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A Numerical Study on Heat Transfer Enhancement and Pressure drop Decrease of Heat Exchanger by Setting Inserted Plates in Duct

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Abstract

This present paper attempts to numerically estimate heat transfer enhancement and pressure drop decrease on a duct flow system of heat exchanger. Inserted plates inside the duct flow are proposed in order to enhance heat transfer coefficient, however the inserted plates cause a significant pressure drop. In order to decrease the pressure drop, the inserted plates with slits were employed. Numerical calculations of two-dimensional, laminar, and steady state conditions of duct flow with and without inserted plates were carried out. Air, as a heated media, with range of velocities 0.5m/s – 1.5 m/s and ambient temperature 20^{0}C was heated by using 50^{0}C isothermal duct wall. The calculation results show that the setting of inserted plates enhances the heat transfer coefficient significantly but pressure drop also increase. The presence of slits hole in the inserted plates decreases the pressure drop significantly but the heat transfer coefficient decrease slightly. The calculated streamlines, isotherms, vector velocity, local and average heat transfer coefficient, total heat transfer, and required fan power (pressure drop) are presented.

Key Words: Heat Transfer Enhancement, Inserted Plate with Slits, Pressure drop Decreasing

Introduction

Heat exchanger is one of the main components of thermal engineering applications and has been extensively studied in order to enhance the performance. In the past, the emphasis was on experimental work due to absence of today's computational power [1]. The nature of experimental work enabled investigators to include in their studies only a few heat exchanger geometries, slightly varying geometry parameters. Furthermore, the flow conditions were limited by available experimental setup. Compare to numerical analysis, these disadvantages of experimental work did not allow investigators to explore a wide range of parameters in order to find an optimal geometry [2]. In the last 20 years, numerical approaches and methods, originally developed for aerospace applications, have been increasingly employed to simulated process in heat exchanger in order to find new and higher performance designs for emerging technological needs.

In this present paper, the numerical analysis was used to study the duct flow heat exchanger. Instead of heat transfer surface increasing technique, inserted plates have been proposed in order to enhance the heat transfer coefficient inside flow regime. However the presences of the inserted plates significantly increase the pressure drop. In order to decrease the pressure drop the slit holes on the inserted plates were employed but, the presences of the slits cause decrease the heat transfer coefficient. The optimum combinations of inserted plated and slits were investigated.

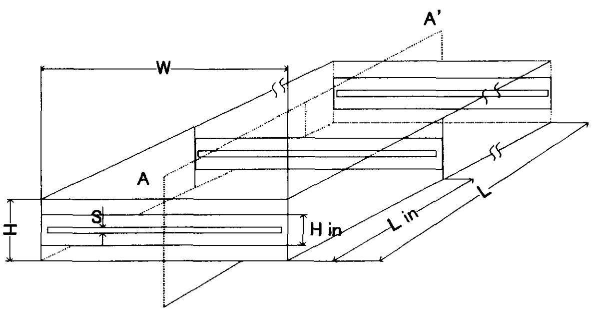


Figure 1 Computational domain (View A-A')

As the computational domain a duct flow heat exchanger consist of isothermal hotter surfaces and cooled by air flowing is depicted in Figure 1. Dimensions of the duct are L=W=380mm, L_{in}=50mm, H=30mm, H_{in}=24mm, and S=1.5mm or 2 mm. Heat exchanger with 5 different arrangements of inserted plates and slit holes were studied.

Case 1 there is no inserted plate, Case 2 inline inserted plates without slit holes are employed, Case 3 inline inserted plates with 1 slit hole are employed, Case 4 inline inserted plates with 2 slit holes are employed, and Case 5 staggered inserted plates with 1 slit hole are employed. The slit holes positions on the inserted plates for case 3, case 4, and case 5 are depicted in Figure 2.

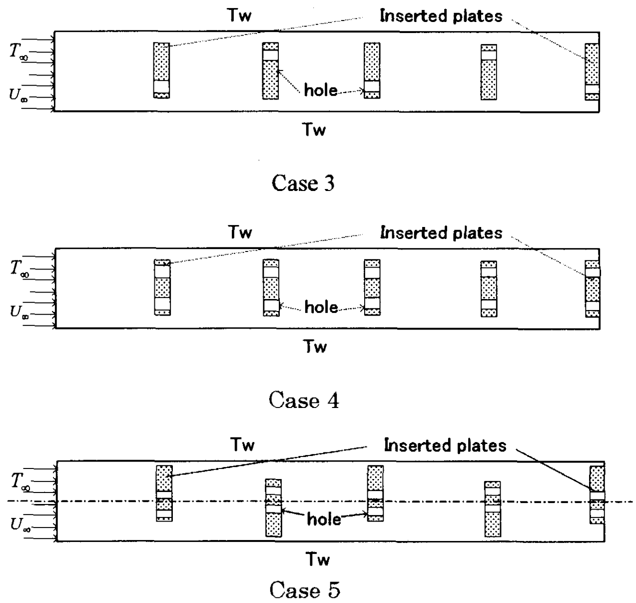


Figure 2 Proposed inserted plates with slit holes

Mathematical Model

The flow is assumed to be steady state, laminar, and two-dimensional. The compressibility, dissipations, buoyancy force, and radiation are negligible. All of the thermal properties are constant. Based on these assumptions, the governing equations are:

Continuity equation $$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (1)$$

X and Y momentum equations

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\upsilon\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\upsilon\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

Energy equation $$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{k}{\rho c_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

Boundary Conditions

The wall surfaces: $u=v=0$, $T=T_w$, $\frac{\partial p}{\partial n}=0$ (5)

The inlet: $u=U_\infty$, $v=0$, $T=T_\infty$, and $\frac{\partial p}{\partial n}=0$ (6)

The outlet: $\frac{\partial u}{\partial x}=0$, $\frac{\partial v}{\partial x}=0$, $\frac{\partial T}{\partial x}=0$ and $p=p_0$ (7)

Inserted plates regions and surfaces $u=v=0$ (8)

Calculations Methods

All of the governing equations are discretized based on control volume approach on staggered grid system. In order to handle the convective-diffusion, the power law scheme is employed. The sets of discreatization linear equations are solved by using line-by-line method. To couple the pressure distributions and velocities the SIMPLE algorithm is employed. Iterations process will be stopped if the continuity equation is satisfied (total mass residual compare to flow rate $\leq 10^{-3}$ or recurred is reached).

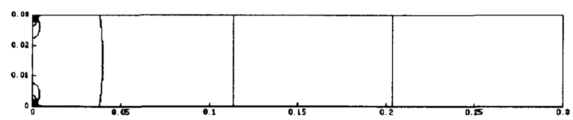
Result and Discussion

As the comparison parameters, required fan power, local and average heat transfer coefficients, and total heat transferred are used. The local and average heat transfer coefficients are calculated by using equation (9).

$$h=-k\left.\frac{\partial T}{\partial y}\right|_{wall} \quad \text{and} \quad \bar{h}=\frac{1}{L}\int_0^L h.dx \tag{9}$$

The calculation results of pressure, streamlines, velocity vector and Temperature are presented in Figure 2.

Case 1: No inserted plated



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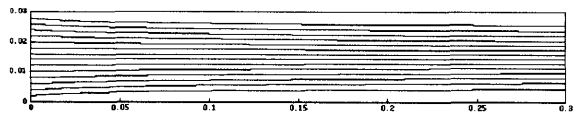
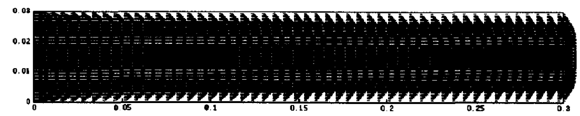
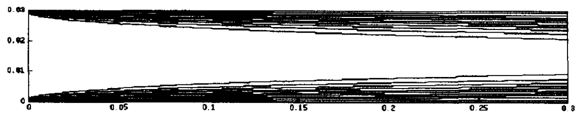


Streamlines

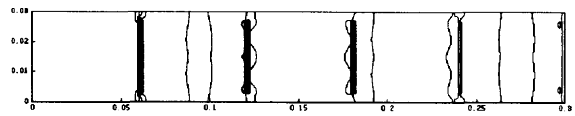


Velocity vector .



Isotherms

Case 2: Inline inserted plated without slit



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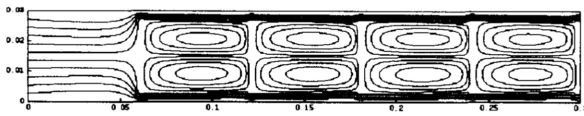
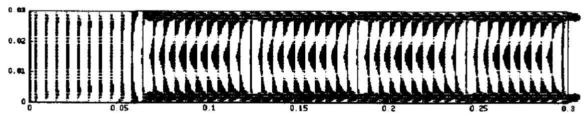
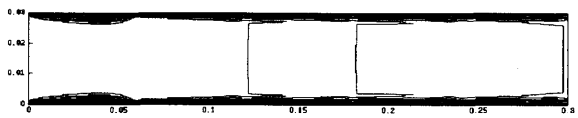


Streamlines

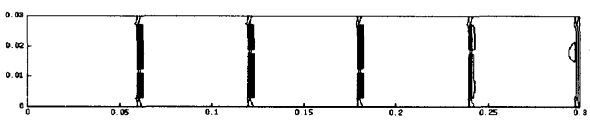


Velocity Vector



Isotherms

Case 3: Inline inserted plates with 1 slit



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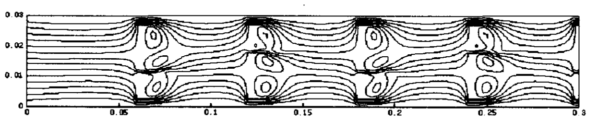
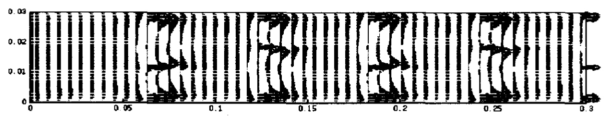
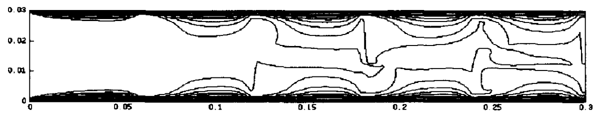


Streamlines



Velocity vector



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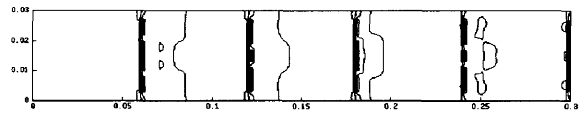
Case 4: Inline inserted plates with 2 slits



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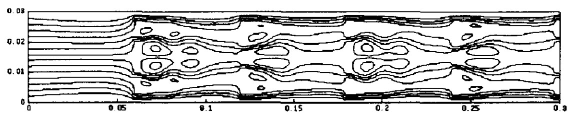
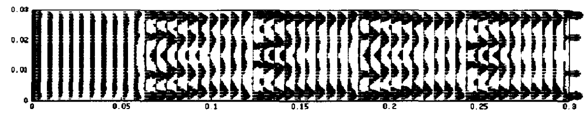
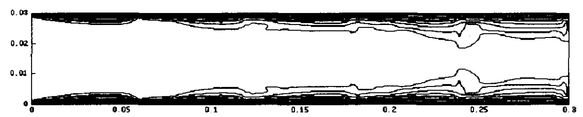


Streamlines

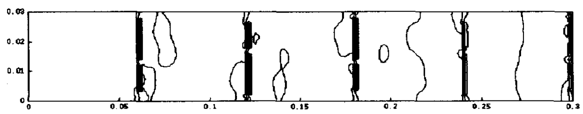


Velocity vector

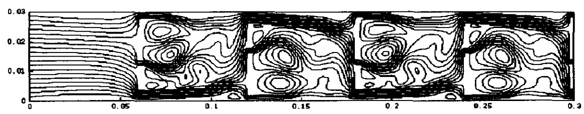


Isotherms

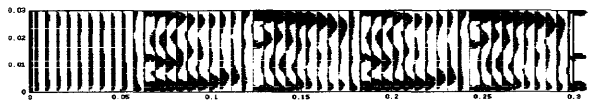
Case 5: Staggered inserted plates with 1 slit



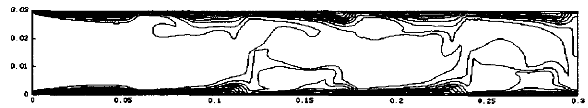
Pressure



Streamlines



Velocity vector



Isotherms

Figure 3 Flow Pattern and Temperature distribution for all cases for $U_\infty = 0.5 m/s$

The flow pattern of case 1 shows that there is no accelerated flow in the computational domain. This fact causes small pressure drop, however temperature gradient close to the wall is small due to absence of accelerated flow. The heat transfer coefficient in case 1 is the lowest due to small temperature gradient near the wall. Contrast with the case 1, the effect of inserted plates can be seen in case 2. Inserted plates cause accelerated flow adjacent the wall. These high speedy fluids result the low pressure area. In order to satisfy the continuity equations the reversal circulation flow will be appear. As the consequences, accelerated flow result higher heat transfer coefficient and reversal circulation flow results higher pressure drop. To decrease the pressure drop in the duct, slit holes were performed on the inserted plates. Effect of the slit holes to the flow pattern and temperature field are presented in Figure 2 also. The presence of the inserted plated, case 3, case 4, and case 5, keep resulting higher heat transfer coefficient due to accelerated flow adjacent the wall and the presence of the slits decrease the pressure drop due to smaller reversal circulation flow.

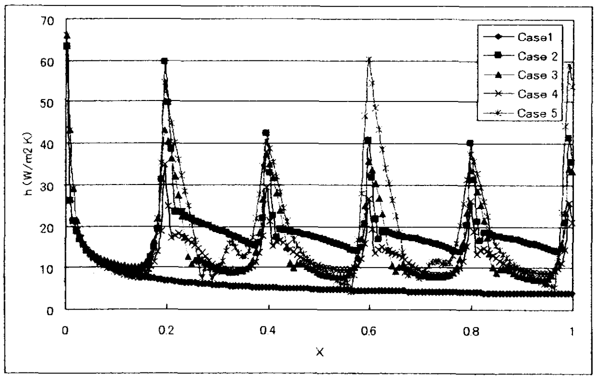


Figure 4 Local heat transfer coefficient for $U_\infty = 0.5 m/s$

The local heat transfer coefficients for all cases are presented in Figure 4. For case 1 the local heat transfer coefficient is higher around inlet and smaller toward the upstream and finally reached almost constant. For cases with inserted plates the local heat transfer coefficient jump up around the inserted plates. Since this numerically study using five inserted plates inside computational domain, the jump up local heat transfer coefficient happened five times. Special case can be seen for case 5, local heat transfer coefficient around inserted plate number 3 and number 4 are higher than all cases, even from case 2.

The average heat transfer coefficients for all cases are presented in Figure 5. For velocity under 0.7 m/s the highest average heat transfer coefficient is Case 2 but, for velocity over

0.7 m/s the highest is case 5. Since staggered position of the inserted plates in case 5 mix the flow inside the computational domain, for velocities over 0.7 m/s, average heat transfer coefficient of cases 5 is highest. Mixed flow in case 5 keep giving high temperature gradient on adjacent the wall inserted plate area.

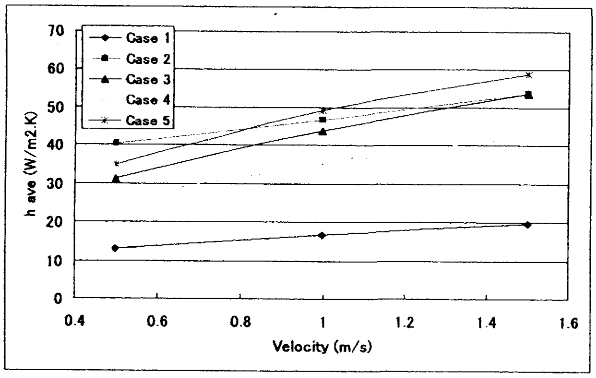


Figure 5 Average heat transfer coefficient

In order to make analysis of the best combination of inserted plate and slit, the total heat absorbed and fan power input for all cases are presented in Figure 6 and Figure 7.

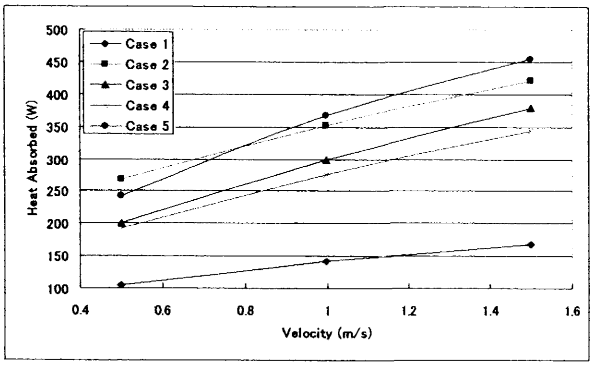


Figure 6 Total Transfer Heat for all cases

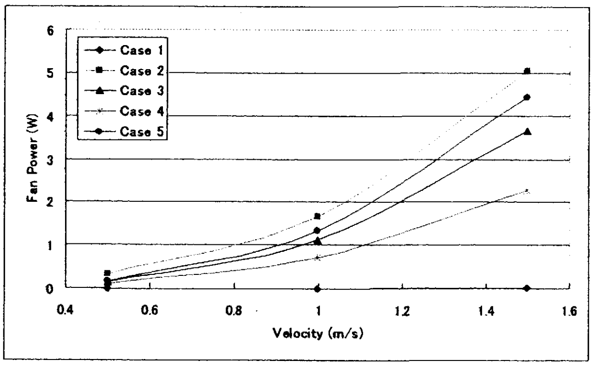


Figure 7 Fan Power input for all cases

The analysis can be made for the same fan power input 1 W condition. Total heat transfer for case 2, case 3, case 4, and case 5 are 320W, 295W, 290W, and 350W respectively. These numbers show that for case 5 give the best enhancement of transfer heat on the same power input. The next analysis can be made for the same amount of total heat transfer at 350W. The fan power input for case 2, case 3, case 4, and case 5 are 1.8W, 3W, 2.4W, and 1W. These numbers show that cases 5 give the best fan power decrease on the same total heat transfer amount.

Figure 8 shows the Number of Transfer Unit (NTU), $S = K.A/m.c_p$, which corresponds to the Heat Exchanger performance. It is seems that increasing the mass flow rate means the decrease of S which corresponds to low temperature efficiency.

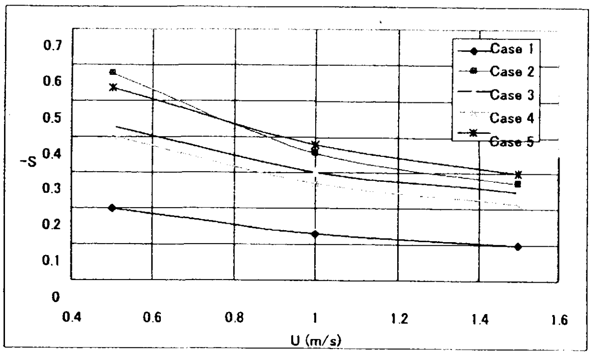


Figure 8 NTU (Number of Transfer Unit)

To specify the S value at constant mass flow rate (U=1 m/s) the values are S=0.36 for case 2, S=0.3 for case 3, S=0.2 for case 4, and S=0.38 for case 5 respectively. Especially the efficiency and the total heat transfer amount of Case 5, for staggered inserted plates with one slit hole, are most superior compare to the others while required fan power is lower than Case 2.

Conclusion

The numerical analysis of duct flow in heat exchanger with and without inserted plate has been carried out. The presence of the inserted plates enhance the heat transfer coefficient significantly, however this inserted plate increase the pressure loss. In order to decrease the pressure loss, the slit holes on the inserted plates were proposed. The presence of the slits on the inserted plates decrease the pressure drop significantly but on the other side the heat transfer coefficient is decreasing also. The staggered inserted plates with one slit holes reveals the best combination of heat transfer enhancement and decrease pressure drop.

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