q_{net} = q_s - q_{lost}$). The lost heat is calculated based on the free convection heat transfer equation in case of vertical flat plate [10]. The temperature difference (ΔT) in the energy equation represents the difference between the average temperatures at exit and inlet cross sections. These calculations are based on the assumption of constant physical properties of air. The effect of channel space ratio on the exit mean temperature and on the mass flow rate are shown in figures (5) and (6) respectively. As shown in figure (5), the temperature decreases slowly with increasing the channel space ratio specially for low values of heat flux, while for high values of heat flux it first decreases with increasing space ratio and then records some increase. In figure (6) an optimum condition of mass flow rate can be observed at channel space ratio of 0.03 for high values of heat flux, while for low values of heat flux the mass flow rate increases with the channel space ratio and no optimum condition can be observed.

the parameters of the problem in dimensionless form. The physical properties of dry air at average temperature of air in the channel(T_{av}) are used to calculate the dimensionless numbers of the problem. Fig.(7) shows the relation between dimensionless mass flow rate (M^*) and the dimensionless quantity $[(S/H).Ra_s]$. Where M^* is defined as $\dot{m}/\rho\nu\ Gr_s$, Gr_s is Grashof number defined as $g\beta S^4 q_s/k\nu^2$. Ra_s is Rayleigh number based on the interplate spacing S and defined as Gr_s Pr. In this figure the obtained experimental results and its proposed correlation are plotted. The dimensionless mass flow rate M^* can be expressed by the following correlation:

$$M^* = a\ [(S/H)\ Ra_s]^b\ ,$$

where a = 0.9850 and b = 0.649. In figure (8) Nusselt number ($Nu_s = h_{av}\ S\ /\ k$) is plotted against the dimensionless quantity $[(S/H)\ Ra_s]$. The proposed correlation is given by;

$$Nu_s = c\ [(S/H).Ra_s]^d\ ,$$

where c = 1.299 and d = 0.1831.

For comparison, the present experimental data and the data obtained using the correlation reported by Akbari and Borgers [9] are plotted in fig.(9). The obtained experimental data of Sparrow and Azevedo [6] are presented on the same figure. It is clear that the present data lay between that obtained in [6] and [9] and are closer to that reported by Sparrow [6]. The deviation of the data obtained by Akbari can be explained due to ignoring the effect of the heated wall on the unheated wall which is considered in the model of Sparrow. In the same time, the present results are acceptable compared with that of Sparrow because his model considers the case of isothermal heated wall, while the present work deals with constant heat flux case. On the other hand, the curve of the dimensionless mass flow rate obtained in the present work lay above the data obtained using the correlation proposed by Akbari [9] for the Trombe wall channel as shown in fig.(10).

4. CONCLUSION

From the presented experimental data, two proposed correlations are obtained describing the thermal performance of different size channels under various operating conditions. Accordingly, one can determine the suitable channel size to fit the design requirements of a desired system.

NOMENCLATURE

C_p specific heat
g acceleration of gravity
Gr_s Grashof number = $[(g\ \beta\ S^4\ q)/\ (k^2 \nu\ )]$
k thermal conductivity of dry air
H channel height
M^* dimensionless mass flow rate = $\dot{m}\ /(\ \rho\ \nu\ Gr)$

$\dot{m}$ mass flow rate
Nu_s Nusselt number based on the interplate spacing (S) = h_{av} S/k
Pr Prandtl number of air = $[(\rho\nu C_p)/k]$
q_{lost} rate of lost heat to surrounding
q_s supplied heat rate
q_{net} net heat rate = $q_s - q_{lost}$
Ra_s Rayleigh number = $Gr_s.Pr$
T temperature
T_e air exit temperature
T_i air inlet temperature
ΔT temperature difference $(T_e - T_i)$
β coefficient of thermal expansion
ν kinematic viscosity
ρ density of air

REFERENCES

1. Kreider and Kreith, Solar energy hand book, Mc Graw-hill book company, pp 12-18, 1981.
2. W. C. Dickinson, P. N. Cheremisinoff, Solar Energy Technology hand book, Marcel Dekker Inc., pp 142-148, 1980
3. J. A. Tichy, The effect of inlet and exit losses on free convective laminar flow in the trombe wall channel, J. of Solar Energy Engineering, Vol.105, pp. 187-193, May 1983.
4. J. T. Beard, R. J. Ribardo, R. R. Adams and J. H. Arthur, Thermal Performance of a heated plate with open-channel infrared radiation shield, Transactions of ASME, Vol. 111, pp 232-236, Aug. 1989.
5. T. S. Chen, H. C. Tien and B. F. Armaly, Natural convection on horizontal, inclined and vertical plates with variable surface temperature or heat flux, Int. J. Heat Mass Transfer, Vol. 29, No. 10, pp 1465-1478, 1986.
6. E. M. Sparrow and L. F. Azevedo, Vertical-channel natural convection spanning between the fully developed limit and the single-plate boundary layer limit, Int. J. Heat Mass Transfer, Vol. 28, No. 10, pp 1847-1857, 1985.
7. L. F. Azevedo and E. M. Sparrow, Natural convection in a vertical channel vented to the ambient an aperture in the channel wall, Int. J. Heat Mass Transfer, vol. 29, No. 6, pp 819-830, 1986.
8. L. F. Azevedo and E. M. Sparrow, Natural convection in open-ended inclined channels, Transactions of ASME, Vol. 107, pp 893-901, Nov. 1985.
9. H. Akbari and T.R. Borgers, Free convective laminar flow within the Trombe wall channel, Sol. Energy 22, 165-174, 1979
10. J. P. Holmann, Heat Transfer, Mc Graw-Hill Book Company, 1981

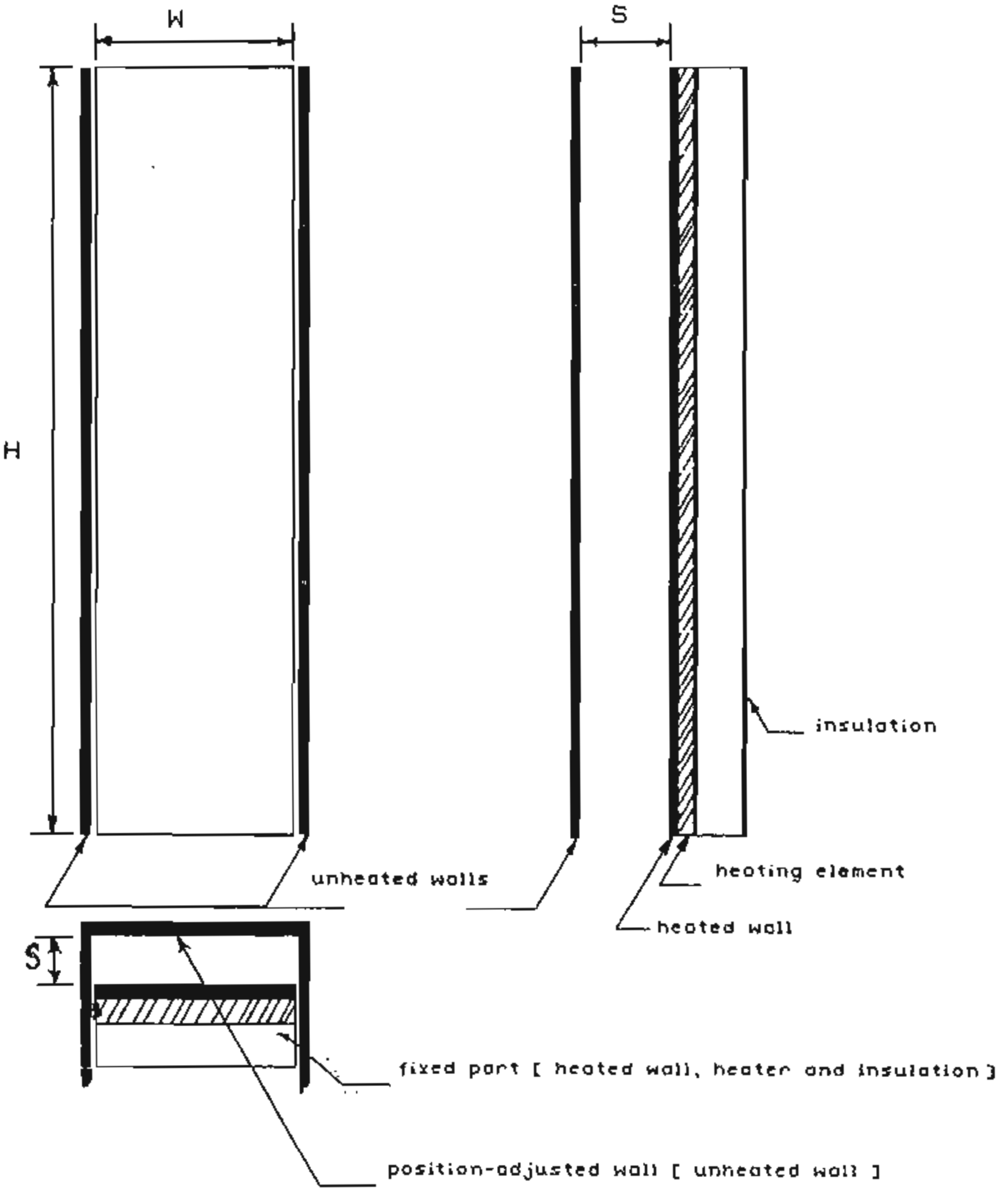


Fig.(1) Schematic description of the tested channel.

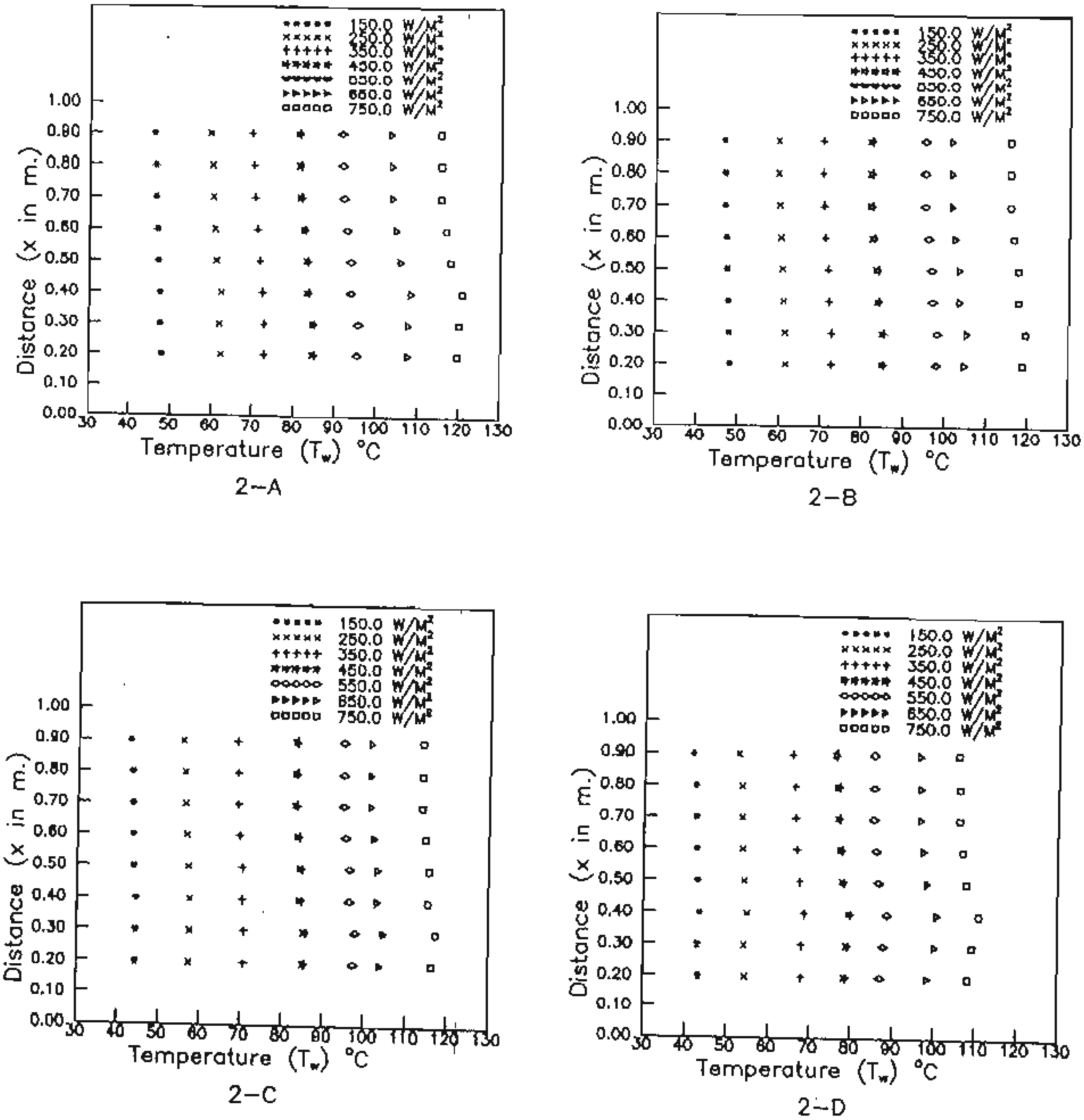


Fig.(2) The hot-wall temperature distribution in the direction of flow for different channel space ratio S/H.
2-A for S/H=0.01 2-B for S/H=0.02
2-C for S/H=0.03 2-D for S/H=0.04

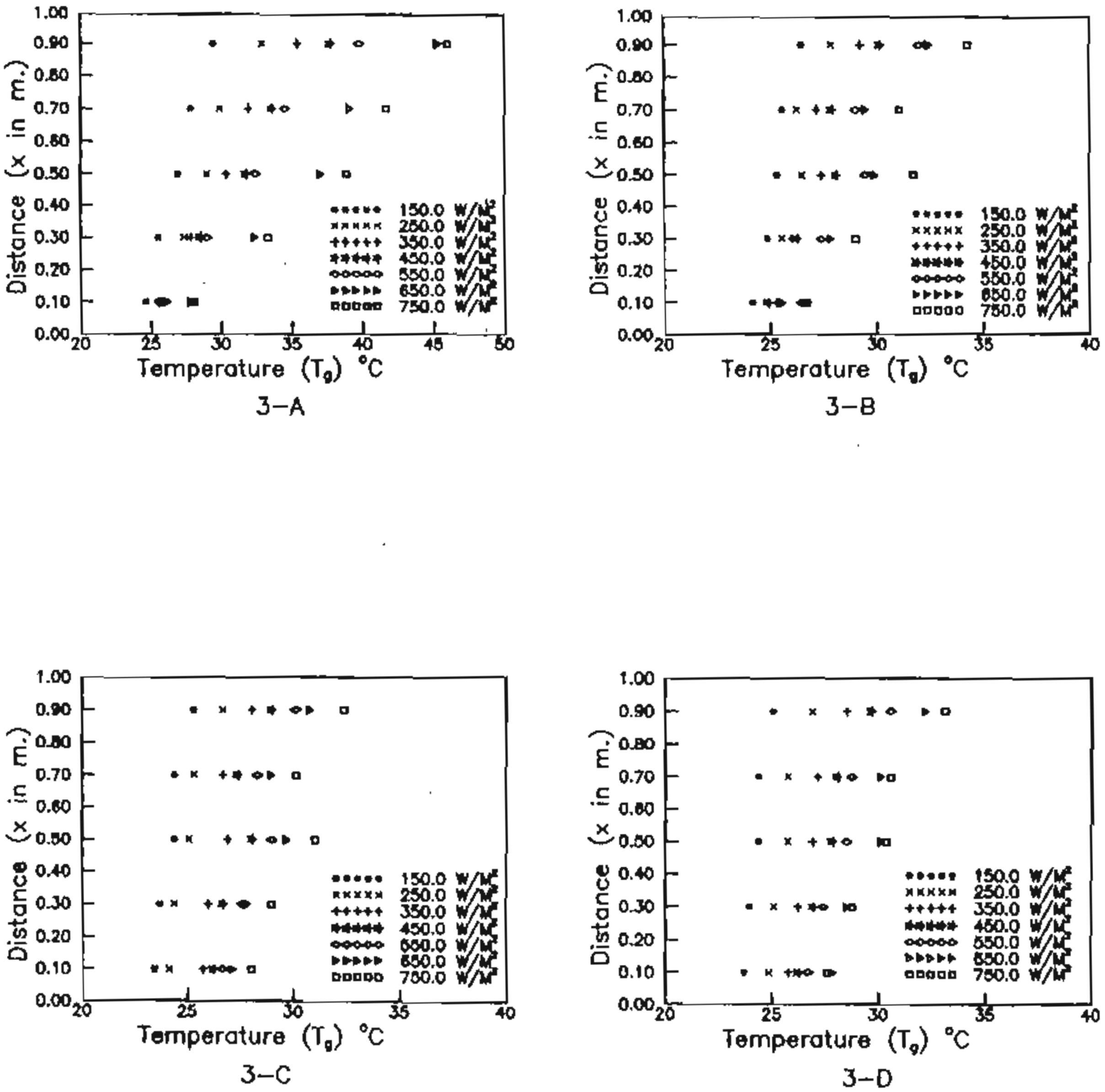
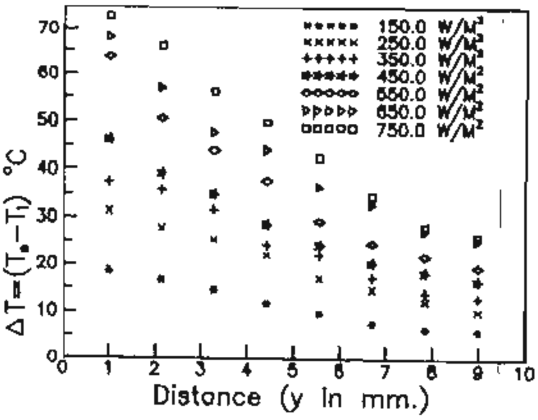
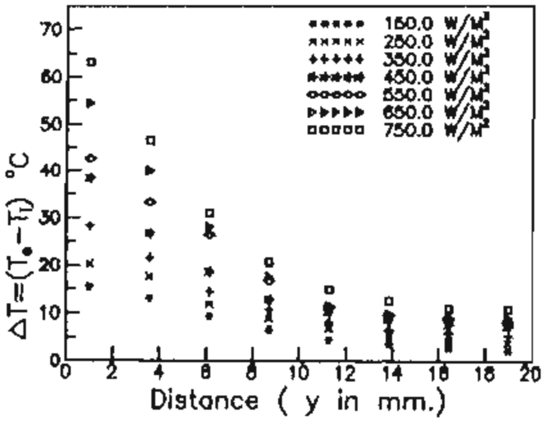


Fig.(3) The unheated-wall temperature in the direction of the flow at different values of channel space ratio (S/H).
3-A for S/H=0.01 3-B for S/H=0.02
3-C for S/H=0.03 3-D for S/H=0.04



4—A



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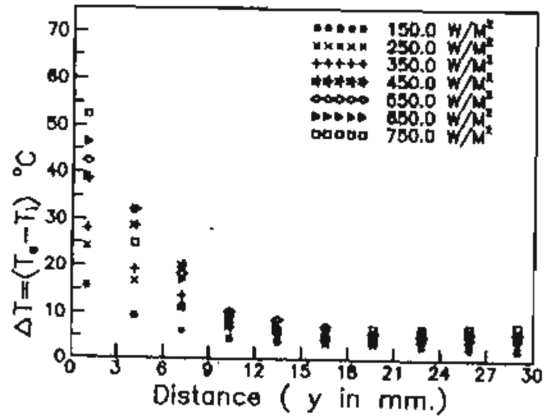

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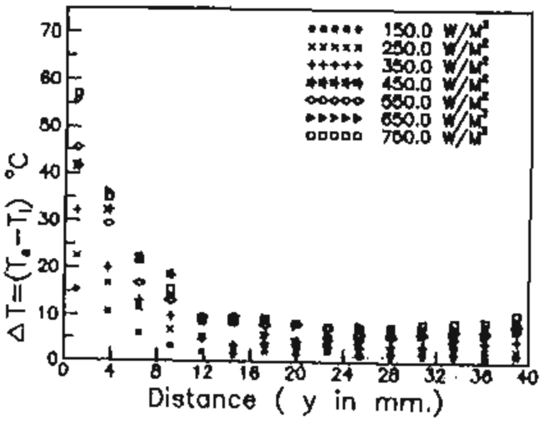


4—D

Fig.(4) The effect of heat flux on the difference between the temperature of air at exit and inlet for different channel space ratios (S/H).
1—A for S/H=0.01 1—B for S/H=0.02
1—C for S/H=0.03 1—D for S/H=0.04

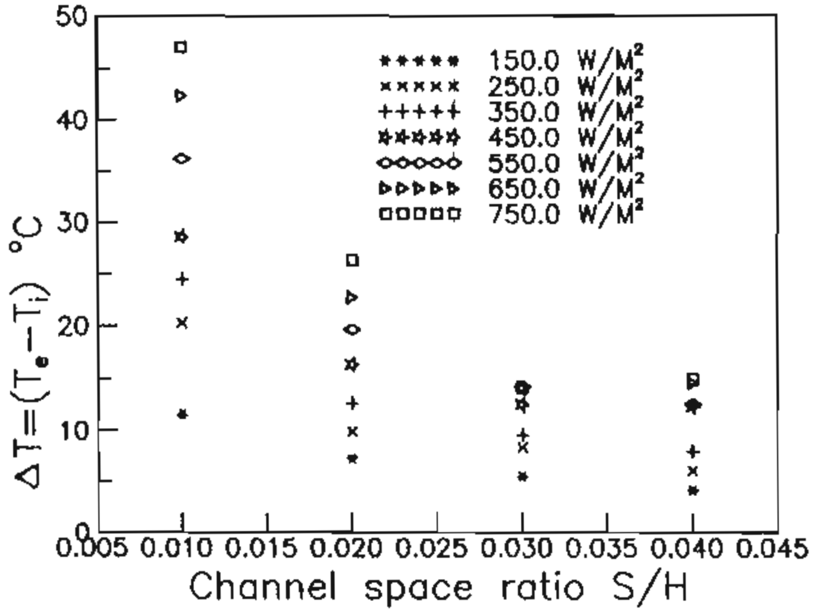


Fig.(5) The effect of the channel space ratio on the difference between the temperature of air at inlet and exit.

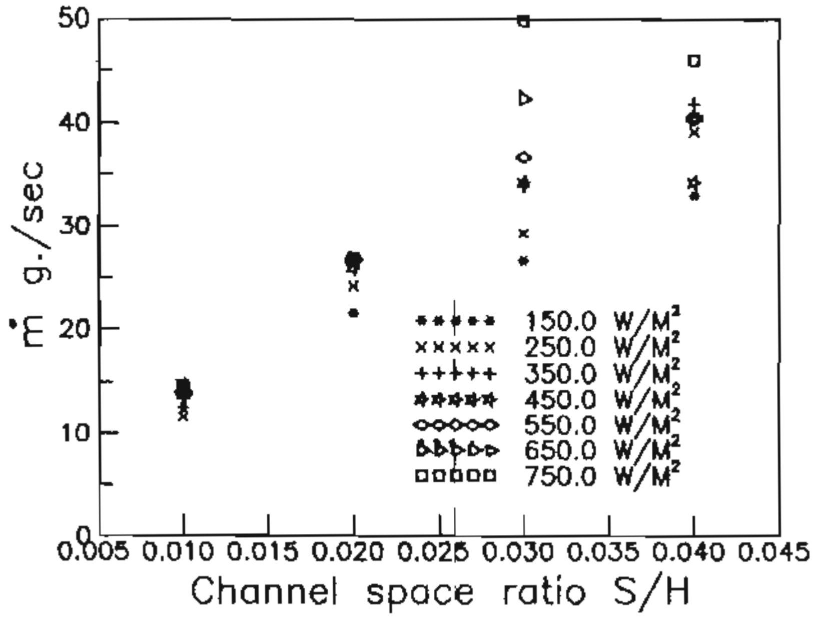
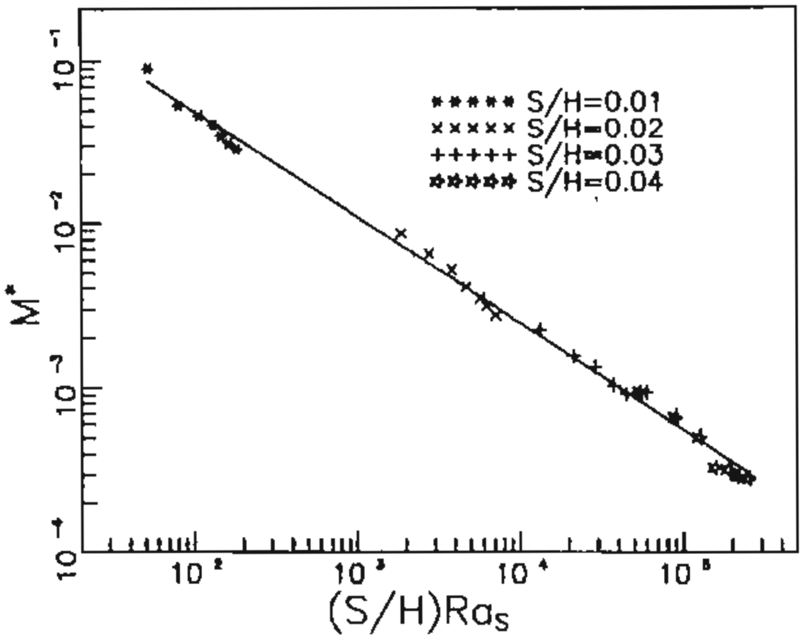
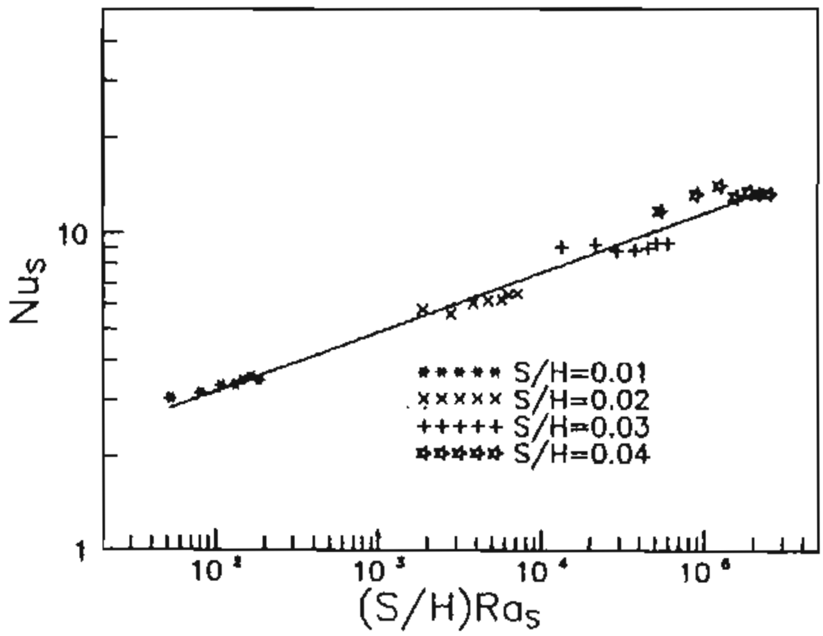


Fig.(6) The effect of the channel space ratio on the air mass flow rate.



Fig(7) Dimensionless Mass flow rate against Rayleigh number for different space ratios and their correlation.



Fig(8) Nusselt number against Rayleigh number for different channel space ratios and their correlation.

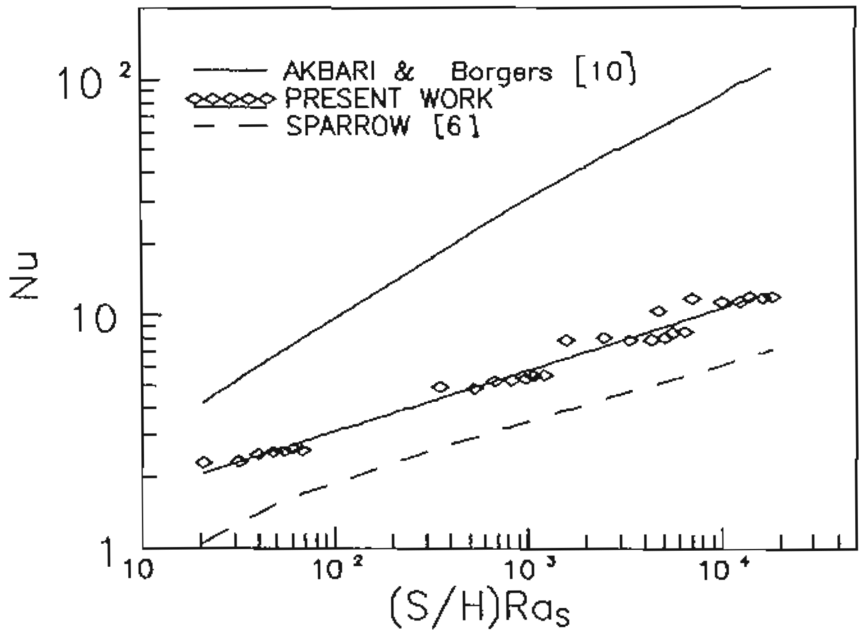


Fig.(9) Coparison between present and previous works (Nusselt number against (S/H)Ra_S)

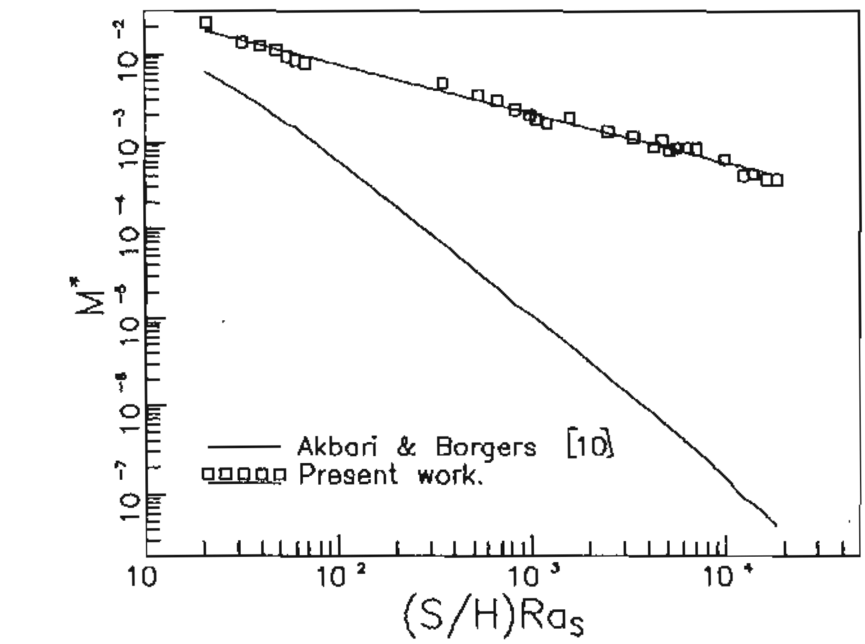


Fig.(10) Coparison between present and previous works (dimensionless mass flow rate against (S/H)Ra_S)