Effect of Boundary Layer Generated on the fin surfaces of a Compact Heat Exchanger on the Heat Transfer and Pressure Drop Characteristics

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컴팩트형 열교환기의 핀 표면에서 발생하는 경계층이 열교환기의 전열 및 압력강하 특성의 변화에 미치는 영향에 관한 수치해석적 연구

김철호, 정지용

As a part of a project related to the development of the design algorithm of a compact heat exchanger for the application of the electronic home appliances, the effect of the discreteness of the airflow boundary generated on the cooling fin surface on the heat transfer and pressure drop characteristics of the heat exchanger was studied numerically. In general, there are two critical design parameters seriously considered in the design of the heat exchanger; heat transfer rate($\dot{Q}$) and pressure drop coefficient(C_p). Even though the higher heat transfer rate with lower pressure drop characteristics is required in a design of the heat exchanger, it is not an easy job to satisfy both conditions at the same time because these two parameters are phenomenally inversely proportional.

To control the boundary layer thickness and its length along the streamline, the surface of the flat fin was modified to accelerate the heat transfer rate on the fin surface. To understand the effect of the discreted fin size(S_w) and its location(S_h) on the performance of the heat exchanger in the airflow field, the flat fin was modified as shown in Fig. 1.

From this study, it was found that the smaller and more number of slits on the fin surface showed the higher energy diffusion rate. It means that the discreteness of the boundary layer is quite important on the heat transfer rate of the heat exchanger. On the other hand, if the fin surface configuration is very complex than needed, higher static pressure drop occurs than required in a system and it may be a reason of the induced aerodynamic noise in the heat exchanger.

Key Words : turbulent generator, slitted fins, heat transfer rate, energy diffusion rate, conjugate heat transfer, J-factor, induced aerodynamic noise

1. Introduction

A compact heat exchanger has been used for home appliances such as airconditioning units, refrigerators, chillers and dehumidifiers for a long time and mechanical engineers have studied to improve the efficiency of the heat exchanger to have higher performance products for a convenient daily life. The previous research works in this area have mainly relied upon the experimental approaches [5,6,8] until now. However, an experimental approach requires repeated time consuming work and the huge budget to build up experimental rigs and models for the better results and in some cases, it is impossible to reappear the nature phenomena in the laboratory by the experimental techniques.

In this work, a mathematical modelling was carried out to understand the effect of the fin surface configuration on the heat transfer rate and pressure drop characteristics of the conventional fin and tube type heat exchanger through the parametric study. Navier-Stokes equations were solved with the conjugate heat

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transfer problem to find out the temperature gradient in the airflow field. A body-fitted grid generation method was incorporated for the transformation of the physical model to the computational domain.

$$\dot{Q} = \dot{m}_{air}\ C_p\ \Delta T \qquad (kcal/s) \qquad (1)$$

To have maximum heat transfer rate of an heat exchanger, the physical phenomena of the generation of the thermal boundary layer on the fin surface should be clearly understood because the fin surface is the primary source of the thermal energy transfer contacting with air. As shown in eq. (1), the heat transfer rate($\dot{Q}$) is linearly proportional to the air mass flowrate(m_{air}) and temperature difference(ΔT) between air and the heat source(or sink). At a given inlet air velocity, the temperature difference is the only factor to have an effect on the heat transfer rate. To increase the temperature difference(ΔT) for the higher thermal energy transfer of a heat exchanger at a given boundary condition, the thermal boundary layer continuously generated on a fin surface should be discontinued with a change of the fin surface configuration. It will accelerate the heat transfer between a fin surface and air. The turbulence generated in the airflow field will also increase the heat transfer rate due to the forced convection effect on the fin surface.

Facts affecting heat transfer rate:

- Air Mass Flow Rate($\dot{m}_{air}$)
- Temperature Difference between Air and Heat Source(ΔT)
- Thermal Boundary Layer Generated on the Fin Surface
- Turbulence Generation for Forced Convection

Due to the discreteness of the thermal boundary layer and the turbulence generated by the slitted fin located in the airflow path, the airflow is complicated and the turbulent kinetic energy is increased in the flow field. This kinetic energy may be a source of the aerodynamic induced noise in the heat exchanger. Static pressure drop in the heat exchanger is mainly due to the form and skin friction drag of the slitted pieces of the fin.

Causes of induced aerodynamic noise

- Complexity of the Airflow Path
- Turbulent Intensity

In this study, the heat transfer performance and static pressure drop characteristics of twelve different types of model fins were numerically analyzed and compared the results with the results of the plane flat fin. Details of the configurations of the model fins and their study parameters are given at Table 1.

2. Heat Transfer Characteristics of a Fin

As mentioned earlier, the heat transfer performance of a fin is greatly influenced by the boundary layer thickness. As shown in Fig. 1, the boundary layer which has bad effects on the heat transfer to air is continuously generated on the fin surface. When the fin surface is discreted as shown in the punched and slitted fin, the previously generated boundary layer is stopped at the end of the first fin surface and air velocity increases again until it reaches to the leading edge of the next fin. This velocity increment accelerates the heat energy convection from the fin surface again. Therefore it should be noticed that the modification of the fin surface configuration is very important to accelerate the energy transfer between the fin surface and air. However, it should be noticed that the slitted pieces of the fin act like obstacles to the airflow path and the reason for the static pressure increase on the heat exchanger system.

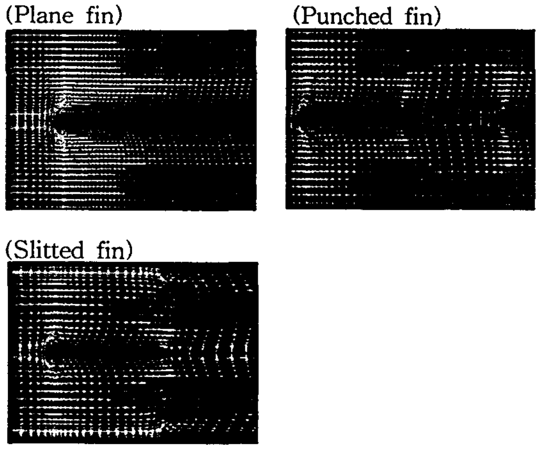

Fig. 1 Airflow velocity vectors around fins ; (U_{in} = 2.0 m/s, T_{fin} = 10°C, T_{air} = 30°C).

Fig. 2 shows the heat energy diffusion rate of each different fin configuration. Compared with the two former fin patterns, the slitted fin shows the higher energy diffusion rate. It is mainly due to the accelerated air velocity introduced into the leading edge of the following slitted fin. Eventually, if the optimized fin configuration at a given inlet boundary condition is found, the heat exchanger with the highest heat transfer performance can be obtained.

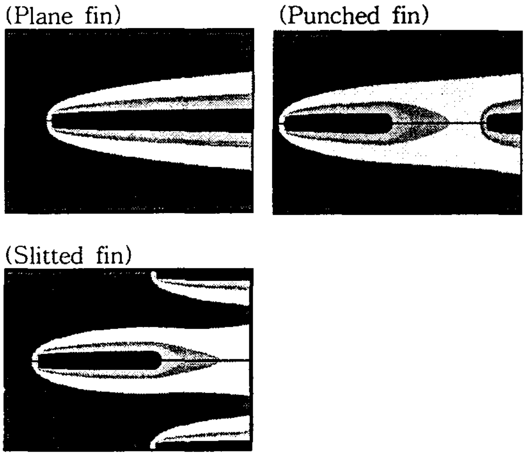

Fig. 2 Temperature distributions around fins by the thermal energy diffusion in an airflow field; (U_{in} = 2.0 m/s, T_{fin} = 10°C, T_{air} = 30°C).

3. Mathematical Models and Numerical Procedure

3.1 Geometry and Numerical Grids of Physical Model

The performance of a compact heat exchanger is decided by two parameters[3]; the heat transfer rate and the static pressure drop. The heat transfer rate is affected by the two factors; size of a heat transfer area and the boundary layer thickness. The pressure drop is also affected by the fin surface shape or configuration and the tube arrangement.

In this study, the twelve different arrangements of the fin slits were incorporated to see the effect of the slit size and its location on the performance of the compact heat exchangers. Fig. 3 shows 3-dimensional prospective view of the Model1 fin. The inlet air was assumed to be dry one and the inlet velocity(U_{in}), temperature(T_{air}) and the fin temperature(T_{fin}) were fixed to be constants but the fin surface configuration(Slit height(S_h), Slit length(S_L)) was changed to see its effect on the boundary layer generated which has a serious effect on the performance of the heat exchanger conclusively.

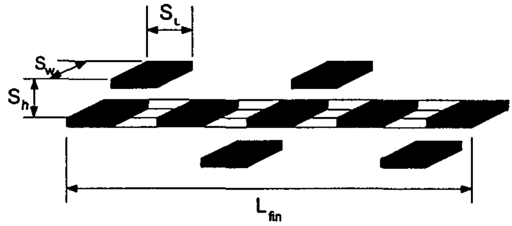

Fig. 3 Prospective view of the Model1 fin

The details of the parameters and their ranges for this numerical experimental study are given below at Table 1.

Table 1 Details of model fin configurations and ranges of parameters.

Model Name	Inlet Air Vel.	No. of Slit(ea)	Slit Height	Slit Length	Fin Solidity
Model1 Model2 Model3	0.5m/s 1.0m/s 1.5m/s 2.0m/s 3.0m/s	4, 6, 8	1/3P_{fin} 1/2P_{fin} 2/3P_{fin} 3/3P_{fin}	1.5mm 1.0mm 0.75mm	$\frac{L_{slit}}{L_{fin}}$ = 0.494
where : L_{fin}=12.15mm, P_{fin}=1.39mm, T_{fin}=0.1mm					

(where L_{fin} = fin length, P_{fin} = fin pitch and T_{fin} = fin thickness)

Fig. 4 shows the two-dimensional typical numerical grid of Model3.

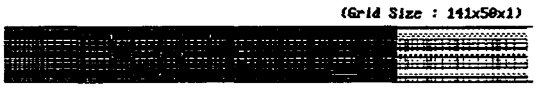

Fig. 4 Two-dimensional view of a typical numerical grid ; (Model3)

- Fixed Boundary Conditions of the Inlet Air and Fins ;

① U_{air} : 0.5 ~ 3.0 m/s
② T_{air} : 27.0°C
③ T_{fin} : 9.0°C

3-2 Computational Scheme

The Reynolds number range is around 350 to 1000 and the material for the fin was defined as aluminum which has high conductivity. For this study Navier-Stokes equations were solved for the airflow field analysis and the conjugate heat transfer equation was solved to find out the temperature gradient in an airflow field. The no-slip condition near solid boundaries is modelled by the logarithmic law. Time differencing is fully implicit backward whilst advection terms are hybrid differenced. Conjugate gradient techniques for pressure corrections in transport equations were incorporated and 'SIMPLE' algorithm[4] was used for the velocity/pressure coupling in this application. For this calculation, the computer program, PHOENICS(V2.2.1)[1] which is a general purpose, multi-dimensional, finite volume code, has been incorporated.

Inlet boundary conditions may be designated either in terms of pressure or velocity. Through the preliminary numerical experiments[2], it was found that there were no differences in the calculation results between the pressure and the velocity boundary condition. However, much less computer CPU time was required for the calculations using the latter. Therefore, the velocity boundary condition has been selected for the computations presented here.

3-3 Governing Equations

The basic equations of fluid mechanics to solve the base airflow in the control volume are the Navier-Stokes equations. They comprise of equations for conservation of mass and momentum.

(1) *Conservation of Mass* :

$$\frac{\partial \rho}{\partial t} + \nabla(\rho U) = 0 \quad (2)$$

(2) *Conservation of Momentum* :

For a laminar flow,

$$\nabla \cdot (\rho\, U \otimes U) - \nabla \cdot (\mu \nabla U) = -\nabla P + \nabla \cdot (\mu (\nabla U)^T) \quad (3)$$

(3) *Conservation of Energy* :

$$\nabla \cdot (\rho U T) - \nabla \cdot (\frac{\lambda}{C_p} \nabla T) = 0 \quad (4)$$

The temperature is calculated from the enthalpy using an equation of state :

$$H = C_p\,(\,T - T_{ref}\,) \quad (5)$$

where T_{ref} is a reference temperature where the enthalpy is zero.

4. Mathematical Analysis for the Performance of the Heat Exchanger

As previously mentioned, the heat transfer performance and pressure drop characteristics should be found to estimate the performance of the heat exchanger. The total heat flux(Q, J/s) and pressure drop coefficient(F)[7] can be calculated from the numerical simulation result by the use of equations given below.

(1) Total Heat Flux (Q) :

$$\dot{Q} = \dot{m}\, C_p\, (T_{out} - T_{in}) \quad (6)$$

(2) Pressure Drop Coefficient (F) :

$$F = \frac{\Delta P}{1/2 \rho U^2} \quad (7)$$

where ΔP : pressure difference between inlet and outlet of the heat exchanger
U : averaged intake air velocity of free stream

5. Results and Discussion

The slitted pieces of the model fins act like the turbulent generator in the flow field and also like the boundary layer braker to accelerate heat exchange between fin and air. Two geometric parameters of the model fins; slit length(S_L), hight(S_h) and air inlet velocity(U_{in}) were chosen as the experimental parameters in this numerical study. The heat transfer and pressure drop characteristics of each model fin were calculated and compared with those of the plane flat fin to see the effect of the change of the fin surface configuration on the heat exchanger performance. The calculation results were analysed by two ways; the heat transfer performance and the static pressure drop characteristics.

Fig. 5 and Fig. 6 show the thermal energy diffusion rate of each model fin. In the case of the plane fin(without slits), the energy diffusion is very low compare with other models. It is due to the thermal boundary layer continuously generated along the fin surface.

Fig. 5 shows the effect of the slit size and its number on the heat transfer rate in the flow field. The inlet air velocity(U_{in}) and slit location(S_h) were fixed to 1.5m/s and ½P_{fin} respectively. The smaller size of the fin slit discreted, the higher diffusion rate of thermal energy occurs in the flow field. It means that as the boundary layer generated on the fin surface is discreted into a smaller size, it should be a reason of the acceleration of the heat transfer rate between air and the heat exchanger in the airflow field.

Fig. 6 shows the effect of the slit location(S_h) on the energy diffusion rate in the flow field. The inlet air velocity(U_{in}) and number of the slit(S_n) were fixed to 1.5m/s and 6, respectively. In all cases, the energy transfer rate has been increased up along with the increase of the slit location to S_h = ⅔P_{fin}. In the last case(S_h= 3/3P_{fin}), the slited pieces of the fin are arranged in-line in the middle of the flow path between two fins and the boundary layer discreted by the previous fin slit was recovered soon at the inlet of the next one. It is the main reason why the energy diffusion rate is decreased in the flow field.

From this numerical experiment, it was found that the proper discreteness of the fin surface of the heat exchanger at the given operating condition is very important fact to be seriously considered to meet the required heat transfer rate in a newly designed heat exchanger.

Fig. 7 and Fig. 8 show the variation of total heat flux(Q, J/s) and *F-Factor* with the change of inlet air velocity of each model fin. As shown in Fig. 7, when the fin slit location(S_h) of all model fins is lower than ⅓ P_{fin} the thermal energy diffusion is less than that of the plane flat fin. It means that even though the fin slits are placed in the flow field and play the role as the discretor of the boundary layer, the flow path between the fin and slits is too narrow to have enough air passing through for the heat exchange. It should be noticed that when the fin slits are made on a fin surface, the slit location(S_h) must be higher than the boundary layer thickness generated on both fin and slit surface. In all other cases, the heat transfer rate is higher than that of the plane flat fin in all velocity ranges.

Fig. 8 shows the variation of *F-Factor* which represents the characteristics of the static pressure drop of the model fins with respect to inlet air velocity. The plane flat fin shows the lowest pressure drop in all velocity ranges. The model fins which have ⅓P_{fin} of the slit location has the second lowest pressure drop characteristics in all velocity ranges. It is also due to the boundary layer generated on the fin surface. From this numerical study, the static pressure drop characteristics can be exactly estimated in all model fins. With this numerical method, the limitation of the static pressure drop of a new designed heat exchanger can be exactly satisfied for the practical application.

Fig. 9 and Fig. 10 show the increment percentage of the averaged total heat flux(Q, J/s) and static pressure of each model fin with respect to those of the plane flat fin. The both values of all models are continuously increased until the slit location(S_h) to ⅔P_{fin} but decreased when the location is reached to 3/3P_{fin} in the flow field. The smaller number of slits placed in the flow field, the lower heat transfer rate occurs between fin and air.

6. Conclusion

The performance of a compact heat exchanger is decided by two facts; the heat transfer rate and the pressure drop in an airflow field. These facts are greatly influenced by the size of the heat transfer area of a fin and the complexity of the fin surface configuration. In this study, slit size(S_L) and its location(S_h) in the airflow field were chosen as the main parameters to see their effects on the heat transfer and pressure drop characteristics of the compact exchanger. In a certain extent, these fin slits placed between two fins of the heat exchanger act like the scatteror or discretor of the boundary layer generated on the fin surface but the static pressure drop must be increased because of these.

The effect of the slit size(S_L) and its location(S_h) on the performance of the heat exchanger was numerically tested and the results were compared those of the plane flat fin. The following results were found;

(1) Heat Transfer Rate Performance :

A discreteness of the thermal boundary layer on a fin surface is necessary to accelerate the thermal energy diffusion rate in an airflow field. Normally it is achieved by making slatted patterns on a fin surface of the heat exchanger. However, the slitted pattern(slit size and location) on a fin surface must be carefully decided considering the inlet air condition. In the case of slit size(S_L), the smaller and more number of slits showed higher energy diffusion rate. It means that the discreteness of the boundary layer is quite important on the heat transfer rate of the heat exchanger. However, in the case of the slit location, the slits should be scattered jig-jag in the flow field. As the slits were located in line in the middle of the flow field, the heat transfer rate was decreased indeed.

(2) Pressure Drop Performance :

The pressure drop characteristics in a compact heat exchanger is greatly influenced by the fin surface configuration at a given operating condition. As shown above the more complex in a fin surface configuration the higher static pressure drop induced. If the fin surface configuration is very complex than needed, higher static pressure drop occurs than required and the airflow energy may be converted into the induced aerodynamic noise in the heat exchanger. As expected, the more number of slits in the field, the higher static pressure drop obtained. As the slits are in-lined in the middle of the flow field, the pressure drop is decreased.

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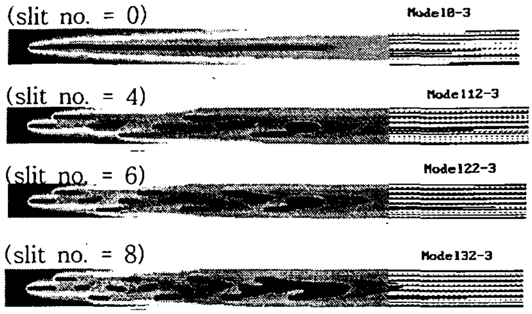

Fig. 5 Effect of no. of slit on the diffusion of the thermal energy in airflow field; (U_{in}=1.5m/s, S_h=½P_{fin})

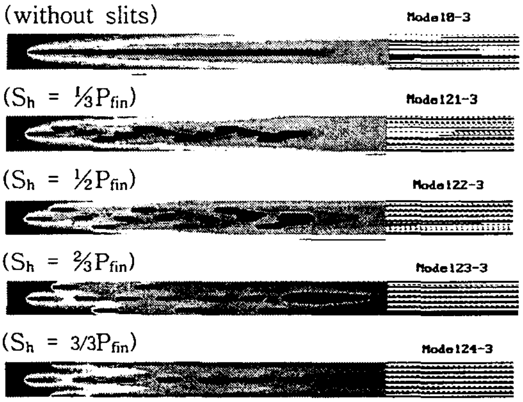

Fig. 6 Effect of slit location(S_h) on the diffusion of the thermal energy in airflow field; (U_{in}=1.5m/s, No. of Slit = 6)

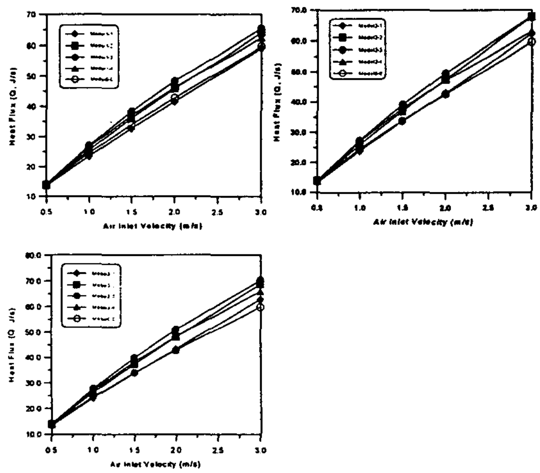

Fig. 7 Effect of slit no. and slit location on the total heat flux(Q) comparing to the plane flat fins : S_h = ⅓P_{fin}, ½P_{fin}, ⅔P_{fin}, 3/3P_{fin}

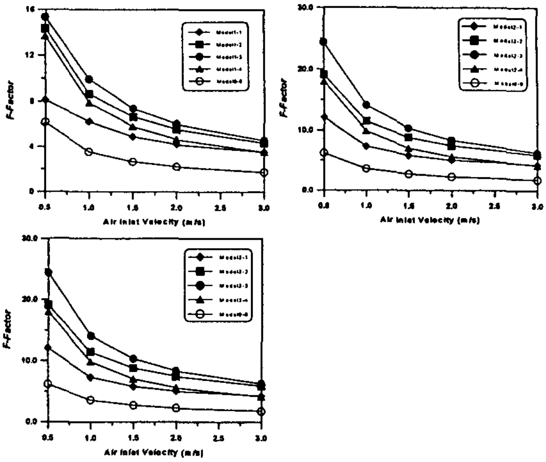

Fig. 8 Effect of no. of slit and slit location (S_h) on pressure drop characteristics(F) comparing to the plane flat fins : S_h = ⅓P_{fin}, ½P_{fin}, ⅔P_{fin}, 3/3P_{fin}

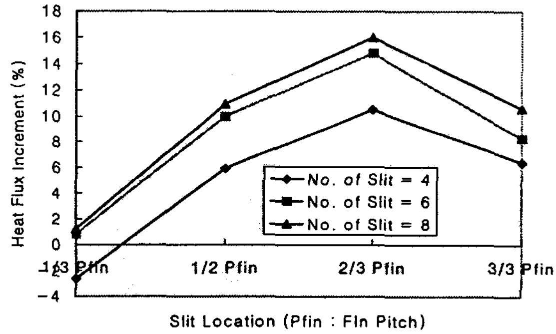

Fig. 9 Averaged total heat flux(Q) increment (%) of the model fins with respect to plane flat fin ; $\frac{Q_{model-x} - Q_{model-0}}{Q_{model-0}} \times 100$ (%)

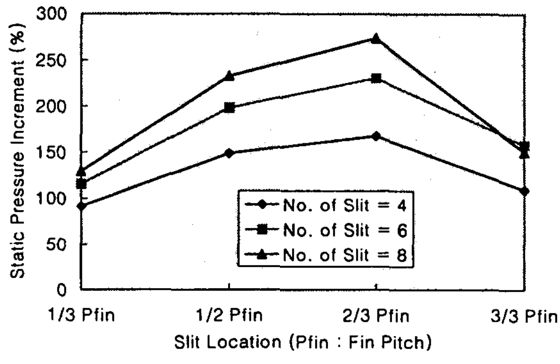

Fig. 10 Averaged static pressure increment (%) of the model fins with respect to plane flat fin ; $\frac{P_{model-x} - P_{model-0}}{P_{model-0}} \times 100$ (%)