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Experimental Investigation of Heat Transfer Enhancement in a Circular Duct with Circumferential Fins and Circular Disks

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The characteristics of heat transfer and pressure drop for fully developed turbulent flow in a tube with circumferential fins and circular disks were experimentally studied. The various spacing and sizes of circumferential fins and circular disks were selected as design parameters, while the effects of these parameters on heat transfer enhancement and pressure drop were investigated. In order to quantify the effect of heat transfer enhancement and the increase of pressure drop due to the fins and disks in a tube, the Nusselt numbers and the friction factors for various configurations and operating conditions were compared to those for a corresponding smooth tube. The results showed that the heat transfer rate was significantly enhanced by increasing the height of circumferential fins and decreasing the pitch of circumferential fins. On the other hand, the influence of the disk size and the fin-disk spacing were not significant. Based on the experimental results, a correlation for estimating the Nusselt number was suggested.

Key Words : Heat Transfer Enhancement, Circumferential Fin, Circular Disk, Compact Heat Exchanger

Nomenclature

c_p : Specific heat (J/kgK)
D : Inside diameter of a tube (m)
f : Friction factor
I : Head loss (m)
h_t : Heat transfer coefficient of the tubeside (W/m²K)
k : Thermal conductivity (W/mK)
L_t : Length of the test section (m)
Nu : Nusselt number
p : Pressure (Pa)
Pr : Prandtl number
Re : Reynolds number
T : Temperature (℃)
V : Mean velocity (m/s)
ρ : Density (kg/m³)

Subscripts

b : Bulk
i : Inside
o : Outside
s : Shellside or smooth tube
t : Tubeside
lm : Log-mean

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1. Introduction

A gas-fired condensing boiler for a domestic appliance has been developed to utilize latent heat of combustion gases, which was conventionally discharged into atmosphere. The most important advantage of condensing boilers is that the thermal efficiencies are 12~17% higher than those of conventional boilers. In order to recover the latent heat of combustion gases from a heat exchanger of which size is similar to that of a conventional one, a highly efficient compact heat

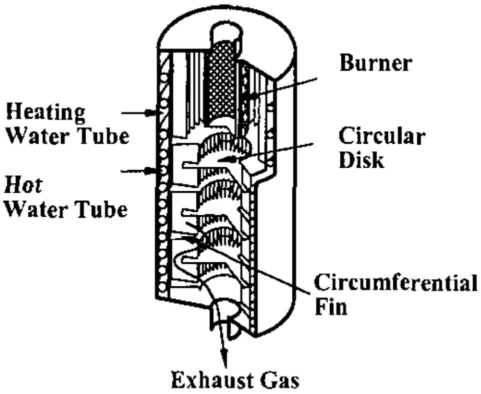

Fig. 1 Heat exchanger suggested for a condensing boiler

exchanger is required. As shown in Fig. 1, installation of circumferential fins and circular disks with longitudinal fins is suggested to increase the heat transfer rate of the heat exchanger for a condensing boiler. What these fins and disks do in a tube is to dramatically deflect the hot gas flow, which allows for good mixing and strong impingement against the tube wall. In order to obtain the optimal performance from the boiler with these inserts, a thorough analysis of heat transfer characteristics of the complex heat exchanger is needed.

Many researches about heat transfer enhancement with fins and baffles have been carried out. Kelkar and Patankar(1987) numerically studied heat transfer in a parallel plate channel with staggered fins. They calculated fully developed flow and heat transfer and showed the effects of fin spacing and height, Prandtl number, Reynolds number, and fin conductance on the Nusselt number and the friction factor. They concluded that heat transfer enhancement for high Prandtl number was more significant than that for low Prandtl number. Webb and Ramadhyani(1985) did similar research with a staggered rib. They discussed the effect of secondary recirculation on heat transfer and presented similar results for the Prandtl number effect to those of Kelkar and Patankar(1987). In addition, Berner et al.(1984) and Habib et al.(1994) carried out experimental research. Berner et al.(1984) visualized the flow patterns around the baffles and showed that the flow became turbulent around the Reynolds number of 1,670. Habib et al.(1994) studied velocity and pressure drop changes in a channel resulting from the different heights of baffles. These researches mentioned adopted parallel plate channels as their geometries. Although the characteristics of heat transfer and pressure drop may be qualitatively similar to those for a circular tube, it is difficult to realize the same results for a condensing boiler heat exchanger. On the other hand, extensive researches on a circular tube dealt with longitudinal fins rather than circumferential fins. Although Rowley and Patankar(1984) studied heat transfer in tubes with circumferential fins, it was not useful for a condensing boiler because they did not consider the circular disks between two consecutive fins. Jeon et al.(1999a) numerically investigated heat transfer and pressure drop for laminar flow in a tube with circumferential fins and circular disks. They presented the effects of the fin height and pitch and the size of the disks on heat transfer and pressure drop. However, it was found that most studies mentioned above were concerned with laminar flow while the flow in the heat exchanger of the condensing boiler was turbulent. On the other hand, Jeon et al.(1999b) did the numerical investigation for turbulent flow in the same geometry. It was known from the study that it was very difficult to simulate such a complex fluid flow and heat transfer accurately with the RNG k-ε model even though the results were qualitatively similar to those in the present study.

The characteristics of heat transfer and fluid flow for fully developed turbulent flow in a tube with circumferential fins and circular disks are experimentally investigated in this study. The heat exchanger shown in Fig. 1 has many longitudinal fins, but only circumferential fins and circular disks are considered in this investigation. A variety of sizes of circumferential fins and circular disks, location of disks, and the Reynolds number are used as important design parameters. The effects of these parameters on heat transfer enhancement and pressure drop are studied. In addition, a correlation for estimating the Nusselt number for similar situation is suggested from the experimental results.

2. Experiment

Figure 2 shows a schematic of the experimental apparatus. It consists mainly of a blower, an air heater, a test section, and an orifice plate. The test section is an annular-type parallel flow heat exchanger. Hot air and cold water flow through the tubeside and the shellside, respectively. The temperature of hot air ranges from 150 to 200℃. The outside diameter and length of the test section are 74.8 mm and 340 mm, respectively. The inside diameter of the tube is 62.6 mm. Circumferential fins and circular disks are inserted into the tube

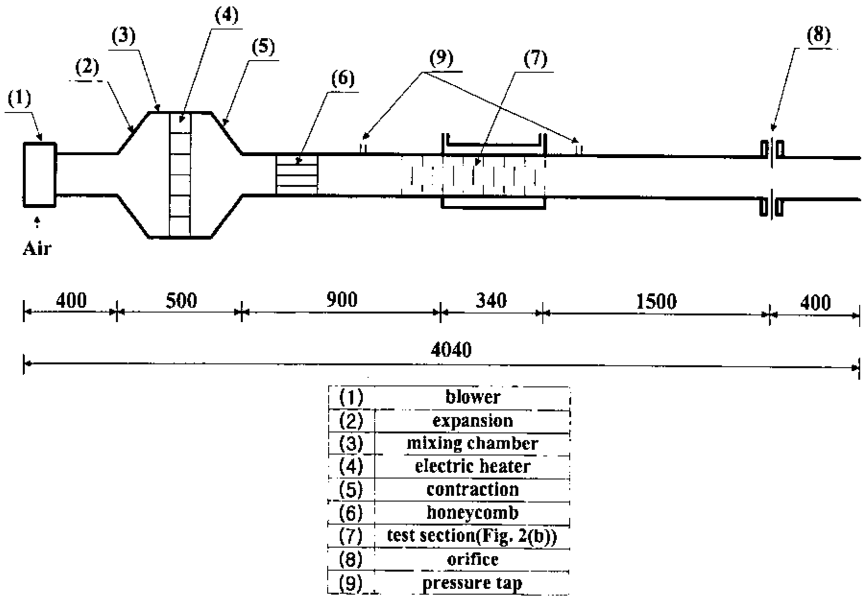

Fig. 2(a) Schematic of the experimental apparatus

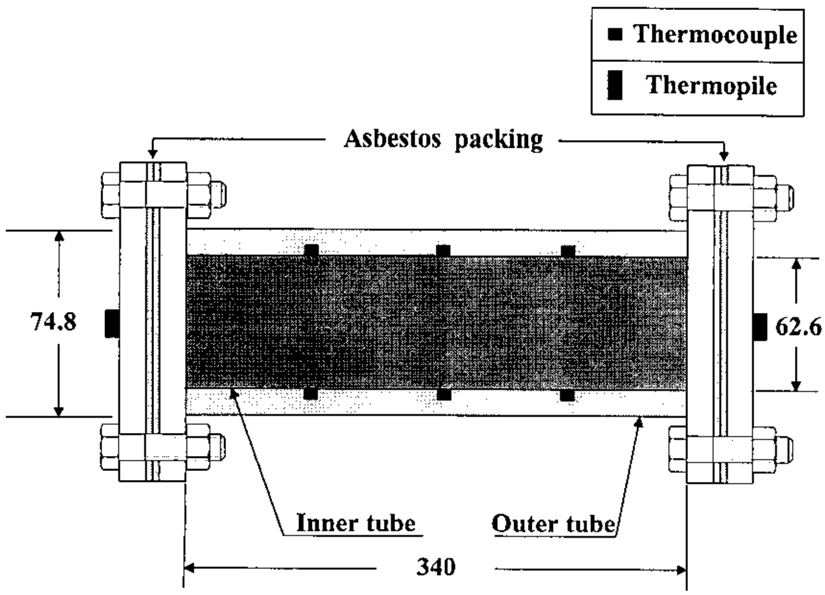

Fig. 2(b) Test section

and two pairs of fins and disks are placed before the test section in order to develop the flow.

The flow rate of hot air is measured by an orifice plate and a micro manometer. The Reynolds number of the tubeside ranges from 3,000 to 7,000, which covers typical operating conditions of a condensing boiler. Thermopiles that consist of eight T-type thermocouples are located at the inlet and outlet of the tube to measure the bulk mean temperatures of hot air. Nine thermocouples are attached to the tube wall; the average temperature of these is assumed to be the wall temperature. Cold-water temperatures at the inlet and outlet are obtained by thermocouples, the flow rate being measured by an electrical weighing scale.

Pressure drop is measured by an adiabatic experiment. Two pressure taps are installed before and after the test section. It was known that the flow in this particular geometry was developed very quickly from other researches (Berner et al., 1984, Habib et al., 1994). In addition, Jeon et al. (1999b) numerically showed that the flow was fully developed after passing the third pair of a fin and a disk. In order to consider the entrance effect, pressure drops for eight pairs of fins and disks are measured first. The pressure drops for three pairs are then obtained in the same operating condition. Dividing the difference between these two pressure drops by five, the average pressure drop for a pair of fins and a disk is found.

In order to verify the experimental setup for this study, the friction factors and the Nusselt numbers for a smooth tube were measured. Good agreement was obtained compared with the well-known correlations. The heat balance between heat loss from the tubeside and heat gain of the shellside was checked. The data were obtained after confirming that the deviations were consistently less than 5%.

3. Data Reduction

The friction factor is defined as follows:

$$f=\frac{-(dp/dx)D}{\rho V^2/2} \tag{1}$$

and the friction factor for a smooth tube is calculated from the correlation developed by Petukhov (1970) as follows:

$$f_s=(0.790\ln \mathrm{Re}-1.64)^{-2} \text{ for } 3{,}000<\mathrm{Re}<5\times10^6 \tag{2}$$

The average Nusselt number of the tube side is defined as follows:

$$Nu=\frac{h_t D}{k}=\frac{\dot{m}c_p(T_{t,i}-T_{t,o})}{k\pi L_t(T_{t,b}-T_w)} \tag{3}$$

For the shellside fluid stream, the average bulk temperature is determined as:

$$T_{s,b}=\frac{T_{s,i}+T_{s,o}}{2} \tag{4}$$

For the tubeside fluid stream, the length-averaged bulk temperature is obtained as follows (Shome, 1995):

$$T_{t,b}=T_{s,b}+\Delta T_{lm} \tag{5}$$

The average Nusselt number of a smooth tube as a reference for the study is calculated using the correlation of Gnielinski (Incropera and DeWitt, 1996), which shows a consistent agreement with a low Reynolds number and is given by:

$$Nu_s=\frac{(f/8)(\mathrm{Re}-1{,}000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3}-1)} \tag{6}$$

The inside diameter of the tube is selected as the characteristic length for the geometry considered because it is easy to compare the results with those for a smooth tube.

4. Results and Discussion

The geometrical details of the circumferential fins and the circular disks that are tested in the present study are shown in Table 1 and illustrated in Fig. 3. As shown in the table, the size and location of the disk, and the height and pitch of the fin were selected as the important design parameters. The effects of these on heat transfer enhancement and pressure drop along the Reynolds number were investigated.

In Figs. 4 and 5, the effects of disk radius on heat transfer enhancement and pressure drop are shown. The Nusselt number increases with the Reynolds number and the size of the disk. How-

Table 1 Baffle geometries tested

Disk radius (L)	H=1D, F=0.35, S=H/2	L=0.28D, 0.32D, 0.35D, 0.38D
Spacing (S)	H=1D, F=0.25D, L=0.38D	S=0.3H, 0.4H, 0.6H, 0.7H
Fin height (F)	H=1D, F=0.25D, L=0.38D	F=0.25D, 0.28D, 0.32D, 0.35D
Pitch (H)	F=0.25D, S=H/2, L=0.38D	H=0.6D, 0.8D, 1D, 1.2D

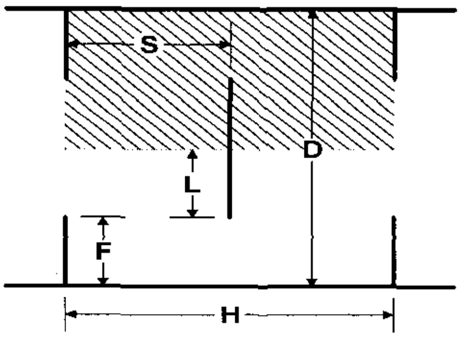

Fig. 3 Geometry considered

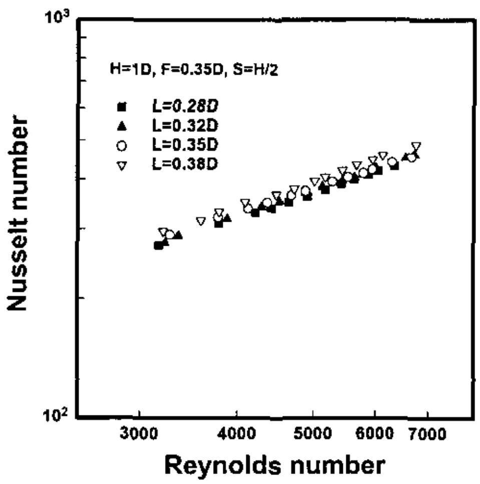

Fig. 4 The effect of the disk size on the Nusselt number

ever, the effect of the disk size is not significant. In addition, the Nusselt numbers for these geometries are about 23~25 times greater than that for

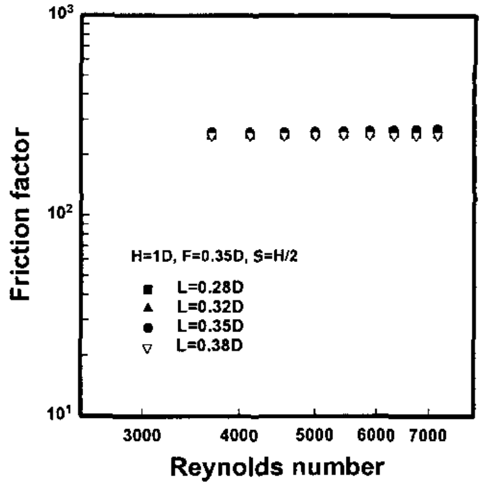

Fig. 5 The effect of the disk size on the Friction factor

a smooth tube. Generally speaking, the convection heat transfer coefficients for liquids are about 10~80 times greater than those for gases. If it is feasible to make it 25 times greater than it used to be, then, we can make the tubeside heat transfer coefficient have the same order of magnitude of the shellside for water. Consequently, the overall heat transfer resistance of the heat exchanger for the condensing boiler decreases significantly.

On the other hand, the friction factor of the Reynolds number ranged between 3,000~7,000 does not change very much with the Reynolds number and the disk size. It is very different from the typical behavior of the friction factor for a smooth tube, which decreases gradually by increasing the Reynolds number. However, a similar relation between the friction factor and the Reynolds number could be seen from other experimental investigations for similar geometrical setups (Berner et al., 1984 and Habib et al., 1994). Although they used a somewhat different arrangement of baffles, it was found that the relation between the friction factor and the Reynolds number was weak. In addition, the size of the disk cannot change the friction factor significantly. According to Jeon et al. (1999b), it was shown that the flow patterns and the length of the recirculation zone behind the fin did not change very much although the size of the disk varied. However, the friction factor for this setup

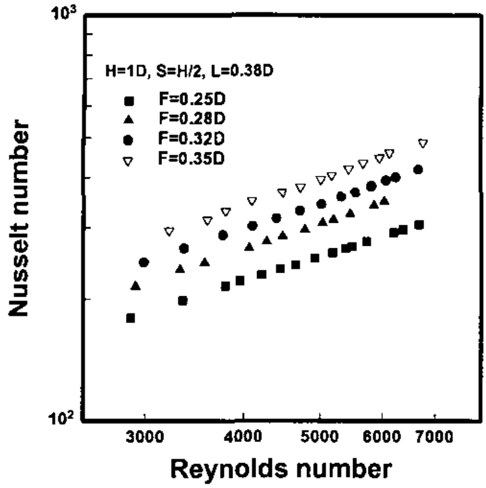

Fig. 6 The effect of the fin height on the Nusselt number

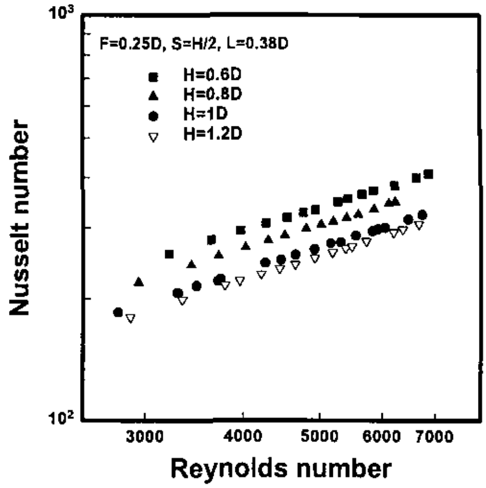

Fig. 8 The effect of the fin pitch on the Nusselt number

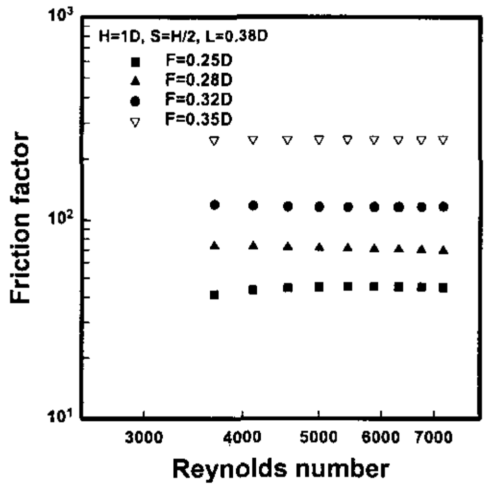

Fig. 7 The effect of the fin height on the Friction factor

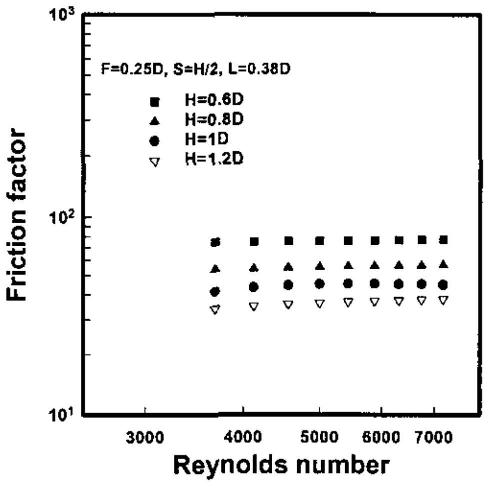

Fig. 9 The effect of the fin pitch on the Friction factor

is about 6500 times greater than that for a smooth tube. It is tremendous increase for pressure drop. Considering that the inside diameter and the length of the heat exchanger of the condensing boiler are about 250 mm and 300 mm, respectively, pressure drop for this particular application can be estimated less than 60 Pa, which is not a serious limitation for a boiler.

In Figs. 6 and 7, the effect of fin height on heat transfer enhancement and pressure drop are shown. The Nusselt number and the friction factor are very sensitive to the fin height. As the circumferential fin height increases, the flow is more accelerated because the cross section at the fin decreases. After striking the circular disk, the main stream is seriously deflected and impinged against the tube wall. Consequently, the effect of heat transfer enhancement becomes significant. It was shown in the numerical investigation of Jeon et al. (1999b) that the flow pattern was significantly affected by the fin height. While the Nusselt number is approximately 15~23 times greater than that for a smooth tube, the corresponding pressure drop is 1,200~6,500 times greater. Once again, the friction factor does not depend upon the Reynolds number from the figure.

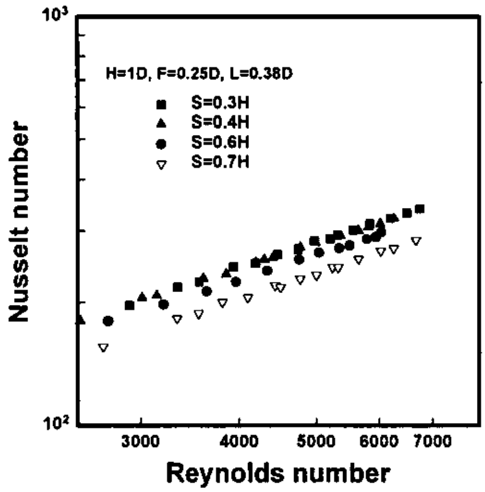

Fig. 10 The effect of the fin-disk spacing on the Nusselt number

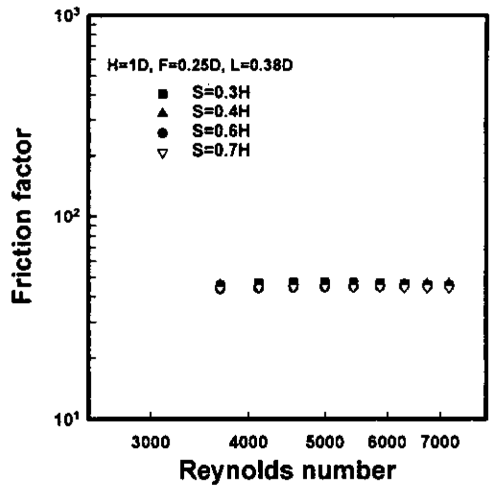

Fig. 11 The effect of the fin-disk spacing on the Friction factor

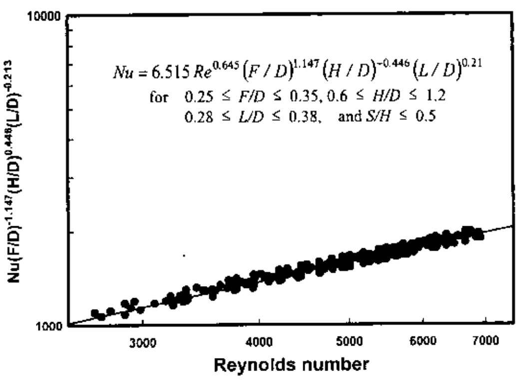

Fig. 12 A correlation for estimating the Nusselt number

The effects of the fin pitch on heat transfer enhancement and pressure drop are shown in Figs. 8 and 9. The Nusselt number and the friction factor increase as the fin pitch decreases. It is because decrease of the fin pitch makes the flow mixing more violent. As the fin pitch decreases, deflection of the flow on the disk, vortices generated at the tip of the fin and the disk, and flow impingement against the tube wall becomes significantly stronger. If the fin pitch is smaller than 1.0D, the increasing rate of the Nusselt number becomes larger. However, the increasing rate of the friction factor increases also. Consequently, caution must be exercised when decreasing the fin pitch.

In Figs. 10 and 11, the effects of the fin-disk spacing on heat transfer enhancement and pressure drop are given. The Nusselt number increases gradually as the fin-disk spacing decreases. However, the increase of the Nusselt number becomes saturated once the spacing S reaches 0.4H. Consequently, the effect of the spacing on heat transfer is not significant because the spacing is normally smaller than 0.5H for real application. The heat transfer rate does not change very much once the spacing is smaller than 0.5H. On the other hand, the pressure drop does not change if the fin-disk spacing varies between 0.3H to 0.7H. Also, the pressure drop is independent of the Reynolds number. Qualitatively similar behaviors of the Nusselt number and the friction factor were observed from the work for both laminar and turbulent flows by Jeon et al. (1999a and 1999b). It was known that the characteristics of heat transfer of a laminar flow for this particular application were qualitatively similar to those of a turbulent flow.

5. Correlation

From the experimental results, a correlation for estimating the Nusselt number for the particular arrangement is suggested as follows:

$$Nu = 6.515 Re^{0.645} (F/D)^{1.147} (H/D)^{-0.446} (L/D)^{0.213} \quad (7)$$

for $0.25 \leq F/D \leq 0.35$, $0.6 \leq H/D \leq 1.2$, $0.28 \leq L/D \leq 0.38$, and $S/H \leq 0.5$. This is shown in Fig. 12. The deviations between the data are about less than 6.7% for the operating conditions used for the present study. As mentioned above, the fin-disk spacing S is not important for heat transfer design once it is less than 0.5H. Hence, it is excluded from the correlation suggested in the present study.

6. Conclusions

Heat transfer of turbulent flow in a circular tube with circumferential fins and circular disks were experimentally investigated. The heat transfer coefficient increased dramatically with these inserts. However, careful consideration was required because of huge increase in the friction factor accompanied by heat transfer enhancement. Hence, the heat transfer enhancement technique suggested in this study could be applicable to particular situations in which the pressure drop is not a serious limitation for the system such as the heat exchanger for a condensing boiler. Also, the results indicated that the effects of the Reynolds number, the pitch of the fins, and the fin height on heat transfer were significant while the fin-disk spacing and the disk size could not change the heat transfer characteristics significantly. Based on the experimental results, a correlation for estimating the Nusselt numbers for various geometrical setups and operating conditions resulted consistently. Information obtained from the study can be utilized for the optimization of the heat exchanger for a condensing boiler.

Acknowledgments

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