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Experimental Investigation of Heat Transfer in the Channel with Two Inclined Perforated Baffles

구멍이 있는 2개의 경사진 배플이 있는 채널에서의 열전달에 대한 실험적 연구

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주요 용어 : 구멍 수(Number of Holes), 사각 채널(Rectangular Channel), 두 개의 경사진 구멍이 있는 배플(Two Inclined Perforated Baffle), 열전달(Heat Transfer)

요 약 : 본 연구는 두 개의 경사진 다공 배플이 설치된 사각채널에서 국부 열전달향상 특성을 조사하였다. 채널은 19.8cm(W)x4cm(H)의 단면적을 가지며 형상계수는 4.95이며 수력직경은 6.66cm이다. 4종류의 배플을 취급하였다. 가열 시험부에 동일한 크기, 경사각, 구멍형태의 경사 배플을 설치하였다. 경사 배플은 모두 19.8 cm의 폭, 2.55cm x 2.55cm의 정 다이아몬드 형 구멍, 그리고 5o의 경사각을 갖는다. 레이놀즈 수 범위는 23,000에서 57,000까지이다. 배플의 구멍의 수가 열전달 향상에 중요한 역할을 하였으며 구멍이 3개 인 (baffle type II)가 가장 우수한 열전달 향상을 보였다.

Nomenclature

A : heat transfer area, m^2
C_p : specific heat of air, J/kg °C
D_h : hydraulic diameter of the channel, m
h : heat transfer coefficient, W/m^2 °C
H : test section height, m
L : test section length, m
$\dot{m}$: mass flow rate, kg/s
Nu : Nusselt number, hD_h/k
Pr : Prandtl number
$\dot{Q}$: heat transfer rate, W
Re : Reynolds number
St : Stanton number
T : temperature, °C
u : air velocity, m/s
W : test section width, m
x : distance from entrance of the heated test section, m

Greek symbols

κ : thermal conductivity of air, W/m °C
ρ : air density, kg/m^3

Subscripts

avr : average
b : bulk average
ra : channel average
ss : smooth channel without tapes

1. Introduction

There are several techniques available to enhance the heat transfer coefficient of gases in the internal cooling systems of gas turbine blades, air-cooled solar collectors, laser curtain seals, labyrinth shaft seals, compact heat exchangers, and microelectronics. The most commonly used

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technique for internal cooling enhancement is the placement of periodic ribs. Ribs are generally mounted on the heat transfer surface, which disturbs the boundary layer growth and enhances the heat transfer between the surface and the fluid.

Another popular heat transfer augmentation technique is impingement cooling that uses high velocity jets to cool the surface of interest. In addition to ribs and impingement, the third common technique is the use of internal flow swirls, tape twisters, or baffles. The twist insert and tape twister techniques create a significant amount of bulk flow disturbance, and the pressure drop penalties are much higher compared to the gain in heat transfer coefficient. Baffles also create bulk flow disturbance, but unlike tapes or swirls, baffles are discrete objects. Therefore, the flow disturbance created by baffles may be localized, but more intense. Usually the baffle plate is attached to the thermally active surface to augment heat transfer by providing additional fin-like surface area for heat transfer and better mixing.

In the past, experimental results were published with the baffle plates orthogonal to the flow direction[1,2]. Since those works are emphasized on baffles that directly block the flow, the pressure penalties are higher than the heat transfer improvements. However, it is possible to obtain enhanced heat transfer with comparably less frictional head loss by inserting inclined baffles in the flow path. Inclined baffles may be considered as a combination of ribs and channel inserts. These baffles are big enough to disturb the core flow, but like ribs, they are mounted on or near the heat transfer surface. Moreover, inclined perforated baffles contain holes, which facilitate jet impingement toward the heat transfer surface. Dutta and Dutta[3] first reported the enhancement of heat transfer with inclined solid and perforated baffles. In that study, the effects of baffle size, position, and orientation were studied for internal cooling heat transfer augmentation. They showed that the inclined perforated baffles works much better than the corresponding arrangement with solid baffles. In a recent experimental work, Dutta and Hossain[4] discussed experimental heat transfer results with two inclined perforated baffles where the first baffle is mounted on the heated top surface and the second baffle is either fixed to the insulated bottom surface or to the heated top surface. They reported that the overall heat transfer coefficients were much higher with two inclined baffles than those with a single baffle placed in the same channel. The perforated baffle had the circular holes of small scale diameter of 1.07cm and hole number of up to 132. Our previous work[5] has numerically dealt with the effect of one baffle on Nusselt numbers and friction factors. In the present work experimental analyses of effects of hole number on heat transfer coefficients in the rectangular channel with two inclined perforated baffles were

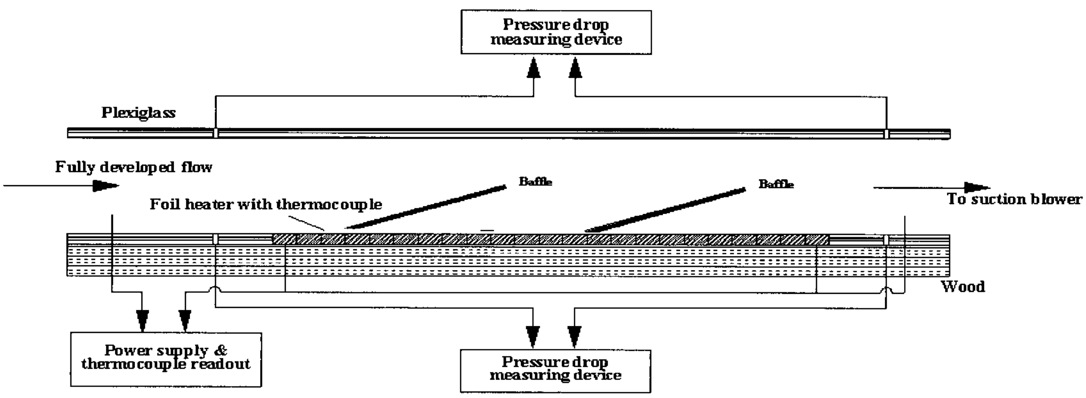

Fig. 1 Schematics of test apparatus

performed. The baffles were mounted on the heated bottom wall and the baffle inclination angle was 5°. The drilled holes consisted of large scale square diamond type having one side length of 2.54 cm and the number of holes were up to 9.

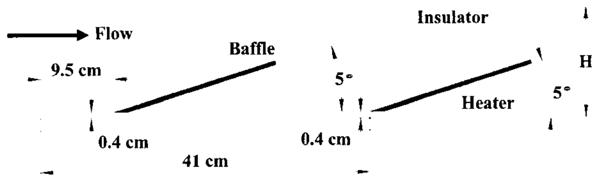

Fig. 2 Position of baffle in the test section

2. Experimental facility

Fig. 1 shows a schematic diagram of the test apparatus. A suction-mode blower is used to draw air at room temperature through flow straighteners, passing through a long unheated straight rectangular channel with a cross-section of 19.8 cm(W) x 4 cm(H) and a length of 171.78 cm, finally through the heated test section with Lo =71.2 cm. The channel has an aspect ratio of 4.95, and hydraulic diameter of D_h = 6.66 cm. The entrance section is long enough to ensure a hydrodynamically fully developed flow just before the heated test channel. The left, right and upper sides of the channel are made of 0.55-cm-thick plexiglass plates, and the bottom side is made of 5-cm-thick wood plate. A total of twenty three iso-flux stainless steel foil heaters of the same size of 19.8 cm x 3.0 cm x 0.01 cm are mounted on the bottom surface of the test section. These foil heaters are aligned perpendicular to the flow direction and connected to a voltage controller to provide uniform heat flux. Twenty three copper constantan thermocouples are laid along the heated test section centerline and glued at each strip of the foil heater to measure the wall temperatures. Moreover, one thermocouple is placed at the inlet (10 cm upstream of the heated test section) and ten others are transversed vertically at the outlet (9.2 cm downstream of the heated test section) to measure the inlet and outlet bulk air temperatures, respectively. Details of the baffle position in the test section are shown in Fig. 2. All baffle types are mounted on the heated wall have a constant inclination angle of five degrees and a gap of 0.4cm between heated surface and the baffle is maintained to avoid flow stagnation. In all configurations, the first baffle and the second baffle placed at the position of 9.5 cm and 41 cm downstream from the start of the heated test section. Leading edge of the baffle is kept sharp to reduce possible flow disturbance by the protruding edge. In this study, all baffle types have the same overall size of length(L) of 23.2 cm, width of 19.8 cm, and thickness of 0.5 cm, but different number of holes. The schematic views of the baffle types used in this investigation is shown in Fig. 3. Four different types such as the solid baffle (baffle type I), the 3-holed baffle (baffle type II),

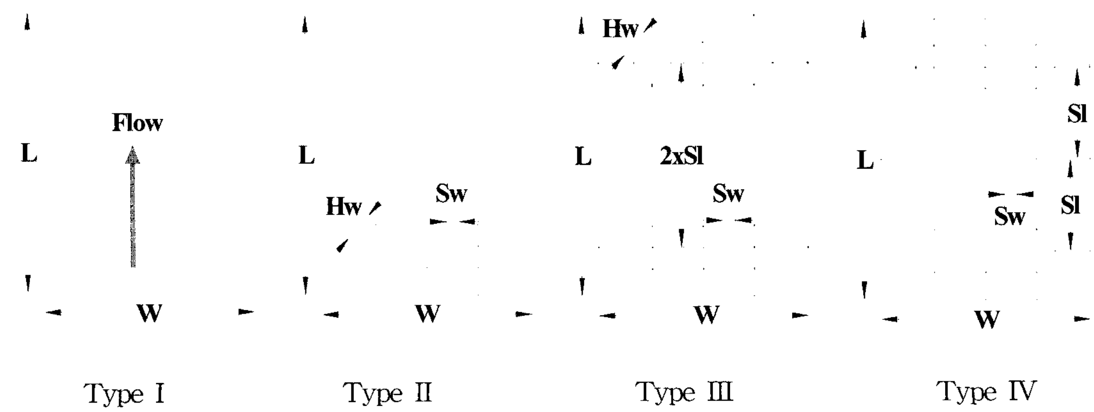

Fig. 3 Baffle plate configuration

the 6-holed baffle (baffle type III), and the 9-holed baffle (baffle type IV) were dealt with, respectively. These square diamond holes of width, H_w = 2.55cm, transverse gap, S_w = 1.2cm, and longitudinal gap S_l = 7.6cm were manufactured. All of the thermocouples used in the experiment are carefully calibrated to an accuracy of 0.5℃. The experimental conditions were controlled with a air-conditioning system to maintain the room temperature of 22 - 25℃ and relative humidity of 40 - 50% shown in Table 1. The radiation heat loss was evaluated using a diffusive gray-surface net work[6] and was less than 0.5% of the total power input. It was because the surface of the foil stainless plates were highly polished to minimize emissivity. The flow rate was measured with the flow meter in the circular tube at the upper stream of test section.

The experimental uncertainties are estimated using the procedure outlined by Kline and McClintock[7]. The variables measured in this experiment are wall temperature, air temperature, velocity. It is found that the uncertainties for Reynolds number and Nusselt number are about ±2.5% and ±7.8%, respectively as shown in Table 2.

Table 1 Experimental conditions

Room temperature	Relative humidity	Working fluid	Reynolds number
22 - 25 ℃	40 - 50%	air	23,000-57,000

Table 2 Experimental uncertainties(Re=23,000).

Parameters	Uncertainties
Thermophysical property of air	±2.0%
Hydraulic diameter, D_h	±0.5%
Bulk velocity, u_b	±1.5%
Pressure difference, ΔP	±9.2%
Reynolds number, Re	±2.5%
Friction factor, f	±9.5%
Temperature, T	±0.5℃
Heat transfer coefficient, h	±7.5%
Nusselt number, Nu	±7.8%

3. Data reduction

The local heat transfer coefficient is calculated from the net heat transfer rate per unit surface area exposed to the cooling air, the local wall temperature (T_w), and the local bulk mean air temperature (T_b) as follows :

$$h = \dot{Q}/[A(T_w - T_b)] \tag{1}$$

where A is the total heat transfer area of the heated wall. The heat transfer rate ($\dot{Q}$) is defined as:

$$\dot{Q} = \dot{m}c_p(T_{b2} - T_{b1}) \tag{2}$$

where T_{b1} and T_{b2} represent the fluid bulk temperatures at the entrance and exit, respectively. The local Nusselt number is defined using the local heat transfer coefficient and the hydraulic diameter D_h for the rectangular channel:

$$Nu = hD_h/k \tag{3}$$

The average Stanton number is normalized by the Stanton number for fully developed turbulent flow in a smooth circular tube correlated by McAdams/Dittus-Boelter[8] as:

$$St_{avr}/St_{ss} = (Nu_{avr}/Re\,\mathrm{Pr})/(0.023Re^{0.8}\mathrm{Pr}^{0.4}/Re\,\mathrm{Pr}) \tag{4}$$

And the average Stanton number is given by:

$$St_{avr} = Nu_{avr}/(Re.\mathrm{Pr}) \tag{5}$$

The Reynolds number was calculated on the basis of channel average velocity and channel hydraulic diameter. The channel average velocity was obtained from the flow rate of the circular tube at downstream of test section.

4. Results and discussion

The heat transfer coefficient calculations are done on the heated bottom surface, while the top and side surfaces are unheated. All heat transfer results are presented in terms of the heat transfer coefficients (h) along the channel centerline shown in equation (1).

Fig. 4 represents the centerline local heat transfer coefficient distributions in the channels with solid baffle (baffle type I) as a function of the dimensionless axial location of x/D_h. The local heat transfer coefficient, h is comparatively high at the entrance of heating section due to the starting of the thermal boundary layer development. The first h peaks take place at x/D_h=2.2. This occurs because the gap between the baffle leading edge and the heating wall (bottom wall) creates the sudden contraction effect. The downward trend of h continues until the end of the baffle. Sharply decreasing on the local heat transfer coefficient immediately after the end of the first baffle occurs due to recirculation effect, and then the h suddenly increases while flow passing through the leading edge of the second baffle.

Fig. 5 represents the distribution of h for baffle type Ⅱ having 3 holes. The heat transfer coefficient peaks for baffle type Ⅱ are higher than those of the baffle type I (solid baffle). This is due to the jet impingement effect that can improve flow momentum. Another heat transfer behavior distinction between two baffle types is that downward trend of the local heat transfer coefficient of the baffle type Ⅱ is more moderate than that of the baffle type I (solid baffle) because less flow recirculation occurs on the downstream of baffle type Ⅱ.

Fig. 6 shows the centerline local convective heat transfer distributions of the channel for baffle type Ⅲ as a function of the dimensionless axial location x/D_h. This figure shows that the h distributions for baffle type Ⅲ at the start of the heated section are lower than those of the baffle type I and Ⅱ. It is because the boundary layer disturbances are weaker. The holes near trailing end of baffle Ⅲ create the secondary peaks of h immediately after the end of the baffle. It is attributed to the fact that the divergent orientation of the baffle creates higher inclination of impinging jet flow in the impact zone at this location. This leads less heat transfer enhancement because the transport (velocity and turbulence) characteristics in the impact zone depend on the inclination of impinging jet flow. The charac- teristics of h distributions in terms of the second baffle have the same behaviour with the first baffle.

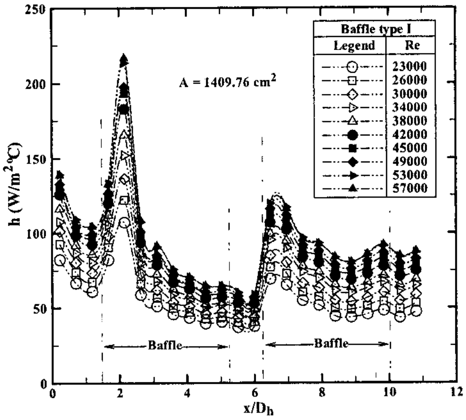

Fig. 4 Local heat transfer coefficient for type I

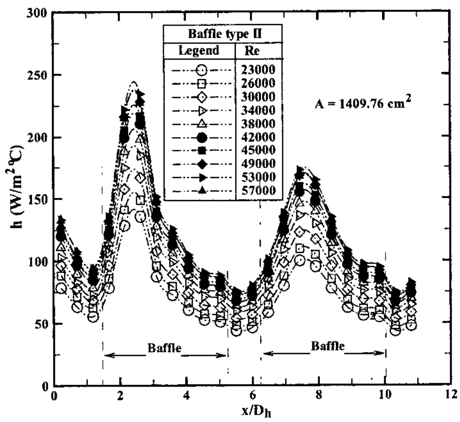

Fig. 5 Local heat transfer coefficient for type Ⅱ

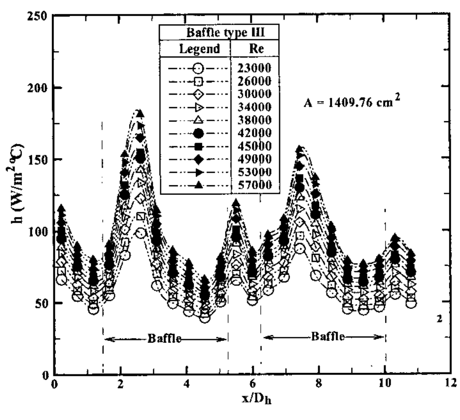

Fig. 6 Local heat transfer coefficient for type III

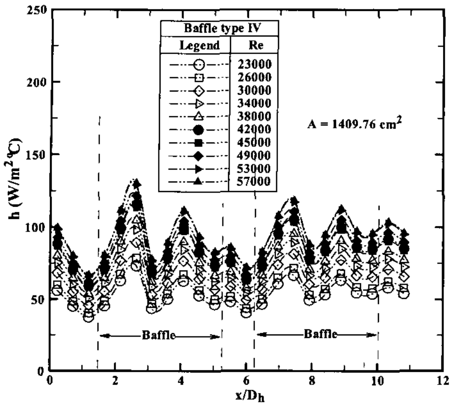

Fig. 7 Local heat transfer coefficient for type IV

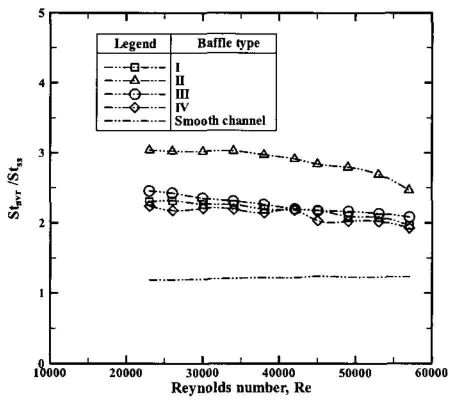

Fig. 8 Normalized Stanton number

Fig. 7 presents the local convective heat transfer coefficient, h at the centerline of the channel for baffle type IV. This figure shows that there are three peaks of h in the channel. It is due to the multiple impingement effect. This figure also shows that the h peaks declines along the dimensionless axial location x/D_h. The declining of the h peaks is due to a divergent orientation of the baffle.

Fig. 8 shows the average heat transfer ratio (St_{avr}/St_{ss}) for different baffle type at different Reynolds number. The Stanton number ratio essentially indicated the amount of enhancement in heat transfer obtained by the flow disturbance promoters over the smooth circular pipe. The experimental procedure was validated by running an experiment through the rectangular setup without any baffle. For this smooth rectangular channel, the fully developed Stanton number ratio (St/St_{ss}) was found between 1.2 to 1.3. There are two reasons for higher than unity: First, the centerline surface temperature is used in the heat transfer coefficient calculation instead of the span averaged values, and second, in higher aspect ratio channels the Stanton number is expected to be greater than that for a circular pipe[8]. The rectangular channel with the baffle type II inserted has the highest average heat transfer rate. An interesting observation is that St_{avr}/St_{ss} isn't highest at the most number of holes. It may be deserved that a more number of holes increases the cross flow by spent jets and reduces impingement effect. Type IV has a relatively lower heat transfer enhancement compared with three other baffle types. It can be argued that the impingement effect is weaker in the channel with more perforated baffles. The heat transfer ratio(St_{avr}/St_{ss}) decreases with increasing Reynolds number. It is because the mixing turbulence in the channel without baffles increases more rapidly with increasing Reynolds number than in the channel with baffles.

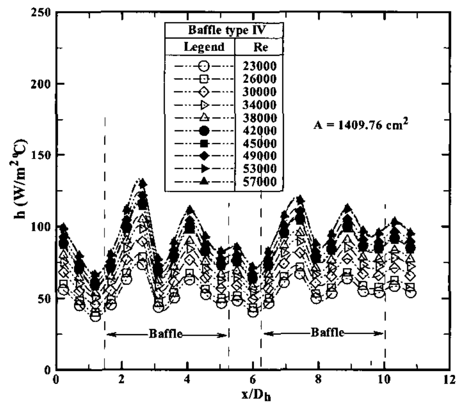

Fig. 7 Local heat transfer coefficient for type IV

5. Conclusions

This paper investigated heat transfer coefficient and friction characteristics in rectangular channel with two inclined baffle and the flow Reynolds number was varied between 23,000 and 57,000. Four samples having different number of holes in the baffle were tested. Listed below are major findings:

1) The rectangular channel with the baffle type II showed the highest heat transfer coefficient, h. The additional holes of baffle didn't improve the increase in heat transfer enhancement because the divergent orientation of the baffle created higher inclination of impingement jet flow in the impact zone.
2) The convective heat transfer coefficient, h for baffle type III at the start of the heated section are lower than those of the baffle type I and II. It is because the boundary layer disturbances are weaker.
3) The heat transfer ratio (St_{avr}/St_{ss}) decreases with increasing Reynolds number. It is because the mixing turbulence in the channel without baffles increases more rapidly with increasing Reynolds number than in the channel with baffles.

Acknowledgements

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