150 International Journal of Air-Conditioning and Refrigeration Vol. 11, No. 3 (2003)/pp. 150-157

Local and Overall Heat Transfer Characteristics of Fin- Flat Tube Heat Exchanger with Vortex Generators

Seong-Yeon Yoo†, **Min-Ho Chung***, **Dong-Seong Park***

Department of Mechanical Design Engineering, Chungnam National University, Daejeon 305-764, Korea
**Graduate School, Chungnam National University, Daejeon 305-764, Korea*

Key words: Fin-tube heat exchanger, Flat tube, Naphthalene sublimation technique, Vortex generator, Heat transfer enhancement

ABSTRACT: Local and overall heat transfer characteristics of fin-flat tube heat exchangers with and without vortex generators were investigated. Local heat transfer coefficients were measured with the heat exchanger model using naphthalene sublimation technique. In case of a fin-flat tube heat exchanger without vortex generators, only the horseshoe vortices formed around tubes augment the heat transfer. On the other hand, longitudinal vortices created artificially by vortex generators additionally enhance heat transfer in case of a fin-flat tube heat exchanger with vortex generators. Overall heat transfer coefficients were measured with the prototypes of the fin-flat tube heat exchanger with and without vortex generators in a wind tunnel and results were compared with those of a fin-circular tube heat exchanger with wavy fin. Friction losses for heat exchangers were also measured and compared. The fin-flat tube heat exchanger with vortex generators is found to be more effective than the fin-circular tube heat exchanger with wavy fin.

Nomenclature

A	: area of heat exchanger [m^2]
F	: correction factor
f	: apparent friction factor
H	: fin spacing [m]
L	: distance between pressure measuring points [m]
P	: fan power [W]
Q	: air flow rate [m^3/s]
$\dot{Q}_a$	: heat transfer rate [W]
U	: overall heat transfer coefficient [W/m^2·K]
V	: air velocity [m/s]

Greek symbols

μ	: viscosity of air [kg/s·m]
ρ	: density of air [kg/m^3]
ΔP	: pressure drop [Pa]
ΔT_m	: log mean temperature difference [K]

1. Introduction

For energy conservation and environmental protection, it becomes increasingly important to adopt heat transfer enhancement technique in the design of heat exchangers used in process industries for heating, cooling, air-conditioning and refrigeration. There are two enhancement technologies of convective heat transfer for compact heat exchangers. One is to extend heat transfer surface area like a fin, the other is to increase heat transfer coefficients between solid

---

† Corresponding author
Tel.: +82-42-821-6646; fax: +82-42-822-7366
E-mail address: syyooh@cnu.ac.kr

surface and fluid. Fin and extended heat transfer region such as slit fin, spiral tube, rubber fin and so on cause the disturbance of fluid flow that causes more power consumption for pump and blower.(1)

In the present work, vortex generators were fabricated on the fin surface of a fin-tube heat exchanger to augment the convective heat transfer. In addition to horseshoe vortices formed naturally around the tube of the fin-tube heat exchanger, longitudinal vortices are artificially created on the fin surface by vortex generators.(2-3) Also, generated vortex has a strong shear stress which could reduce fouling on the fin surface. The purpose of this study is to investigate the local and overall heat transfer phenomena in the fin-flat tube heat exchangers with and without vortex generators, and to evaluate the effect of vortices on the heat transfer enhancement.

Fiebig and Chen(4) have summarized their systematic study on the characteristics of heat transfer surface with vortex generators. According to their research, the fin heat exchanger surface area may be reduced with vortex generators by more than 50% compared to a plain fin for identical heat duty and pressure loss. Zhu et al.(5) have studied numerically the effect of vortex generator on the flow and heat transfer in a rib-roughened channel. Yoo(6) have developed fin-flat tube heat exchanger with vortex generators, and showed that vortex generators increase heat transfer rates on the fin surface by almost double. Kang et al.(7) have used liquid crystal method to investigate the thermal characteristics on the fin of finned tube heat exchanger.

When the heat is transferred by forced convection from the fin-tube heat exchanger with the vortex generators, complex flow phenomena, such as stagnation, separation, vortex formation or wake affect the heat transfer characteristics. In a complicated flow situation, it is very difficult to measure local heat transfer coefficients by conventional methods of heat transfer measurement. Goldstein et al.(8) showed that mass transfer experiment using naphthalene sublimation technique is an effective way for measuring local heat transfer distribution in a complex, three-dimensional flow region. In the present study, the naphthalene sublimation technique was employed to measure local mass transfer from fin-flat tube heat exchangers. Then, mass transfer data were converted to their counterpart of heat transfer processes using a heat/mass transfer analogy.

Experiments were performed to investigate local heat transfer characteristics for the model of fin-flat tube heat exchangers with and without vortex generators. To investigate overall heat transfer characteristic, the prototypes of heat exchanger with and without vortex generators were manufactured, performance tests were done at various conditions, and test results were compared each other.

2. Experimental apparatus and procedure

2.1 Experimental apparatus

The experimental apparatus for model test comprised a wind tunnel, a naphthalene casting facility and a sublimation depth measurement system. The open-circuit blowing type wind tunnel, which has a square test section of 300 × 46 mm, was used. The air speed was controlled by an inverter and the freestream turbulence intensity was less than 1.0% over the entire range of speed.

The naphthalene casting facility has a mold, heating plate, hot air gun, mold separating device and suction hood. Automatic depth measurement system was used to measure local mass transfer coefficients. The depth measurement system consisted of a depth gage, a signal conditioner, two stepper motor-driven traversing table, a data acquisition board and a personal computer. Fig. 1 shows the schematic

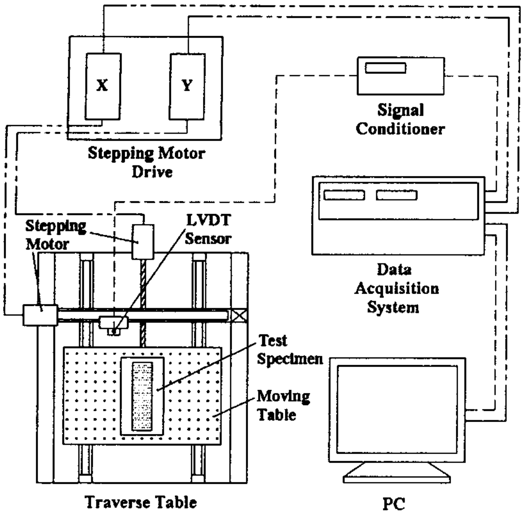


Fig. 1 Schematic of sublimation depth measurement system.

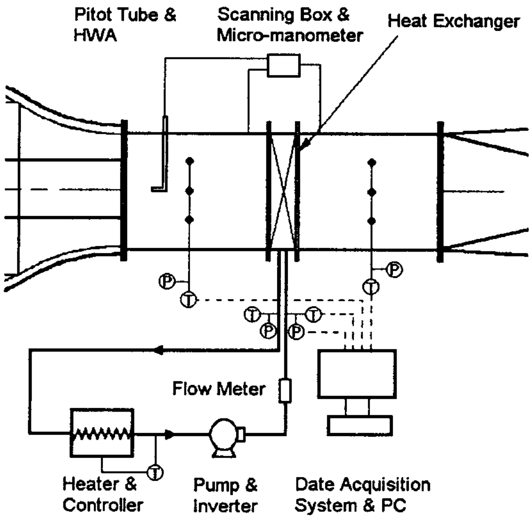


Fig. 2 Schematic of prototype test apparatus on heat exchanger.

of sublimation depth measurement system. The depth gage used to measure the naphthalene surface profile was a linear variable differential transformer (LVDT) which has a ±0.254 mm (0.01 in) linear range and a resolution of ±25.4 nm (1 μ-in). It is connected to a signal conditioner, which supplies excitation voltage to LVDT and amplifies the output signal from LVDT.

The experimental apparatus for prototype test, shown in Fig. 2, comprised a wind tunnel, hot water supply system, heat exchanger, and measurement and control system. The open-circuit suction type wind tunnel, which has a square test section of 400×300 mm is used. Air speed and temperature were controlled by inverter and electric heater, respectively. Water supply system consisted of constant temperature bath, pump and flow meter. The temperature of bath was controlled with RTD sensor, PID controller, and circulation pump was operated by an inverter to control flowrate.

2.2 Experimental procedure

In order to measure the local heat transfer coefficients, a new naphthalene casting was made for each test run. The testpiece was clamped to a polished mold, and molten naphthalene was well poured into the mold. After the naphthalene solidifies, the mold was separated from the testpiece by applying a shear force. The casted testpiece was placed and then clamped on the sublimation depth measurement table. Initial readings of the naphthalene surface elevation were taken at predetermined locations using the automated sublimation depth measurement system. The testpiece was then installed in the wind tunnel and exposed to the air stream for about one hour. During a test run, the naphthalene surface temperature, tunnel inlet air temperature and pressure, and free-stream velocity were measured. The testpiece was then removed and a second set of surface elevation was obtained at the same locations as before. Finally, data reduction program calculates Sherwood and Nusselt numbers from the measured sublimation depth using heat/mass transfer analogy.

Overall heat transfer measurements for the prototypes of fin-flat tube heat exchanger with and without vortex generators and those of fin-circular tube heat exchanger with wavy fin are performed in the wind tunnel. When the circulated water temperature reaches steady state, inlet and outlet temperatures of air and water are measured using thermocouples, and ambient temperature is also measured for heat loss calculation. Pitot tube is used to measure air velocity, and electro- magnetic flow meter is used to measure water flowrate.

3. Results and discussion

3.1 Local heat transfer

Fin-flat tube heat exchanger is often adopted in air-conditioning and gas-boiler system instead of fin-circular tube heat exchangers, to reduce the pressure loss. But it is well known that heat transfer performance of the fin-flat tube heat exchanger is much lower than that of fin-circular tube heat exchanger. Yoo(6) have reported that dual effects, the enhancement of heat transfer and the reduction of pressure loss, could be accomplished by implementing vortex generators on the fin surface. Fig. 3 shows the schematic of the fin-flat tube heat exchanger with vortex generator. Flat tubes having 4 staggered rows were arranged with 33 mm distance in the streamwise (X) direction and 38 mm distance in the spanwise (Y) direction. Dimension of flat tube was 31 mm in length and 7 mm in width. Fins were made of 0.15 mm thick copper sheet, vortex generators, 3.2×3.2 mm, are punched vertically out of the fins, and fin spacing was the same as the height of vortex generator. The bottom plate of the wind tunnel was casted with naphthalene and heat exchanger model was mounted on it.

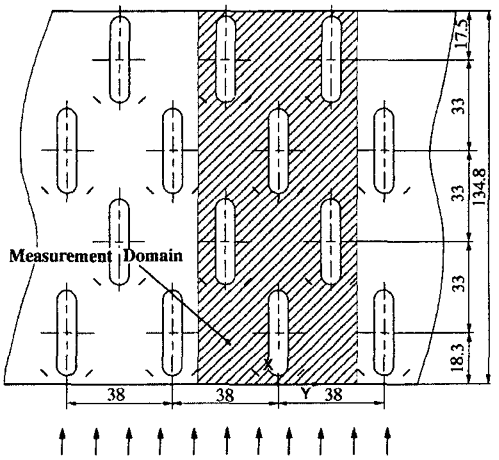


Fig. 3 Schematic of fin-flat tube heat exchanger with vortex generator.

Fig. 4 shows the distributions of local heat transfer coefficients in the fin-flat tube heat exchanger without vortex generator. When the boundary layer flow meet the protruding object, tube in this study, flow is slowdown and dynamic pressure is converted to static pressure. A pressure gradient across the boundary layer is, therefore, created. This pressure gradient causes the flow to move toward the fin and reverse flow at the region nearest to the fin. This reverse flow of the boundary layer produces the horseshoe vortex, which is then carried around the tube by the main flow. This

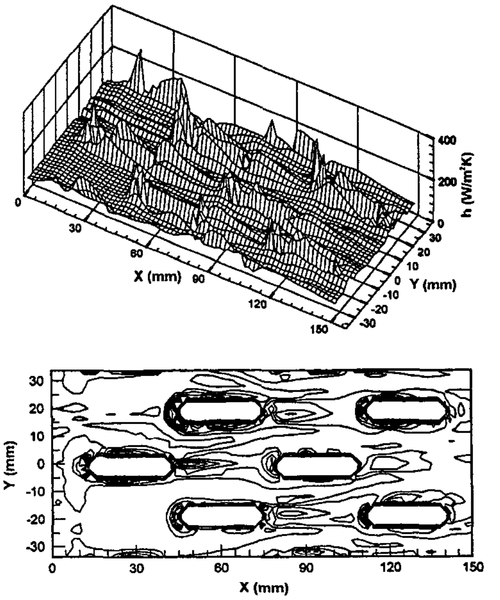


Fig. 4 Distribution of local heat transfer coefficient for fin-flat tube heat exchanger without vortex generator.

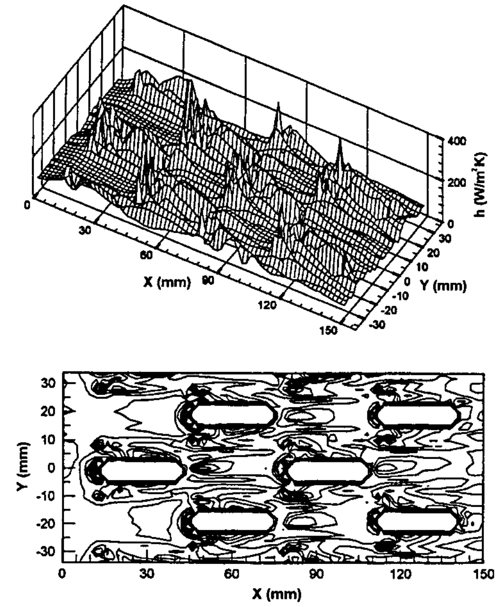


Fig. 5 Distribution of local heat transfer coefficient for fin-flat tube heat exchanger with vortex generator.

horseshoe vortex enhances the convective heat transfer dramatically. Horseshoe-like peak, starting in front of the tube and continuing around the tube, corresponds to the trail of horseshoe vortex. Heat transfer enhancement by horseshoe vortices is found in front of flat tubes, but their magnitude is much less than that of fin-circular tube.(6) Blockage area of flat tube is smaller than that of circular tube, so intensity of horseshoe vortices formed around flat tubes is weaker. This is the reason why fin-flat tube heat exchanger has lower heat transfer performance than fin-circular tube. Distribution of local heat transfer coefficients in the fin-flat tube heat exchanger with vortex generators is shown in Fig. 5. In addition to heat transfer enhancement by horseshoe vortices, longitudinal vortices created by vortex generators are found to augment heat transfer drastically.

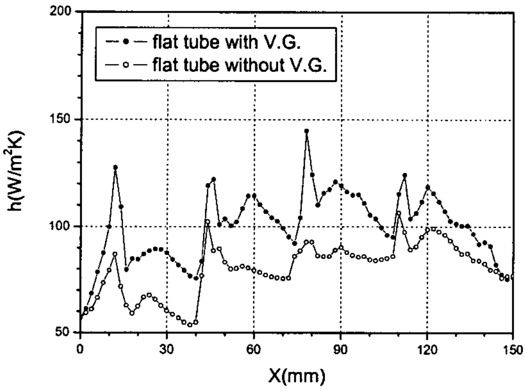


Fig. 6 Effect of vortex generator on heat transfer enhancement for fin-flat tube heat exchanger.

Effect of vortex generators on the heat transfer enhancement in the fin-flat tube heat exchanger is shown in Fig. 6. Because vortex generators were installed nearby upstream side of flat tube, heat transfer enhancement caused by horseshoe vortices and by longitudinal vortices are coupled. The increase in Nusselt number by vortex generators is found to be much higher in the fin-flat tube heat exchanger than fin-circular tube.(6) Total average heat transfer coefficient of fin-flat tube heat exchanger with vortex generators increases by 40% compared to that of the fin-flat tube without vortex generator.

3.2 Overall heat transfer

Fig. 7 shows the schematic of wavy fin of fin-circular tube heat exchanger which is used for AHU. Prototype heat exchanger has a dimension of 400 (W) × 304 (H) × 252 (D) mm, which was fabricated to fit wind tunnel test section.

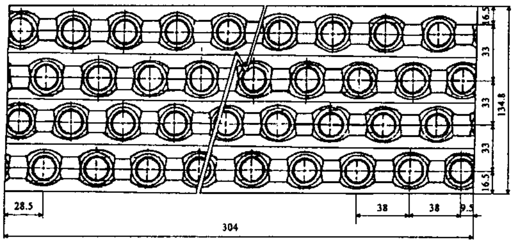


Fig. 7 Schematic of fin-circular tube heat exchanger with wavy fin.

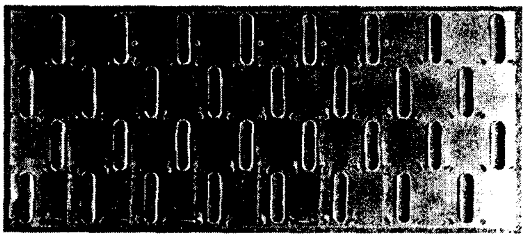


Fig. 8 Photo of fin-flat tube heat exchanger with vortex generators.

It consists of 32 (4 rows×8 columns) copper tubes in a staggered arrangement, and 124 copper fins of 0.15 mm thickness and 3.2 mm pitch. In a water flow line, header was attached to the first row of tubes so that 8 tubes in the equivalent row have same temperature condition, and air vent and drain are mounted on the inlet and outlet of header.

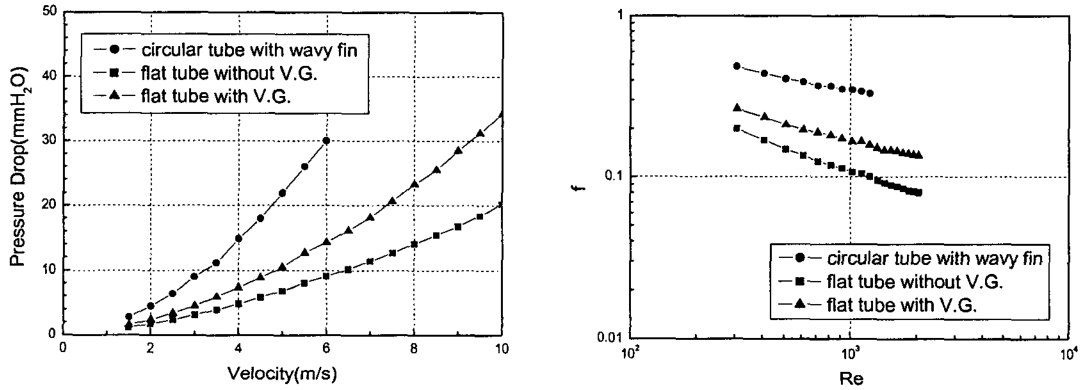
The prototypes of fin-flat tube heat exchanger were fabricated with and without vortex generators. Square wing type vortex generator (3.2×3.2 mm) is used, and two vortex generators for each tube are punched from the fin surface with an angle of 45° to the main flow direction, as shown in Fig. 8. The overall size, material, number of fins and tubes, cross-sectional area of tube, tube location of fin-flat tube heat exchangers are the same as in model and also those of wavy fin-circular tube heat exchanger.

Fig. 9 (a) shows the pressure drop of circular tube heat exchanger with wavy fin, flat tube heat exchanger without vortex generator, and flat tube heat exchanger with vortex generator. Pressure drop increases as the air velocity increases, and that of circular tube heat exchanger with wavy fin is much higher than that of flat tube heat exchanger because of relatively larger blockage area. To evaluate pressure drop, apparent friction factor in the air side, defined by Eq. (1), is plotted in Fig. 9 (b).

$$f = \frac{\Delta P}{\frac{1}{2}\rho V^2} \cdot \frac{H}{L} \qquad (1)$$

$$\mathrm{Re} = \frac{\rho V H}{\mu} \qquad (2)$$

where H is fin spacing, the distance between fins, L is the distance between pressure measuring points and V is the face velocity of heat exchanger. Friction factor decreases as the Reynolds number, based on fin spacing, increases and is linearly dependent on Reynolds number in the log-log scale. Friction factor of fin-circular tube heat exchanger is almost three times than that of fin-flat tube heat exchanger without vortex generators, and two times than that of fin-flat tube heat exchanger with vortex generators.



(a) Pressure drop (b) Apparent friction factor

Fig. 9 Variation of pressure drop and apparent friction factor of air side.

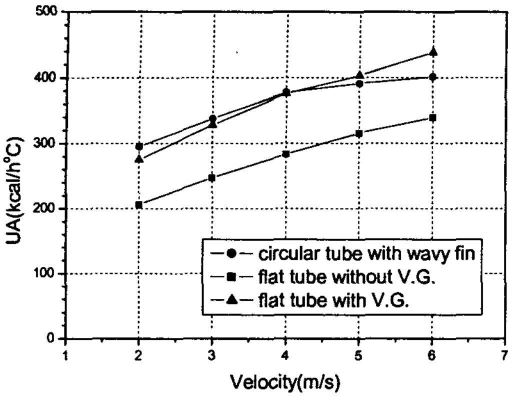


Fig. 10 Variation of UA against air velocity.

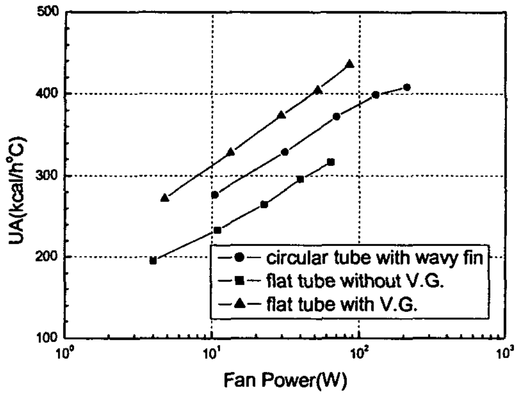


Fig. 11 Variation of UA against equivalent fan power.

Thermal performance in the fin-tube heat exchangers was expressed in terms of UA, which is defined by the following equation:

$$\dot{Q}_a = (UAF)_a \Delta T_m \qquad (3)$$

where $\dot{Q}_a$ is heat transfer rate from hot water to air, U is overall heat transfer coefficient between them, A is heat transfer area, F is correction factor, which is approximately 1 in the present study because of 4 row tubes. and ΔT_m is LMTD (Log Mean Temperature Difference), which is calculated from the inlet and outlet temperature of hot water and cooling air. Fig. 10 shows the UA against air velocity. The UA increases as the air velocity increases, and the UA of fin-flat tube heat exchanger without vortex generators is lower than that of fin-circular tube heat exchanger. But the UA of fin-flat tube heat exchanger with vortex generators is almost same as that of the fin-circular tube heat exchanger. Heat transfer rates of fin-circular tube heat exchanger are shown to be higher than those obtained in the model test, because wavy fin was used in the prototype test instead of plain fin, which was used in the model test.

Fig. 11 shows thermal performance against equivalent fan power, which is defined by the following equation:

$$P = \Delta P \cdot Q = \Delta P \cdot (VA) \qquad (4)$$

where ΔP is the pressure loss in air side, and Q is the air flowrate. Thermal performance of fin-flat tube heat exchanger with vortex generators increases by 12% compared to that of wavy fin-circular tube heat exchanger. Much higher heat transfer enhancement is expected, if comparison is made with plain fin-circular tube heat exchanger. And thermal performance of fin-flat tube heat exchanger with vortex generators increases by 29% compared to that of fin-flat tube heat exchanger without vortex generators.

4. Conclusions

Local and overall heat transfer characteristics of the model and the prototype of fin-flat tube heat exchangers with and without vortex generators are measured and compared. Summary of major results are as follows.

(1) Horseshoe vortices formed around the tube dramatically enhance the heat transfer, and longitudinal vortices created by vortex generators also augment the heat transfer.

(2) Average heat transfer coefficient on the fin surface of fin-flat tube heat exchanger with

vortex generators increases by 40% compared to that of the fin-flat tube without vortex generator.

(3) Fin-flat tube heat exchanger with vortex generators has much lower pressure loss than conventional fin-circular tube heat exchanger without vortex generator.

(4) Thermal performance of fin-flat tube heat exchanger with vortex generators increases by 12% compared to that of wavy fin-circular tube, and 29% compared to that of fin-flat tube without vortex generators.

Acknowledgements

This research was performed for the Carbon Dioxide Reduction & Sequestration Center, one of 21st Century Frontier R&D Programs funded by the Ministry of Science and Technology of Korea.

References

1. Kays, W. M. and London, A. L., 1964, Compact Heat Exchangers, McGraw-Hill.
2. Tiggelbeck, S., Mitra, N. K. and Fiebig, M., 1994, Comparison of wing-type vortex generators for heat transfer enhancement in channel flows, J. of Heat Transfer, Vol. 116, pp. 880-885.
3. Biswas, G., Mitra, N. K. and Fiebig, M., 1994, Heat transfer enhancement in fin-tube heat exchangers by winglet type vortex generators, Int. J. of Heat Mass Transfer, Vol. 37, No. 2, pp. 283-294.
4. Fiebig, M. and Chen, Y., 1998, Heat transfer enhancement by wing-type longitudinal vortex generators and their application to finned oval tube heat exchanger elements, Proc. of the NATO Advanced Study Institute on Heat Transfer Enhancement of Heat Exchangers, pp. 79-105.
5. Zhu, J. X., Fiebig, M. and Mitra, N. K., 1995, Numerical investigation of turbulent flows and heat transfer in a rib-roughened channel with longitudinal vortex generators, Int. J. of Heat Mass Transfer, Vol. 38, No. 3, pp. 495-501.
6. Yoo, S. Y., 1997, Development of high efficiency fin-flat tube heat exchanger using vortex generators, MOTI Report, 95-P-10-2.
7. Kang, H. C., Kim, M. H. and Kim, M. S., 2000, A study on the thermal characteristics of finned-tube heat exchanger by using the liquid crystal technique, Korean Journal of Air-Conditioning and Refrigeration Engineering, Vol. 12, No. 4, pp. 414-421.
8. Goldstein, R. J., Yoo, S. Y. and Chung, M. K., 1990, Convective mass transfer from a square cylinder and its base plate, Int. J. of Heat and Mass Transfer, Vol. 33, pp. 9-18.