An Experimental Study on Heat Transfer Characteristics of a Thermal Diode Type Enclosure with a Guide Vane

Sukhyun Kim* and Young-Keun Jang**

Key words: Thermal diode, Heat source, Heat sink, Guide vane, Dimensionless channel depth, Driving force

Abstract

An experimental study for free convective heat transfer in a thermal diode type enclosure is presented. The thermal diode is a device which allows heat to be transferred in one direction by convection due to density difference of the fluid, and consists of a rectangular-parallelogrammic enclosure with a guide vane. It is used as heat collection system of solar energy due to its simple construction and low cost. Experimental parameters were guide vane thickness, the inclination angles of the parallelogrammic enclosure, and the lengths of the rectangular enclosure part. The parameter range of the flux Rayleigh numbers was $2.4\times10^8 \sim 9.8\times 10^8$. The heat transfer rate of this system was shown 10~47% higher than that of other earlier research results without the guide vane. The correlation for fixed $\phi=60°$ was obtained, $\mathrm{Nu}=0.0037(\mathrm{Ra}^*)^{0.429}(d^*)^{0.050}(Lr/H)^{0.0415}$.

Nomenclature

A : aspect ratio, Lr/H
C_p : specific heat [kJ/kg℃]
D : datum depth of flow channel [m]
d_x : actual depth of flow channel [m]
d^* : dimensionless channel depth, d_x/D
g : gravitational acceleration [m/s^2]
H : height of heat source and sink plate [m]
k : thermal conductivity [W/m℃]
L : length of the composed enclosure [m]
Lp : length of the parallelogrammic part [m]
Lr : length of the rectangular part [m]
Nu : overall Nusselt number, Eq. (1)
Pr : Prandtl number, Eq. (3)
q'' : input heat flux [W/m^2]
Ra* : Rayleigh number based on heat flux, Eq. (2)

Greek symbols

α : thermal diffusivity [m^2/s]
β : coefficient of thermal expansion [1/K]
ΔT : temperature difference between heat source and sink [℃]

* School of Mechanical Engineering, Kookmin University, Seoul 136-702, Korea
** Graduate School of Automotive Engineering, Kookmin University, Seoul 136-702, Korea

θ : inclination angle in radian
μ : viscosity [Ns/m^2]
ν : kinematic viscosity [m^2/s]
ϕ : inclination angle [°]

1. Introduction

Free convective heat transfer in enclosures has received increasing attention in recent years, due to the importance of such a device in many diverse applications, such as home heating, solar collectors, cryogenic storage, thermal insulation, nuclear reactor design and furnace design.

A thermal diode is a device which allows heat to be transferred in one direction by convection due to density difference of fluid, and blocks heat flow in the opposite direction, and is composed rectangular-parallelogrammic enclosure with a guide vane. Important examples of the thermal diode type enclosure geometry are the so-called thermal diode wall[1,2] and solar wall.

A number of investigators have studied the free convective heat transfer in various types of enclosure.[3-7] The works thus far, however, appear almost to be restricted to relatively simple rectangular, cylindrical and spherical enclosures. Dropkin and Somerscales[8] performed an experimental study of a free convective heat transfer in a liquid between two parellel plates at various inclination angles. They determined a power-law form correlation between Nusselt, Rayleigh and Prandtl numbers. Free convective heat transfer through inclined fluid layers of high aspect ratio was investigated by Hollands et al.,[9] who covered Rayleigh numbers up to 10^5 and inclination angles from 0 to 70 deg measured from the horizontal. Anold et al.[10] studied the free convective heat transfer in a rectangular enclosure of various aspect ratios with Rayleigh numbers ranging up to 10^6. They claimed that the variation of Nusselt number with inclination angle showed a local minimum Nu between 30 and 70 deg, while Nu had a local maximum at 0 deg inclination angle. Turner and Flack[11] investigated experimentally the free convection heat transfer between an isothermal heated vertical wall and a concentrated cooling strip on the opposing wall of rectangular enclosures with adiabatic top and bottom plates. The aspect ratio of the enclosure and the size and location of the cooling strip were parametrically varied for Grashof numbers from 5×10^6 to 9×10^6. They found that the effects of the aspect ratio from 0.5 to 2.0, on Nu are negligible for $Lr/H=0.25$.

Seki et al.[12] performed an experimental study for free convective heat transfer across a parallelogrammic enclosure with the various inclined angles for the parameter range of Ra= $3.4\times10^4-8.6\times10^7$, Pr=0.70−480, and various inclination angles. They found that the heat transfer coefficients for $\phi=-70$ deg were decreased by the factor of 0.05 compared with those for $\phi=0$ deg. Nakamura and Asako[13] also investigated the free convective heat transfer in a parallelogrammic enclosure filled with air both analytically and experimentally. They reported that their analytical results are in good agreement with the experimental ones for the case of one vertical side uniformly cooled, the opposing vertical side uniformly heated, and linear temperature top and bottom inclined.

The foregoing literatures give only a limited information on the heat transfer characteristics of the composed rectangular-parallelogrammic enclosure geometries (thermal diode type enclosure).

The objective of this paper is to provide with the experimental data for an experimental study of free convective heat transfer in a thermal diode type enclosure with a guide vane. The enclosure is a two dimensional composed rectangular-parallelogrammic one with two vertical side isothermally heated and cooled, and the rectangular-parallelogrammic adiabatic upper and lower walls. The test rig will use air as a working fluid. In particular, the effect of the inclination angle and heat flux on the heat transfer characteristics of the composed rectangular-parallelogrammic enclosure was extensively investigated.

2. Experimental apparatus and procedures

A schematic diagram of the experimental apparatus is shown in Fig. 1. The apparatus is constructed for measuring the heat flux in the test section, the details of which is explained in Fig. 2. It consists of two vertical copper plates 1 and 3, each 400 mm long and 800 mm wide, and the composed rectangular-parallelogrammic enclosure, made of acrylic material, between the two vertical copper plates.

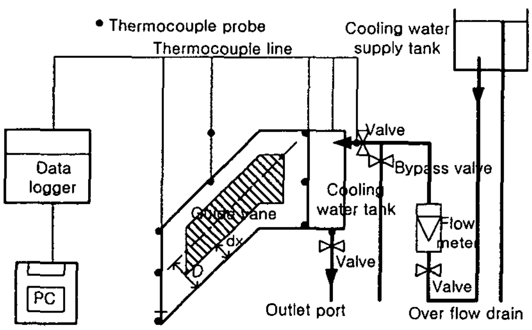
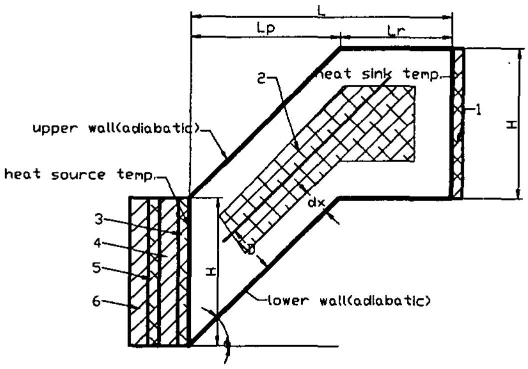


Fig. 1 Schematic diagram of the experimental apparatus.



1. Copper plate for heat sink
2. Guide vane
3. Copper plate for heat source
4. Main heater
5. Asbestos
6. Auxiliary heater

Fig. 2 A cross sectional view and components of the thermal diode.

The width of the parallelogrammic enclosure, Lp, is 300 mm giving an aspect ratio $Lp/H=$ 0.75. The lower vertical plate 3 is made of a 6 mm thick copper plate and was heated by two foil-type heaters. Each heater can be individually controlled so that a desired condition of isothermal wall obtained. The other plate 1 is cooled by temperature controlled water. Asbestos insulating gaskets are plased between the copper and acryl plates to reduce the thermal conduction.

The power to the main heaters was obtained by measuring the electric current and voltage of the heating circuit. The power to the main heaters was increased stepwise to a prescribed temperature. It took 7~10 hours to reach a steady state.

Surface temperature is measured from 18 T-type (copper-constantan) thermocouples, placed underneath the front face. The output of the thermocouples was carefully calibrated and continuously recorded on Yokogawa multiplex recorder (DA2500). The obtained temperature difference between the two plates was up to 37 ℃. the temperature variations of the plates were limited with in 0.3 and 0.6℃ for the cold and hot sides, respectively.

3. Results and discussion

The present study is concerned with the experiments of free convective heat transfer across the thermal diode type enclosure with the adiabatic rectangular-parallelogrammic upper and lower walls. Guided by a dimensional analysis, the relavant dimensionless clusters were found as

$$q''L/(\Delta Tk),\ g\beta q''L^4/(\alpha\nu k),\ \mu C_p/k,\ Lr/H,\ \phi$$

For the sake of convenience, the Nusselt number, Nu, the Rayleigh number based on

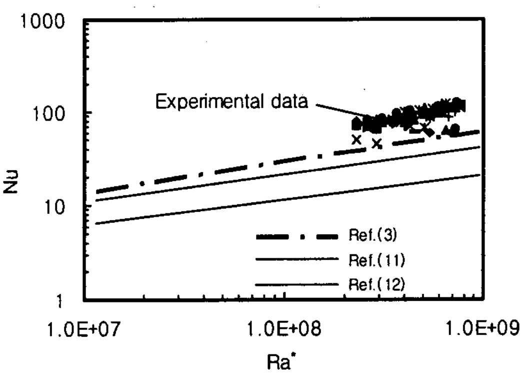


Fig. 3 Nusselt number variation compared with the low Rayleigh number predictions[3,11,12] and the present experiment.

heat flux, Ra*, and the Prandtl number, Pr, may be defined as follows.

$$\mathrm{Nu} = q''L/(\Delta T k) \quad (1)$$

$$\mathrm{Ra}^* = g\beta q''L^4/(\alpha \nu k) \quad (2)$$

$$\mathrm{Pr} = \mu C_p/k \quad (3)$$

More than 400 experiment were conducted for various parameter ranges. The dimensionless parameters will be evaluated at the reference temperature, which is the average temperature of the heated and cooled walls.

A comparison between the present data and those obtained by previous investigators, Eckert and Carlson,[3] Turner and Flack[11] and Seki et al.[12] is shown in Fig. 3. The Fig. 3 shows the relationship between the Nusselt numbers and Rayleigh numbers using air as working fluid. The previous results are for the rectangular only enclosure,[3] for the rectangular enclosure with concentrated energy source[11] and for the parrallelogrammic only enclosure.[12] The present experimental results indicate a good agreement with the linear extrapolation of the results in references 3, 11 and 12.

Fig. 4 shows the relationship between Nusselt number and inclination angle for d^*=0.5, Lr/H=0.5, and various numbers of Ra. The value of Nu increases gradually with ϕ and Ra. The driving force is increased by buoyant force from the heat source of lower vertical plate to the heat sink of upper vertical plate as increasing inclination angle.

Fig. 5 gives the effects of the inclination angle and Ra on Nu for d^*=1.0, Lr/H=0.5, and a various Ra without a guide vane. As can be seen, the variation of Nu with ϕ is very small and has a different characteristics from the Fig. 4, showing a minimum Nu near 45 deg. This behavior of Nu against ϕ is similar to previous results obtained by Seki et al.[12] using air as working fluid. This comparison between Figs. 4 and 5 is probably due to the following effects: (i) the heat transfer may

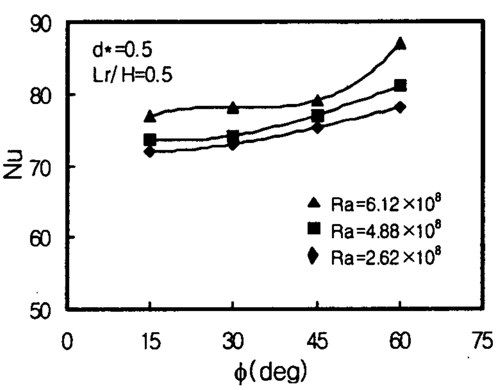


Fig. 4 Effects of the inclination angle on the Nusselt number.

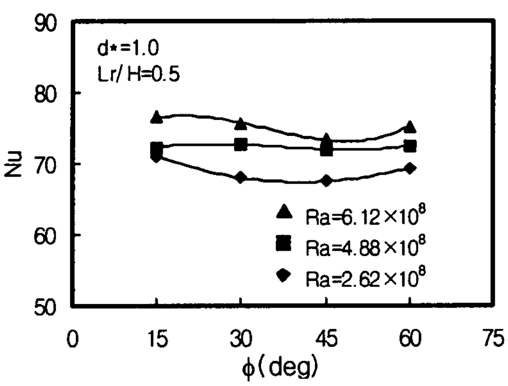


Fig. 5 Effects of the inclination angle on the Nusselt number without the guide vane.

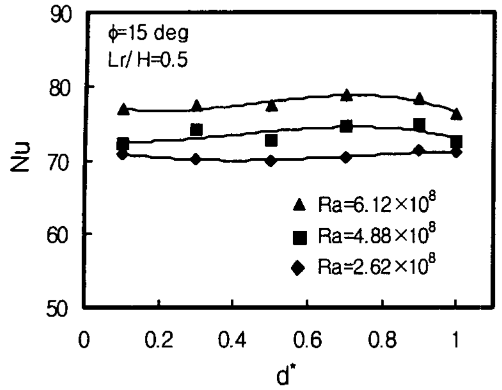


Fig. 6 Effects of the dimensionless channel depth on the Nusselt number.

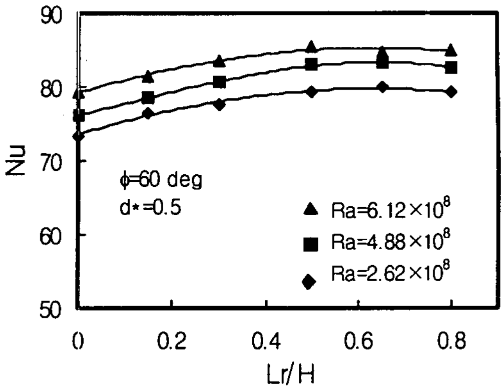


Fig. 8 Effects of the aspect ratio on the Nusselt number.

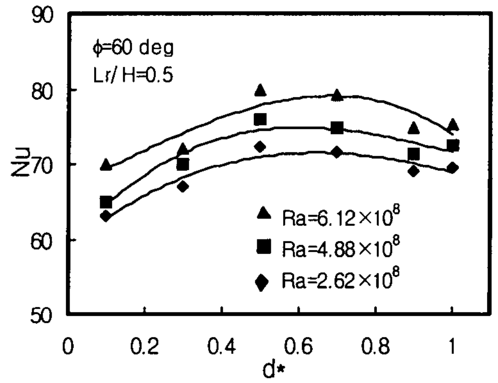


Fig. 7 Effects of the dimensionless channel depth on the Nusselt number.

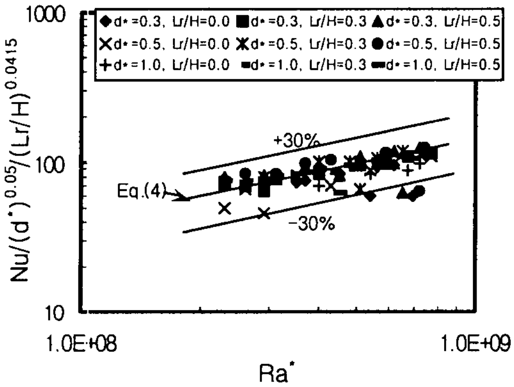


Fig. 9 Correlation of the experimental results for $\phi=60$ deg.

be promoted by the fact that the heated wall is below the cooled one with increasing ϕ; (ii) the guide vane with increasing ϕ may enhance the heat transfer across the enclosure.

The effects of the dimensionless channel depth on Nu with $Lr/H=0.5$ are shown in Figs. 6 and 7. The maximum Nusselt number occurred for values of $d^* \fallingdotseq 0.6$, which is more apparent in Fig. 7, with $\phi=60$ deg. This is due to the roll of the guide vane, which increases an energy moving density with increasing ϕ. Further experimental tests for the cases with $\phi=30$ and 45 deg also supported this explanation.

The effects of the aspect ration on the Nu are shown in Fig, 8. In this case, the inclination angle was 60 deg and the dimensionless channel depth was fixed at 0.5. In this figure, Nu increases slightly with increasing Lr/H, and it seems to reach its asymptotic at about $Lr/H=0.5$.

Fig. 9 gives all the data entered in Fig. 4 to 8 for $\phi=60$ deg, using the plots of $(Nu/(d^*)^{0.05}/(Lr/H)^{0.0415})$ versus Ra^*. All the dimensionless parameters are computed using properties evaluated at the mean temperature between the hot and cold walls. The overall correlation of the heat transfer rate was obtained by the least square technique,

$$Nu = 0.0037(Ra^*)^{0.429}(d^*)^{0.050}(Lr/H)^{0.0415} \quad (4)$$

for $\phi=60$ deg, $2.4\times10^8<\mathrm{Ra}^*<9.8\times10^8$, $\mathrm{Pr}=0.71$ and $Lp/H=0.75$.

The obtained equation correlates the data within a deviation of less than about ±30 percent.

4. Conclusions

This paper reports the results of an experimental study of free convective heat transfer between two plates across a thermal diode type enclosure with a guide vane and constant Lp/H. Rayleigh number was varied from 2.4×10^8 to 9.8×10^8, and the inclination angles, guide vane thickness and aspect ratio were changed accordingly. The results are summarized:

(1) The present experimental data for $d^*=1.0$ and $Lr/H=0.0$ are proved in a good agreement with those of the previous study.

(2) For $\phi=60$ deg and $Lr/H=0.5$, the Nusselt number has a maximum value.

(3) The optimum thickness of the guide vane is $d^*=0.6$ for $\phi=60$ deg and $Lr/H=0.5$.

(4) The parametric correlation for the ranges of the parameters investigated was found to be

$$\mathrm{Nu}=0.0037(\mathrm{Ra}^*)^{0.429}(d^*)^{0.050}(Lr/H)^{0.0415}$$

for $\phi=60$ deg, $2.4\times10^8<\mathrm{Ra}^*<9.8\times10^8$, $\mathrm{Pr}=0.71$ and $Lp/H=0.75$.

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