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TURBULENT FLOW AND CONVECTIVE HEAT TRANSFER IN AN ANNULUS WITH PERFORATED DISC-BAFFLES

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Abstract

Measurements were carried out to investigate the turbulent flow and heat transfer behavior in an annulus with perforated disc-baffles aligned along the inner heated tube surface, using air as a working fluid. The test section is a horizontal annular passage formed by two concentric tubes with an aspect ratio (D_i/D_o) of 0.3. The outer one is a circular (PVC) tube of 72 mm inner diameter, 3 mm thickness. The inner tube is a circular brass tube of 22 mm outer diameter and 6 mm thick. Heat is only transferred from the outer surface of the inner tube while outer tube is well insulated. The heater used for heating the brass tube is ready-made heater of outer diameter 10 mm which is fitted into the brass tube. The effects of the baffle spacing, the baffle open area ratio, and flow Reynolds number on the thermal performance were examined. Circular steel disc-baffles with dimensions of 42 mm outer diameter, 22 mm inner diameter, and 1 mm thickness are used in the present study. The disc-baffle outer diameter to the annulus outer diameter was fixed at 0.58. The present work was conducted for solid and perforated disc-baffles with baffle open-area ratio of 6%, 12 %, and 18% within a range of flow Reynolds number (based on the annulus equivalent diameter) from about 20,000 to 50,000, and for three different values of baffle pitch to equivalent diameter ratios of 2, 4, and 8. The results show that the Nusselt number enhancement ratio increases with increasing the baffle open area ratio and decreases with increasing the baffle pitch to equivalent diameter ratio. Also, for a given Reynolds number and baffle pitch ratio, the normalized friction factor decreases monotonically as the baffle open area increases from 0 to 18 % because of less cross-sectional blockage for baffles with larger open area ratio. The perforated disc-baffles with open area ratio of 18% and S/D_e=2 perform the highest Nusselt number enhancement ratio, about 3 times more than that for the smooth annulus corresponding to 6.6 fold increase in the flow friction factor ratio. The perforated-baffled annulus has a higher thermal performance than the solid-baffled annulus for all baffles configurations studied. The perforated baffles with open area ratio of 18% perform the highest efficiency indices under a constant power constraint. New correlations were obtained for the effect of Reynolds number, baffle pitch to equivalent diameter ratio, and open area ratio on both the

Nusselt number enhancement and the flow friction factor ratios.

1. Introduction

The flow separation in ducts with segmented baffles has many engineering applications, e.g. shell-and-tube heat exchangers with segmented baffles, air-cooled solar collectors, internally cooled turbine blades, and cooling of microelectronics. Augmentation techniques usually employ baffles attached to the heated surface so as to provide an additional heat transfer surface area and to promote turbulence. The presence of baffles causes the flow to separate, reattach and create reverse flow. Recently, many investigations have been focused on the baffle-walled channel heat exchangers. Most studies discussed the optimal baffle geometry that enhance heat transfer performance for a given pumping power or flow rate. While the use of solid baffles results in significant heat transfer enhancement, the associated increase in pressure drop and higher local thermal stress at the root of the baffle is of concern. Thus warranting the exploration of the use of perforated baffles to enhance the heat transfer while keeping the pressure drop to a minimum.

Saffar-Avval et al [1] studied the effect of baffle spacing on heat transfer area and pressure drop. They concluded that baffle spacing has a decisive effect on pumping power, and noticeable effect on required heat transfer area. They developed a guideline to calculate optimum baffle spacing for single phase shell and tube heat exchangers. They also used the same optimization procedure to find some correlations in order to calculate baffle spacing in all types of single phase shell and tube heat exchangers [2]. Afify and Abd-Elghany [3] carried out an experimental study of the turbulent flow and heat transfer in a circular pipe with doughnut-and-disc baffles at uniform wall heat flux condition. The experimental runs were carried out for different Reynolds numbers and baffle spacing ratios. Li and Kottke [4] investigated experimentally the local heat transfer and pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles for different baffle spacing. Li and Kottke [5] used a mass transfer measuring technique to visualize and determine the shell side local heat transfer coefficients at each tube in two representative baffle compartments of a shell and tube heat exchanger with disc and doughnut baffles. Compared to the single segmental baffle, the disc and doughnut baffles have a higher effectiveness of heat transfer to pressure drop. Afify et al [6] studied the turbulent flow and heat transfer behavior in a circular tube with disc-baffles aligned along tube axis, using air as the working fluid. The effects of the disc-baffle diameter-to-tube diameter ratio, disc-baffle pitch-to-tube diameter ratio, and flow Reynolds number on the local and average heat transfer coefficients and friction factors were studied. Soltan et al [7] developed a computer program which enables designers to determine the optimum baffle spacing for segmentally baffled shell and tube condensers. Zhnegguo et al. [8] studied experimentally the heat transfer and pressure drop of helically baffled heat exchanger combined with petal-shaped finned tubes for oil cooling with water as coolant. Tandiroglu and Ayhan [9] conducted an energy dissipation analysis to evaluate the performance of nine circular tubes with baffle plate inserts having different pitch to tube inlet diameter ratio and baffle orientation angle in the range of Reynolds number $3000 \leq Re \leq 20{,}000$ for the case of constant heat flux. The experimental study by Mobarak et al [10] and Habib et al

[11] of the flow and heat transfer across segmental baffles indicates significant effect of baffle height and baffle material inside a rectangular duct on the flow pattern, heat transfer, and pressure drop for a fixed baffle spacing. Augmentation of heat transfer was obtained with increase in Reynolds number, thermal conductivity of baffle, and baffle height. Wang et al [12] investigated the enhancement of heat transfer due to unsteady laminar flow in channels with in-line and staggered baffles. The in-line baffle configuration appears to provide a slightly larger Nusselt number than the staggered case at the same Reynolds number. Also, the associated friction factors are nearly the same. Chen and Chen [13] studied the effect of the distance between a single baffle and the solid wall on the local heat transfer in a rectangular channel due to an oscillatory flow. Hwang [14] conducted experiments to study turbulent heat transfer and fluid flow in a porous baffled channel. The porous baffles were mounted on the top and bottom walls in a staggered manner. In addition, experiments were conducted with solid baffles for the sake of comparison. The results showed that the porous baffled channel have a significantly lower friction. Dutta et al [15] investigated experimentally frictional loss and heat transfer behavior of turbulent flow in a rectangular channel with isoflux heating from the upper surface for different sizes, positions, and orientations of inclined baffles attached to the top and bottom surface. Both solid and perforated baffles were used. Rachedi and Chikh [16] numerically studied forced convection cooling in the presence of porous inserts in electronic devices. Results showed that the temperature dropped down by half. Yilmaz [17] carried out an experimental work to determine how a single baffle placed at the inlet of a rectangular channel affects heat transfer and friction characteristics. The baffle was mounted on the upper inner surface of the channel, and the channel base was heated by a constant heat flux. Two different clearance ratios and four baffle inclination angles were used. The performance analysis by the equal friction factor criterion showed that the use of the inlet baffle is not thermodynamically advantageous on the basis of heat transfer enhancement. Ko and Anand [18] carried out an experimental investigation to measure a module average heat transfer coefficients in uniformly heated rectangular channel with wall mounted porous baffles, which were mounted alternatively on top and bottom walls. The use of porous baffles resulted in heat transfer enhancement as high as 300% compared to heat transfer in straight channel with no baffles. Yang and Hwang [19] present the numerical predictions on the turbulent fluid flow and heat transfer characteristics for rectangular channel with porous baffles which are arranged on the bottom and top walls in a periodically staggered way. Afify et al [20] investigated experimentally the effects of the baffle alignment (in-line and staggered), Reynolds number, open area ratio, baffle pitch and baffle height on the heat transfer enhancement, friction factor, and the thermal performance for turbulent flow of air in a rectangular duct with perforated baffles. Dutta and Hossain [21] investigated the local heat transfer characteristics and the associated frictional head loss in a rectangular channel with inclined solid and perforated baffles. A combination of two baffles of same overall size was used. The upstream baffle was attached to the top heated surface, while the position, orientation, and the shape of the other baffle were varied to identify the optimum configuration for enhanced heat

transfer. Karwa et al [22] studied the heat transfer and friction in rectangular ducts with solid and perforated baffles attached to one of the broad walls. The baffled wall of the duct was uniformly heated while the remaining three walls were insulated. Performance comparison with the smooth duct at equal pumping power shows that the baffles with the highest open area ratio give the best performance. Lin [23] explored the local heat transfer in a rectangular channel with baffles and analyzes the experimental results of baffles with different heights and pores in the event of five Reynolds numbers and three heating quantities.

From the above-mentioned analysis of the available literature, it is found that there have not been any investigations on enhancing the heat transfer in an annulus with perforated disc-baffles. Therefore, the objective of the present work is to study the turbulent flow and heat transfer behavior in an annulus with perforated disc-baffles aligned along the inner heated tube surface, using air as a working fluid. The experimental measurements were performed to investigate the effect of the baffle spacing, the baffle open area ratio, and flow Reynolds number on the thermal performance.

Nomenclature

A cross-sectional area of the test-section
c_p air specific heat
D_e annulus equivalent diameter (D_e=D_o-D_i)
D_i inner-tube outer-diameter of the annulus
D_o outer-tube inner-diameter of the annulus
d_i inner diameter of the disc baffle
d_o outer diameter of the disc baffle
H baffle height, $H=(d_o-d_i)/2$
h heat transfer coefficient
k air thermal conductivity
L test-annulus length
m air mass flow rate
P pressure
q" heat flux
S spacing between baffles
T temperature
V axial flow velocity
x axial distance from annulus inlet

Greek letters:

Δ difference
ν air kinematic viscosity
ρ air density
η baffled-annulus efficiency index

Subscripts:

f for fluid
in at annulus inlet
m mean value
o case of smooth annulus
out at annulus exit
w for inner tube outer surface
x local value

Dimensionless terms:

B open area ratio
F Fanning friction factor
F^+ Nikuradse's friction similarity factor
H^+ Sabersky's heat transfer similarity factor
Nu average Nusselt number
Pr Prandtl number
Re Reynolds number
Re^+ Modified Reynolds number
St Stanton number

2. Test rig and measuring instruments

The experimental apparatus employed in this investigation is shown schematically in Fig.(1-a). It consists of a blower assembly, an orifice flow meter, a main entrance duct, a test-section, and instrumentations to measure temperatures, pressure drop, air flow rate, and electrical power input. The details of the apparatus are depicted as follows:

(a) Air blower: A centrifugal type air blower, driven by an electric motor of 5.5 hp capacity and 3000 rpm normal speed, is used to supply the system with air at the required flow rate. The air flow rate can be controlled via the blower intake gate.

(b) Test section: The test section is a horizontal annular passage formed by two concentric tubes with an aspect ratio (D_i/D_o) of 0.3. The outer one is a circular (PVC) tube of 72 mm inner diameter, 3 mm thickness. The inner tube is a circular brass tube of 22 mm outer diameter and 6 mm thick. Heat is only transferred from the outer surface of the inner tube while outer tube is well insulated. The test section has 2 m length, and an additional 2 m hydrodynamic entry length is allowed upstream of the test section inlet. The test section is connected to the entrance tube through a flexible connection. The heater used for heating the brass tube is ready-made heater of total resistance 36.6 ohm with stainless steel sheath of outer diameter 10 mm and length of 2 m which is fitted into the brass tube. The outer tube is insulated with a high temperature insulating asbestos rope, which is 13 mm diameter wound around the tube and provides high thermal resistance. A longitudinal section of the baffled annulus is illustrated in Fig.(1-b).

(c) Disc-baffles: Circular steel disc-baffles are used in the present study with dimensions of 42 mm outer diameter, 22 mm inner diameter, and 1 mm thickness. The disc-baffles are attached to the inner heated tube surface by a thin layer of thermal glue. The present perforated baffles having circular holes of 1.5 mm diameter machined normal to the disc-baffles. The disc-baffle open area ratio is increased by increasing the number of the holes. Three rows of perforations are in-line alignment. The open area ratio of the perforated baffle is defined as the ratio between the circular holes area ($n \pi r^2$) and the baffle frontal area [$\pi/4(d^2_o- d^2_i)$]. The present work is conducted for solid and perforated disc-baffles with baffle open-area ratio of 6%, 12 %, and 18%, and for three different values of baffle pitch to annulus equivalent diameter ratios of 2, 4, and 8. The ratio of the disc-baffle outer diameter to the annulus outer diameter is fixed at 0.58. Details of the test annulus cross section are shown in Fig.(1-c).

d) Instrumentation: The surface temperature of the inner heated tube was measured at several axial positions (14 different test positions with higher concentration at annulus inlet) using calibrated 0.5 mm teflon insulated alumel-chromel (K type) thermocouples. The air flow radial temperature profile is measured via radiation shielded thermocouples mounted on vertical traverse mechanisms at annulus inlet and exit, respectively. The thermocouple probe with its radiation shield is made with small sizes compared with the annular flue gap. To avoid flow disturbance, the thermocouple probe with its shield are resided in a tubular-cavity which is externally assembled normal to the test annulus such that the probe appears only in the flow field at the instant of reading. All thermocouples used in the present experimental study are calibrated prior to their installation in the apparatus. These thermocouples are wired into a thermocouple selector switch which is connected to a digital compensating thermometer able to read temperature to one tenth of a degree Celsius. A voltage regulator of output voltage varies from 0 to 250 Volt is used to provide and control the power input to the heater. The power input to the heater is estimated by measuring the voltage drop across the heater using a digital multi-meter (one decimal point) and the known resistance of the heater, which is measured by the same multi-meter. The air flow rate was measured by a calibrated orifice-meter. Two pressure taps were located just upstream and downstream of the test section to measure the pressure drop along the test section. The pressure drop across the orifice-meter and the test section were measured by a system of U-tube manometers. The steady state was assumed to be established if the inner tube surface temperature and air flow temperatures changed within 0.2 ^{o}C during 15 minutes. Then, the data of electric power input, the temperature distribution on the heating surface, inlet and exit air flow temperatures, and the manometers readings were recorded.

3. Data reduction

The power input to the heaters was computed from measurements of the voltage and current and then the average heat flux to the test section, based on the inner-tube surface area, was obtained. The heat lost by axial conduction, radial conduction through the insulation layers and by radiation from the two ends of the test section is very small and can be neglected for the tested range of parameters. Also, the average heat flux to the test section was obtained by measurement of the average increase in bulk temperature of flowing air across the test section. The average increase in bulk temperature may be found by measuring the bulk temperatures at the entrance and exit of the test section. Since the values of the mass flow rate and the specific heat of air are known quantities, the heat flux may be calculated. This measurement was used just as a check of the heat flux measurement by the power input to the electrical heater. The heat balance was found to be less than 5 %.

The annular test section is divided into 14 axial segments with unequal lengths (highly concentrated at the entrance) and the local heat transfer coefficient at each annulus segment was calculated simply by;

$$h_x = \frac{q''}{\overline{T}_{w,x} - \overline{T}_{f,x}} \qquad (1)$$

where,

q'' is the heat flux ($Q_{heater}/\pi D_i L$)

$T_{w,x}$ is the temperature of the outer surface of the inner tube of the annulus-segment.

$T_{f,x}$ is the local flow bulk temperature at the annulus segment, calculated from the heat balance for each segment as follows:

$$m\, C_p(T_{f,x} - T_{f,x\text{-}\Delta x}) = q''\,(\pi D_i \Delta x) \qquad (2)$$

Thus, the flow outlet temperature at the stream-wise station x is:

$$T_{f,x} = T_{f,x\text{-}\Delta x} + q''\,(\pi D_i \Delta x)/m\, C_p \qquad (3)$$

Where, Δx is the annular segment length. Therefore, the flow mean temperature for each segment was calculated as the arithmetic mean between the inlet and outlet as:

$$\overline{T}_{f,x} = \left(T_{f,x} + T_{f,x\text{-}\Delta x}\right)/2 \qquad (4)$$

Also, the local flow bulk temperature at the annulus segment was checked with the mean value calculated by integrating the measured radial temperature profiles at inlet and exit, respectively.

The local Nusselt number based on the annulus equivalent diameter is calculated, and in accordance the average Nusselt number is calculated by simply integrating the local values as follows;

$$Nu_x = \frac{h_x D_e}{k} \qquad (5)$$

$$Nu = \frac{1}{L} \int_{x=0}^{L} Nu_x \, dx \qquad (6)$$

Also, the flow Reynolds number $Re = \frac{V D_e}{\nu}$, Stanton number $St = \frac{Nu}{Re\,Pr}$, and Prandtl number Pr are calculated, while the air properties are evaluated at the flow bulk temperature, $(T_{in}+T_{out})/2$.

The flow friction in terms of Fanning friction factor is estimated from the measurements of the pressure drop along the annular test section as:

$$F = \frac{\Delta P}{2\rho V^2} \frac{D_e}{L} \qquad (7)$$

Moreover, the flow friction and heat transfer data are reformulated in terms of Nikuradse's friction similarity factor F^+ and Dipprey and Sabersky's heat transfer similarity factor H^+ which have been reported by Abdel-Moneim and Co-workers [24,25] and expressed as follows:

$$F^+ = \sqrt{\frac{2}{F}} + 2.5\,\ell n\left(\frac{2H}{D_e}\right) + 3.75 \qquad (8)$$

$$H^+ = \left(\frac{F}{2\ St} - 1\right) / \sqrt{\frac{F}{2}} + F^+ \qquad (9)$$

The efficiency index, which is defined by the ratio between the Nusselt number enhancement ratio (Nu/Nu_o) to the friction factor ratio (F/F_o), $\eta = \frac{Nu/Nu_o}{F/F_o}$ is an important criterion in evaluating any technique for heat transfer enhancement. The modified Reynolds number Re^+, based on the baffle height $(H = (d_o-d_i)/2)$, is also calculated as:

$$Re^+ = Re \frac{H}{D_e} \sqrt{\frac{F}{2}} \qquad (10)$$

An uncertainty analysis was performed to evaluate the accuracy of the present measurements. Percentages of uncertainties in the measurements of electric voltage, resistance, heating surface area, surface temperature, and flow mean temperature were 0.125%, 0.137%, 0.25%, 0.333%, and 0.666%, respectively. Therefore, a total uncertainty of 1.51% was found for the heat transfer coefficient. Moreover, a maximum uncertainty of 3.0% was estimated in the reported Reynolds number due to the measurement of the air discharge using an orifice instrumented with a differential manometer. Also, a maximum uncertainty of 4 % was estimated for the friction factor.

4. Results and discussions

A preliminary series of experiments was carried out for smooth annulus flow to check the validity of the present measurements. The average Nusselt number and the Fanning friction factor were experimentally determined for a smooth annulus, for different mass flow rates of air corresponding to Reynolds numbers ranging from 20,000 to 50,000. Values of these parameters so obtained were compared with the predicted values of Nusselt number using Dittus and Boelter's correlation and friction factor using Blasius's equation for smooth ducts listed in [20] and good agreements were found as shown in Figs.(2, 3). The present results for Nusselt number and friction factor for smooth annulus flow were correlated and the following correlations were obtained:

$$Nu_o = 0.031\ Re^{0.747} \qquad (11)$$

and

$$F_o = 0.0636\ Re^{-0.216} \qquad (12)$$

These correlations are valid within ±8% and 6%± maximum deviations (for Nu_o and F_o respectively) with the present experimental data within a range of Reynolds number from 20,000 to 50,000 for air. These good agreements give assurance and confidence in the present experimental data. Beside establishment of the present experimental technique, the preliminary experiments were carried out to normalize the present experimental results for disc-baffled annulus. The results of the average Nusselt number for solid and perforated disc-baffles at different baffle open area and pitch to equivalent diameter ratios within a range of flow Reynolds number from about 20,000 to 50,000 are shown in Fig.(2). The presence of baffles in the flow field interrupt the wall boundary layer causing flow separation, reattachment and also generate intense, highly unsteady secondary flows and vortex pairs, which increase secondary advection and turbulent transport over the entire annulus cross section. This enhance the heat transfer in general. Higher Nusselt number values were observed for the perforated disc-baffled annulus. The results show that the average Nusselt number increases with increasing the open area ratio and decreases with increasing the baffle pitch to equivalent diameter ratio. Also, the associated pressure loss due to different disc-baffles configurations was measured and the Fanning friction factor was calculated. Fig.(3) shows the results for Fanning friction factor for the investigated disc-baffles configurations. As expected higher heat transfer coefficients were accompanied by higher friction factors. It is noted that, for all disc-baffles investigated, as the baffle pitch to equivalent diameter ratio increase the corresponding Fanning friction factor decrease. Also, the friction factor for the perforated disc-baffled annulus is lower than that for the solid disc-baffled annulus.

The difference in the behavior of the solid and perforated baffles can be attributed to the different flow structures for the two. In the case of the solid baffles, there is separation of the flow at the baffles and its reattachment downstream the baffles. The laminar sublayer is practically completely destroyed at the point of reattachment. However, a zone of recirculating flow appears behind the baffle and there is local heat removal by the shedding of vortices only. The acceleration produced by the baffles also contributes to the heat transfer enhancement. The friction factor increases significantly because of the form drag. In the case of perforated baffles, a part of the flow passes through the holes in the baffle reducing or eliminating the hot zone and form drag. The jet like flow through the holes of the baffles and interacts with the reattaching and accelerating flow passing over the baffles. The effect of the reattachment is somewhat reduced but the flow through the perforations creates turbulence at the wall which causes a decrease in the thickness of the laminar sublayer and mixing of the fluid near the wall with the turbulent core. With the increase in the open area ratio, the reattachment effect reduces further but the mixing at the wall increases because of a greater proportion of the flow through the perforations. The Reynolds number dependence of the average Nusselt number enhancement ratio (Nu/Nu_o) for different baffle open area and pitch to equivalent diameter ratios was plotted in Fig.(4), where the Nusselt number is normalized by the present data for the smooth annulus. The enhancement ratio (Nu/Nu_o) slightly increases with increasing Reynolds number. The heat transfer enhancement is affected by the frequency of flow acceleration which decreases with increasing the stream-wise pitch and a maximum enhancement in heat transfer was obtained for disc-baffles with the ratio $S/D_e = 2$, minimum value in the present work, as shown in Fig.(4-a). Also, the

Nusselt number enhancement ratio increases with increasing the baffle open area ratio. The perforated disc-baffles with open area ratio of 18% and $S/D_e = 2$ perform the highest Nusselt number enhancement ratio, about 3 times more than that for the smooth annulus with no baffles. The results of Ko and Anand [18] for uniformly heated rectangular channel with wall mounted porous baffles with baffle height to duct equivalent diameter ratio of 1/3 and baffles with 40 pores per inch are presented in Fig.(4-a) for comparison. Also, the results of Afify et al [20] for perforated baffled rectangular duct with baffle open area ratio of 15% and baffle pitch to duct height ratio of 3 are displayed. Some discrepancies are noticed when comparing the present results with the results of [18] and [20]. These may be due to the effect of the great difference in the vortical flow structures. The friction factor of the disc-baffled annulus is normalized by the present data for the smooth annulus. The variation of the Fanning friction factor ratio with Reynolds number at different baffle open area and pitch to equivalent diameter ratios are shown in Fig.(5). The friction factor ratio increases as the Reynolds number increases for fixed values of open area and baffle pitch ratios. The normalized friction factor ratio for the perforated baffle geometry is smaller than that for the solid baffle geometry at the same baffle pitch. For a given Reynolds number and baffle pitch ratio, the normalized friction factor decreases monotonically as the baffle open area increases from 0 to 18 % because of less cross-sectional blockage for baffles with larger open area ratio. Also, the friction factor ratio decreases with increasing the disc-baffle pitch ratio at the same Reynolds number and open area ratio.

Figure (6) shows the variations of the friction similarity factor with the modified Reynolds number for the baffled annulus at different baffle open area and S/D_e ratios. The friction similarity factor decreases as the modified Reynolds number increases. The perforated disc-baffles with open area ratio of 18 % have the highest friction similarity factor, which implies the lowest pressure drop.

The present data of the friction similarity factor were introduced in calculating the heat transfer similarity factor. The effects of baffle open area and pitch to equivalent diameter ratios on the heat transfer similarity factor are shown in Fig.(7). Generally, the heat transfer similarity factor increases with increasing the modified Reynolds number for all baffle configurations studied. Also, the heat transfer similarity factor for the perforated disc-baffles is lower than that of the solid disc-baffles, which indicates that the perforated disc-baffles provide higher heat transfer rates.

The efficiency index was employed to evaluate the performance benefits of the disc-baffled annulus at the different configurations and the results are shown in Fig.(8). The shape of the efficiency index curves reflects the different heat transfer and pressure drop mechanisms for different baffle configurations. Higher values of the efficiency index at lower Reynolds number were observed and then the value of the efficiency index decreases at higher values of Reynolds number. It can be seen that the perforated-baffled annulus has a higher thermal performance than the solid-baffled annulus for all baffles configurations studied. The efficiency index increases with increasing perforation density and baffle pitch to equivalent diameter ratio. The perforated baffles with open area ratio of 18% perform the highest efficiency indices under a constant power constraint.

The present experimental data for the Nusselt number enhancement ratios and the friction factor ratios were correlated in terms of Reynolds number, baffle pitch to annulus equivalent diameter ratio, and open area ratio and the following correlations were obtained:

$$\frac{Nu}{Nu_o} = 1.537\, Re^{0.068} \left(S/D_e\right)^{-0.346} (1 - B)^{-0.988} \tag{13}$$

$$\frac{F}{F_o} = 0.308\, Re^{0.364} \left(S/D_e\right)^{-0.744} (1 - B)^{1.845} \tag{14}$$

As shown in Fig.(9), the present correlations, Equations (13) and (14), are valid within ± 6% and ± 10% maximum deviation, respectively, for the present experimental data within the investigated range of Reynolds number from about 20,000 to 50,000, for three different values of baffle pitch to equivalent diameter ratios of 2, 4, and 8, and for open-area ratios ranging from 0 to 18%.

5. Conclusions

An experimental study is presented to provide a detailed investigation of the friction loss and heat transfer characteristics in an annulus with perforated disc-baffles aligned along the inner heated tube surface. The experimental runs are carried out for different values of Reynolds number, disc-baffle pitch to equivalent diameter ratio, and open area ratio. Conclusions emerging from the results of the present experimental study include the following:

1- For perforated baffles, a part of the flow passes through the holes in the baffle reducing or eliminating the hot zone and form drag. The jet like flow through the holes of the baffles and interacts with the reattaching and accelerating flow passing over the baffles causing a decrease in the thickness of the laminar sublayer and mixing of the fluid near the wall with the turbulent core. With the increase in the open area ratio, the reattachment effect reduces but the mixing at the wall increases because of a greater proportion of the flow through the perforations, which reflects an increase in heat transfer.

2- The Nusselt number enhancement ratio increases with increasing the baffle open area ratio and decreases with increasing the baffle pitch to equivalent diameter ratio.

3- For a given Reynolds number and baffle pitch ratio, the normalized friction factor decreases monotonically as the baffle open area increases from 0 to 18 % because of less cross-sectional blockage for baffles with larger open area ratio.

4- The perforated disc-baffles with open area ratio of 18% and $S/D_e = 2$ for the flow with Reynolds number of 48,024 perform the highest Nusselt number enhancement ratio, about 3 times more than that for the smooth annulus corresponding to 6.6 fold increase in the flow friction factor ratio.

5- The heat transfer similarity factor for the perforated disc-baffles is lower than that of the solid disc-baffles, which indicates that the perforated disc-baffles provide higher heat transfer rates.

6- The perforated-baffled annulus has a higher thermal performance than the solid-baffled annulus for all baffles configurations studied. The efficiency index increases with increasing perforation density and baffle pitch to equivalent diameter ratio. The perforated baffles with open area ratio of

18% perform the highest efficiency indices under a constant power constraint.

7- Empirical correlations are obtained for the influence of Reynolds number, baffle pitch to equivalent diameter ratio, and open area ratio on both the Nusselt number enhancement and the flow friction factor ratios.

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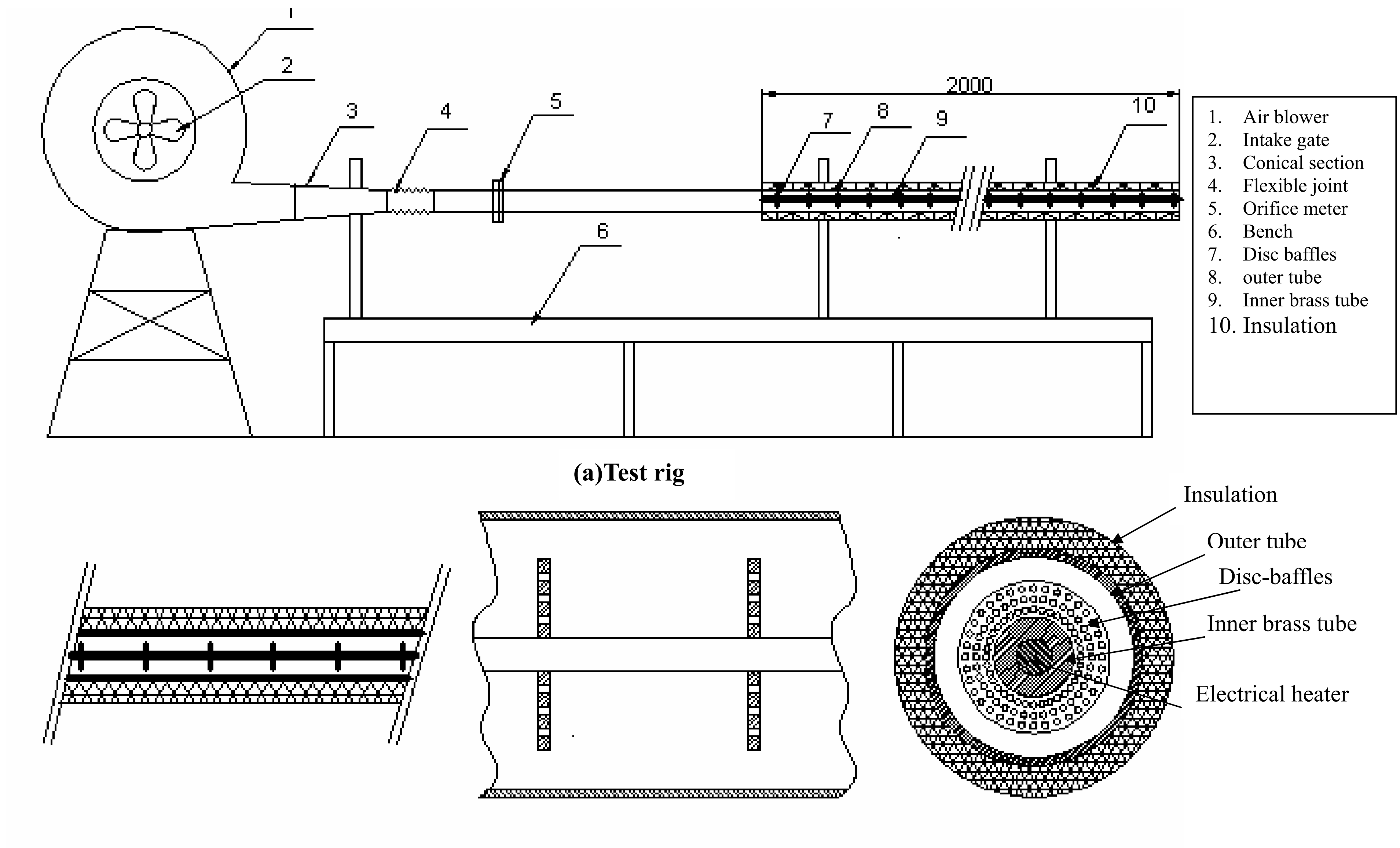
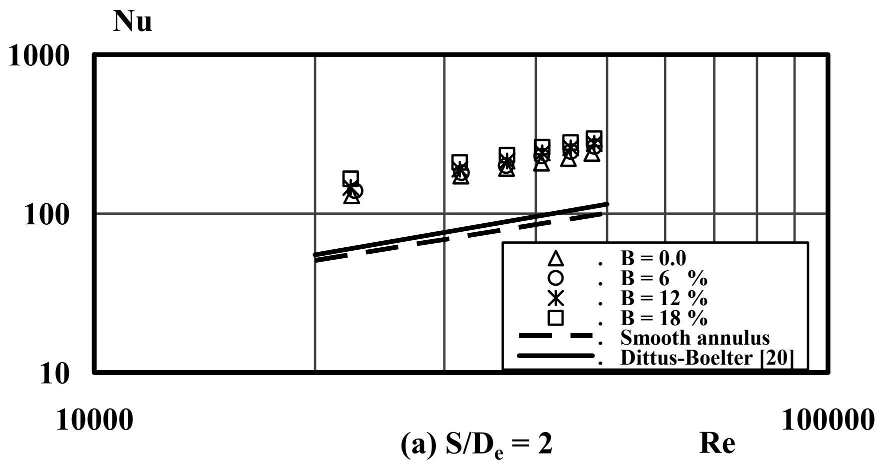


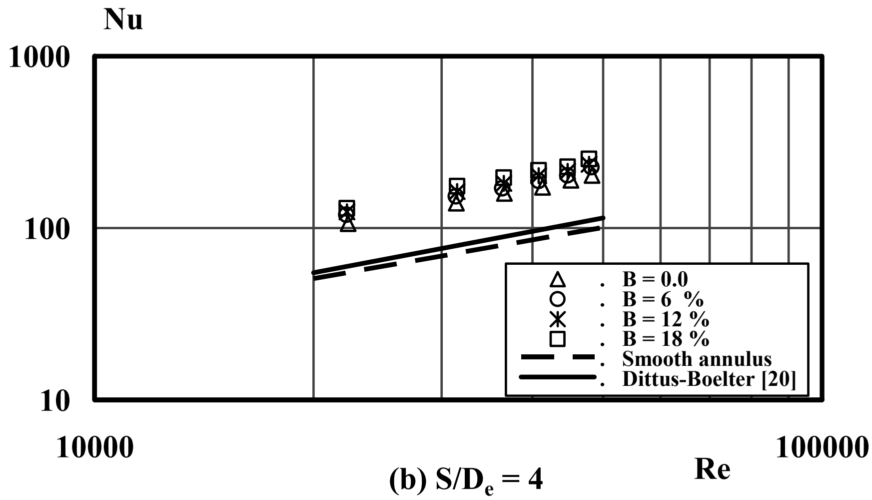
(a)Test rig

(b) Longitudinal -section in the baffled annulus **(c) Cross-section in the test section**

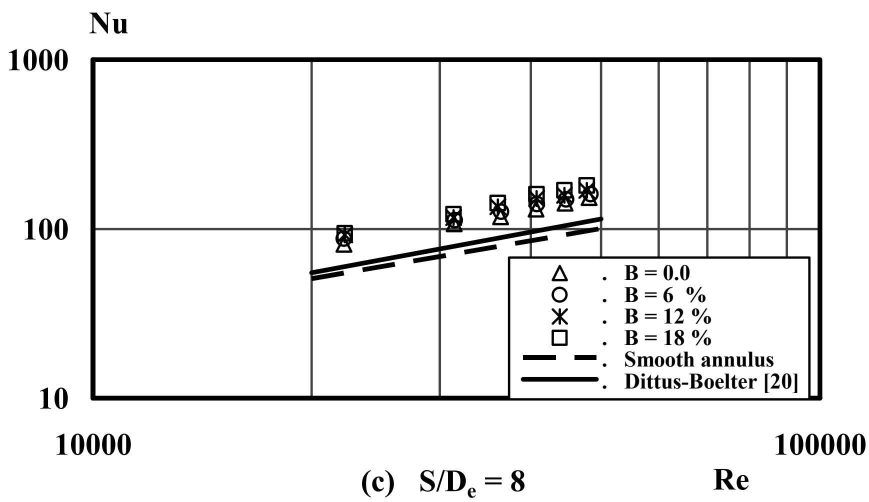
Fig.(1): Schematic diagram of the experimental set-up



(a) $S/D_e = 2$

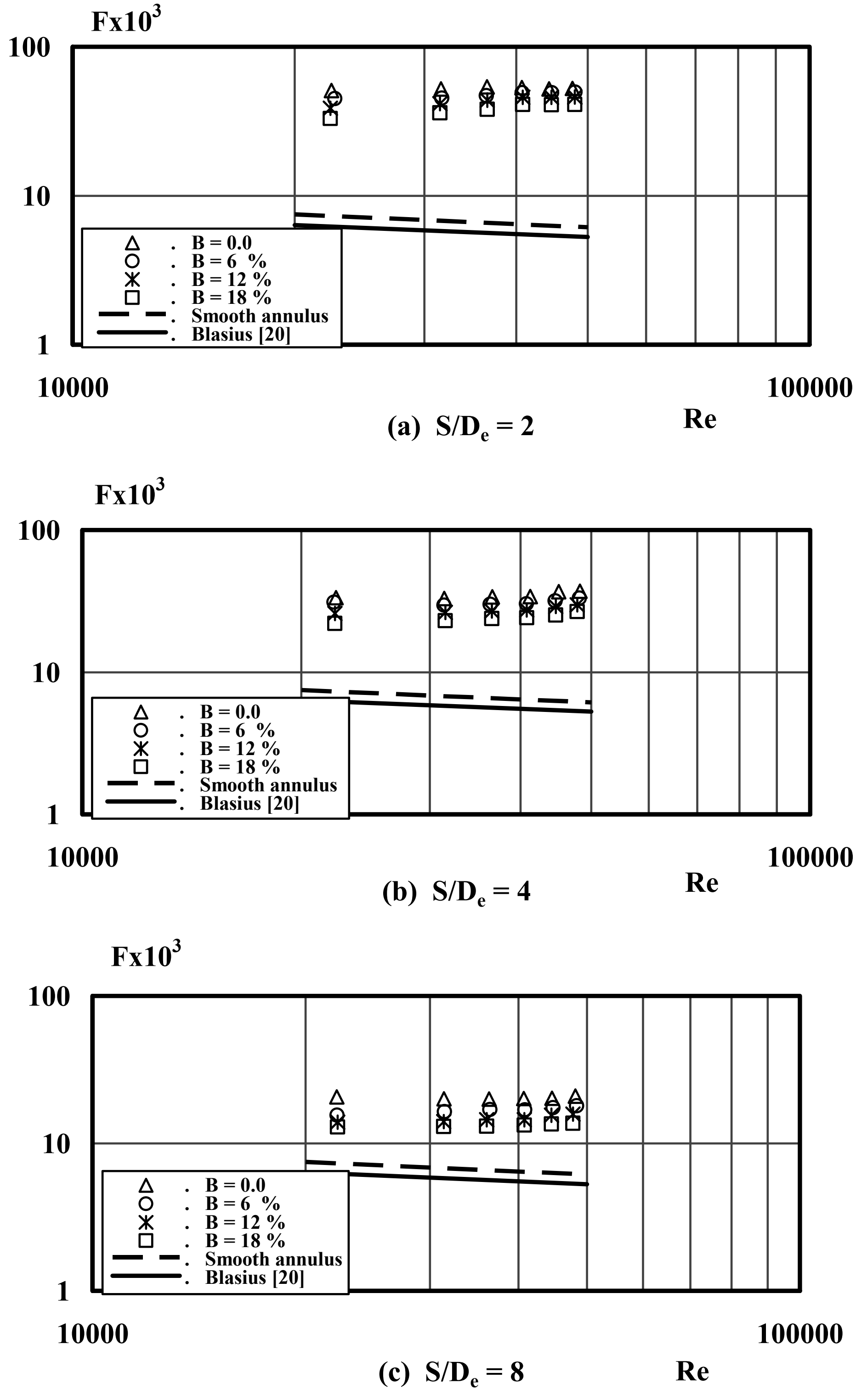


(b) $S/D_e = 4$



(c) $S/D_e = 8$

Fig.(2): The average Nusselt number versus Reynolds number for solid and perforated baffles at different S/D_e ratios.

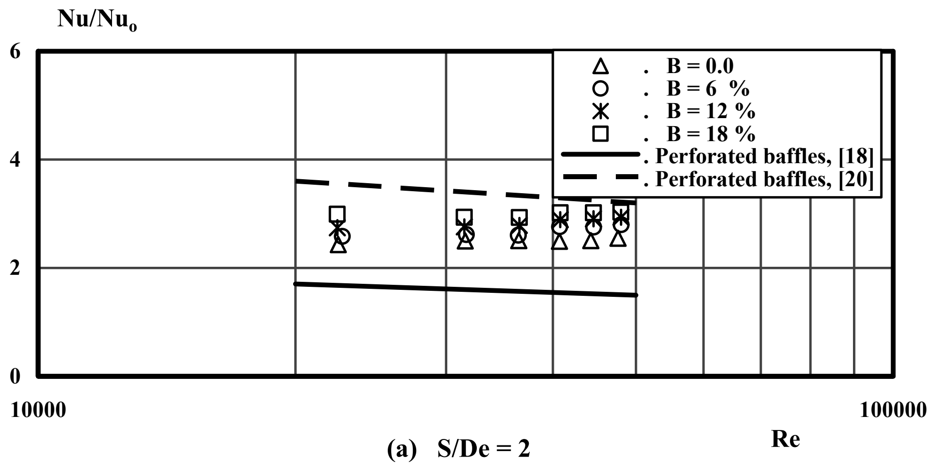


(a) $S/D_e = 2$

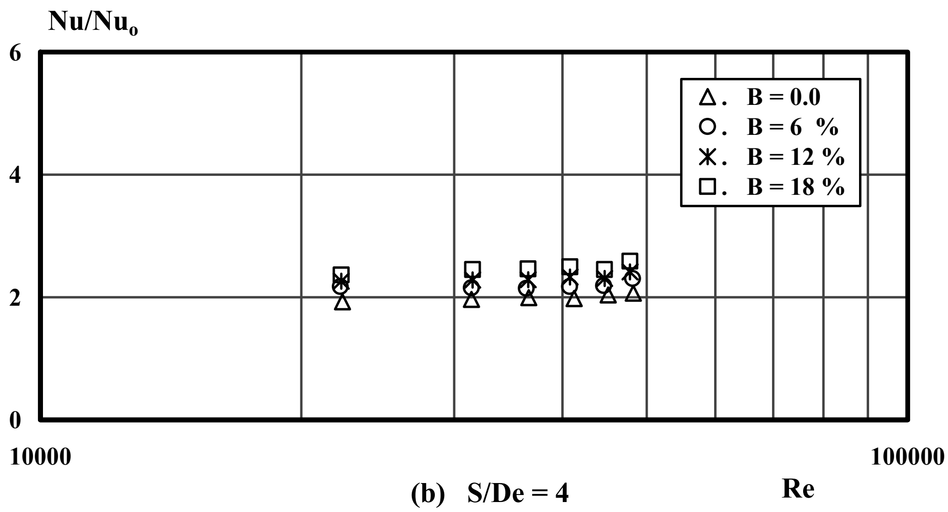
(b) $S/D_e = 4$

(c) $S/D_e = 8$

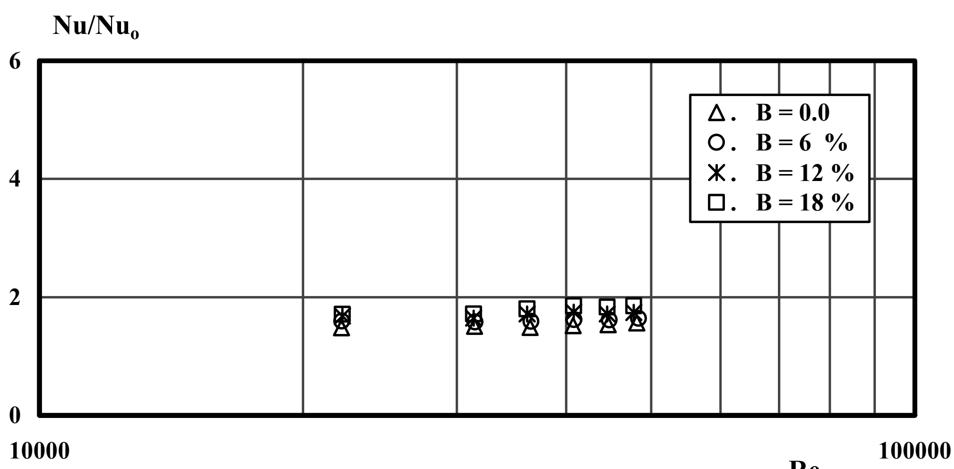
Fig.(3): Fanning Friction factor versus Reynolds number for solid and perforated baffles at different S/D_e ratios.



(a) S/De = 2



(b) S/De = 4



(c) S/De = 8

Fig.(4): The Nusselt number enhancement ratio versus Reynolds number for disc-baffled annulus at different S/D_e ratios.

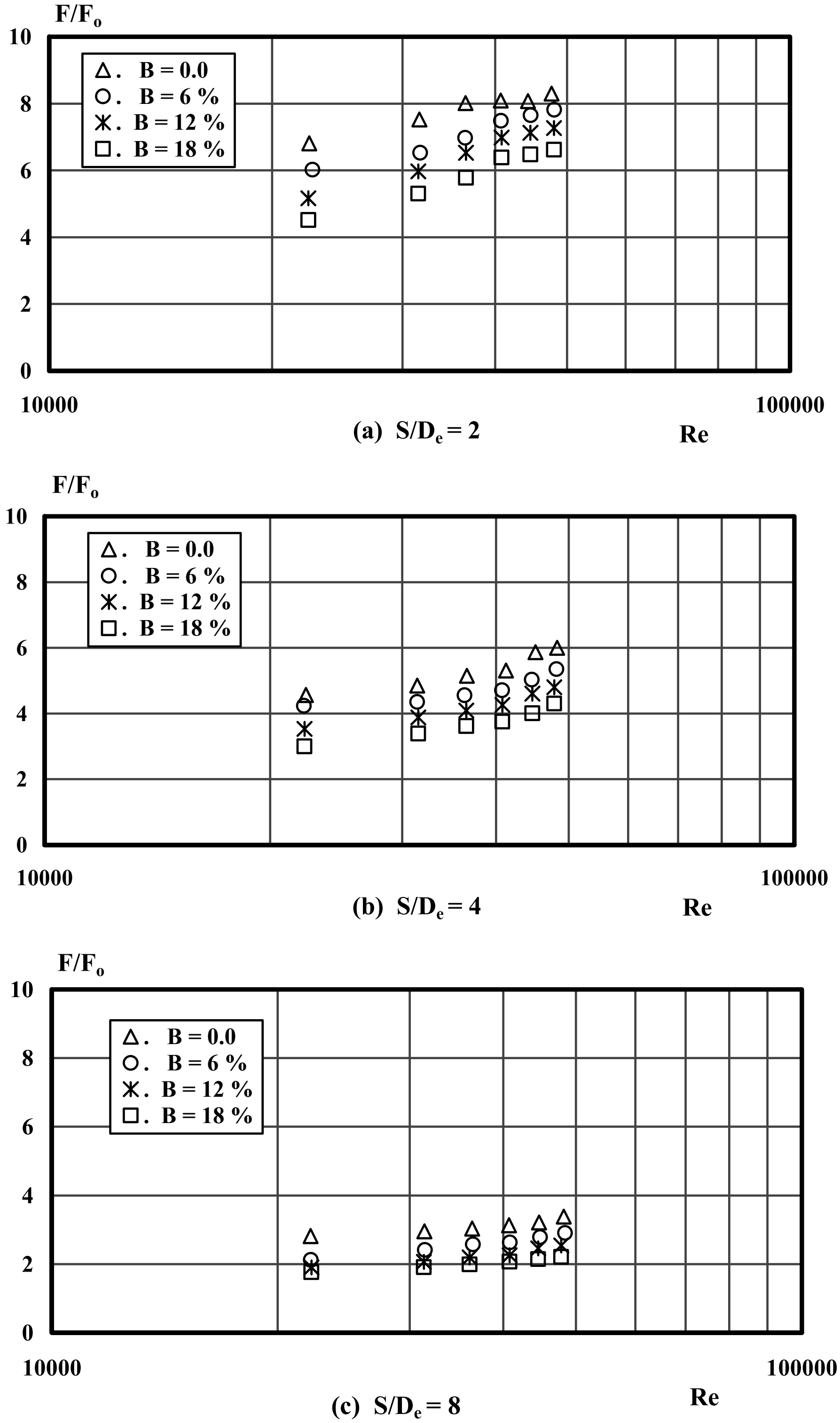


Fig.(5): The friction factor ratio versus Reynolds number for disc-baffled annulus at different S/D_e ratios.

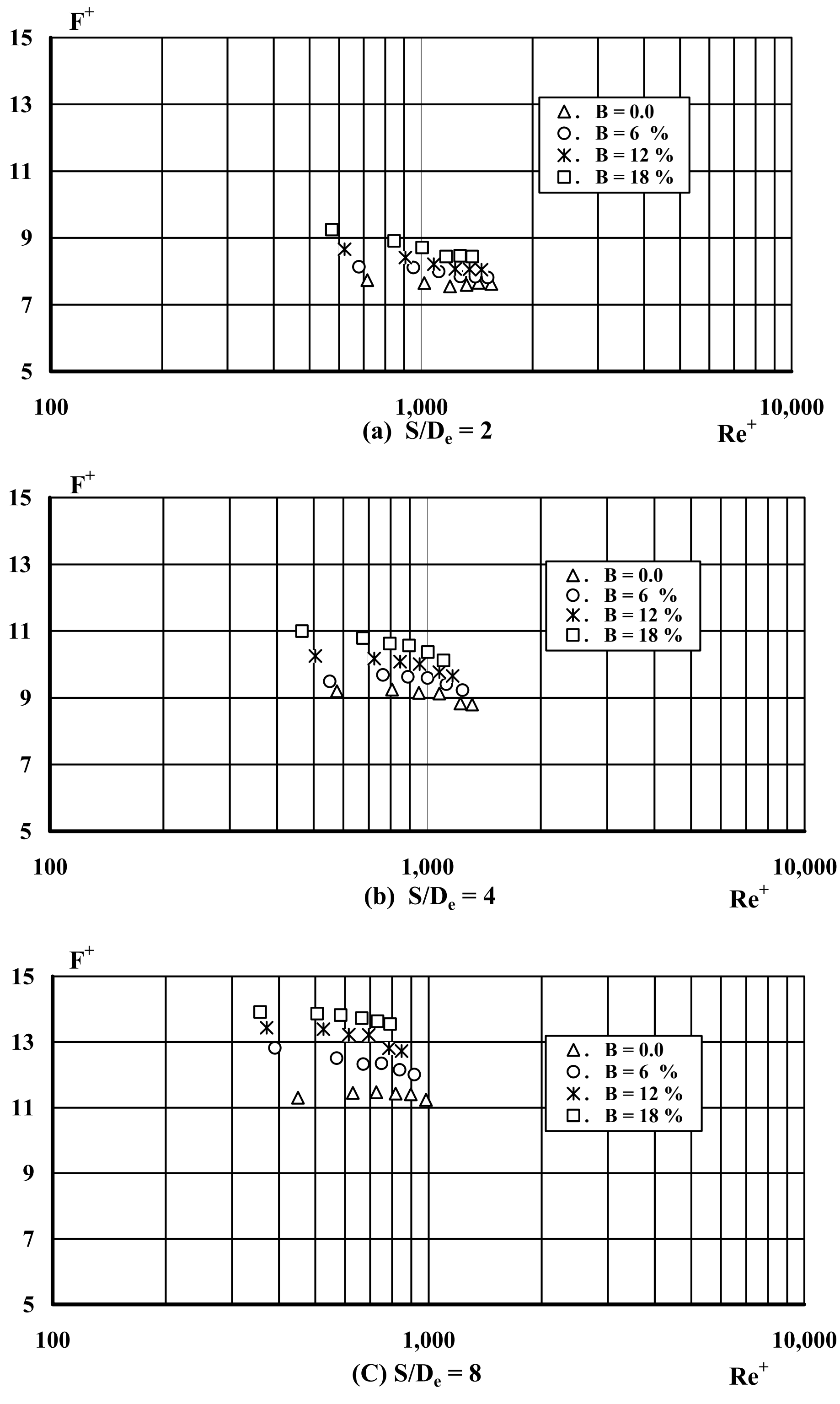
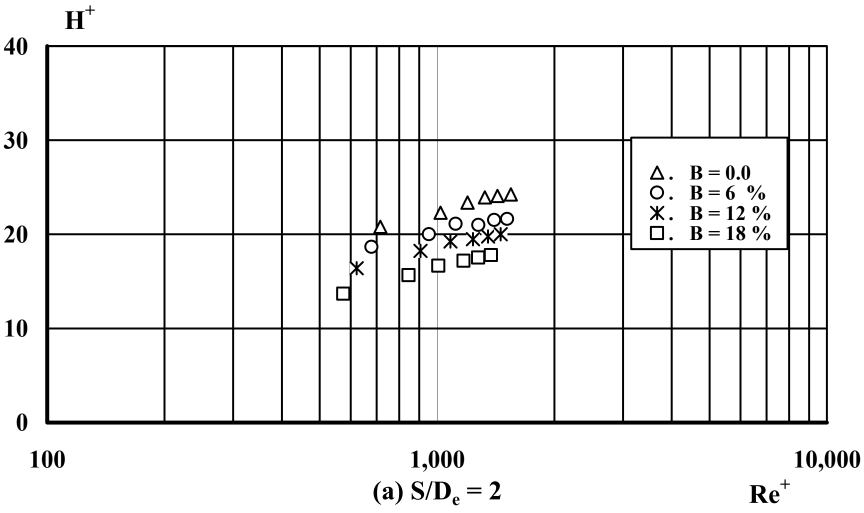
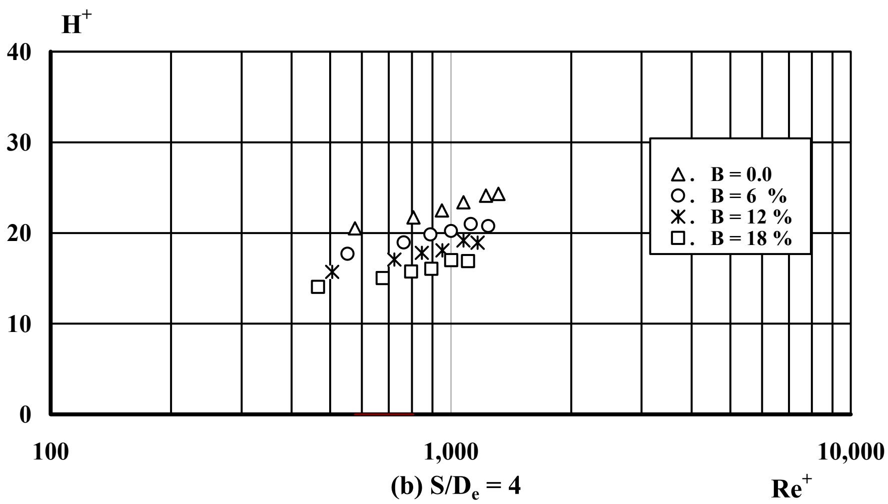


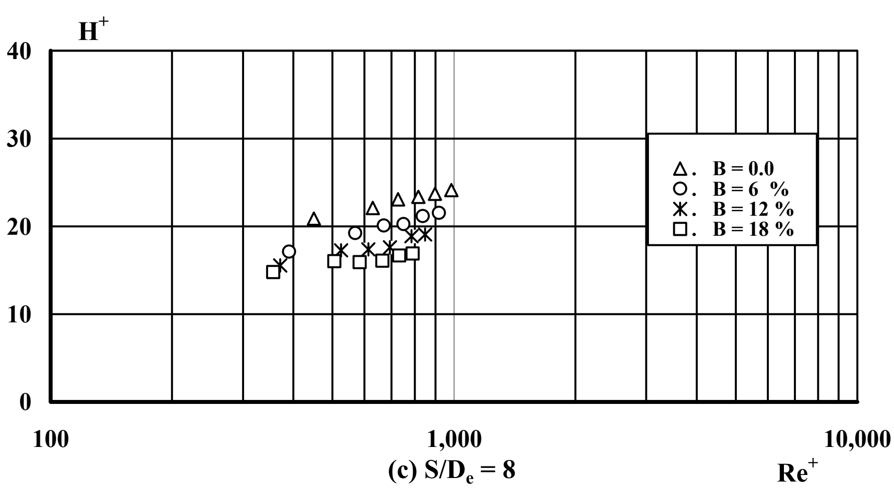
Fig.(6): Friction similarity factor versus modified Reynolds number at different baffle open-area and S/D_e ratios



(a) $S/D_e = 2$



(b) $S/D_e = 4$



(c) $S/D_e = 8$

Fig.(7): Heat transfer similarity factor versus modified Reynolds number at different baffle open-area and S/D_e ratios

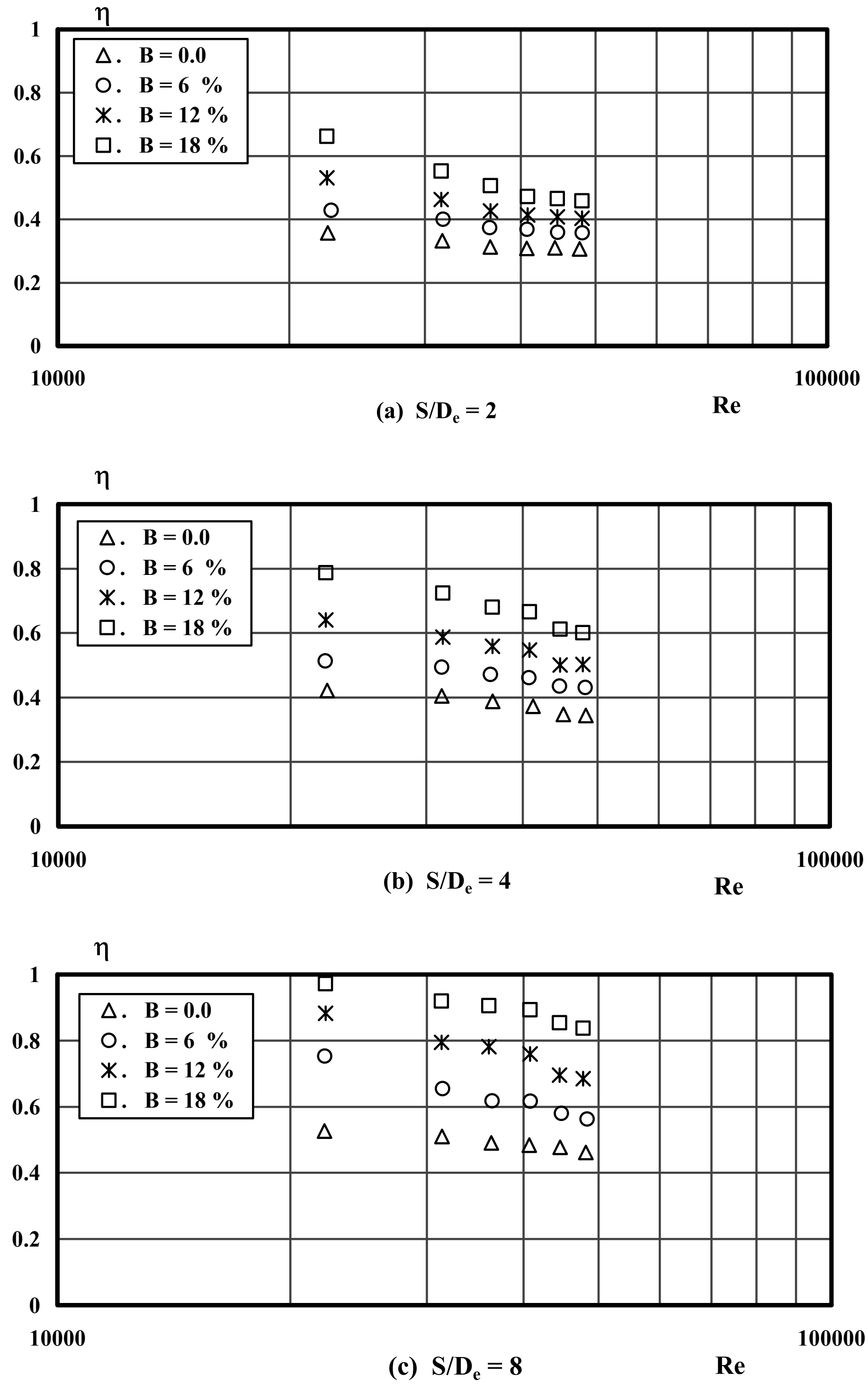
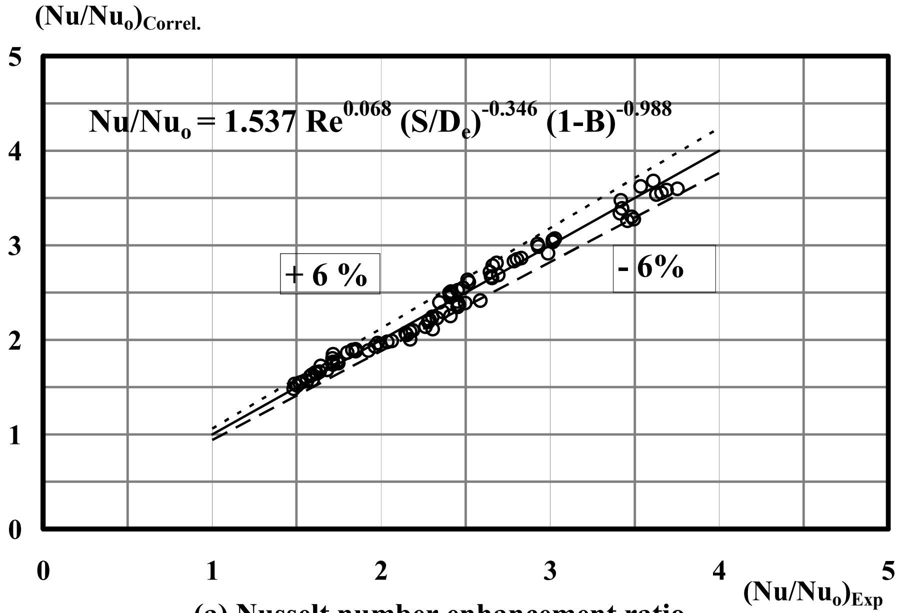
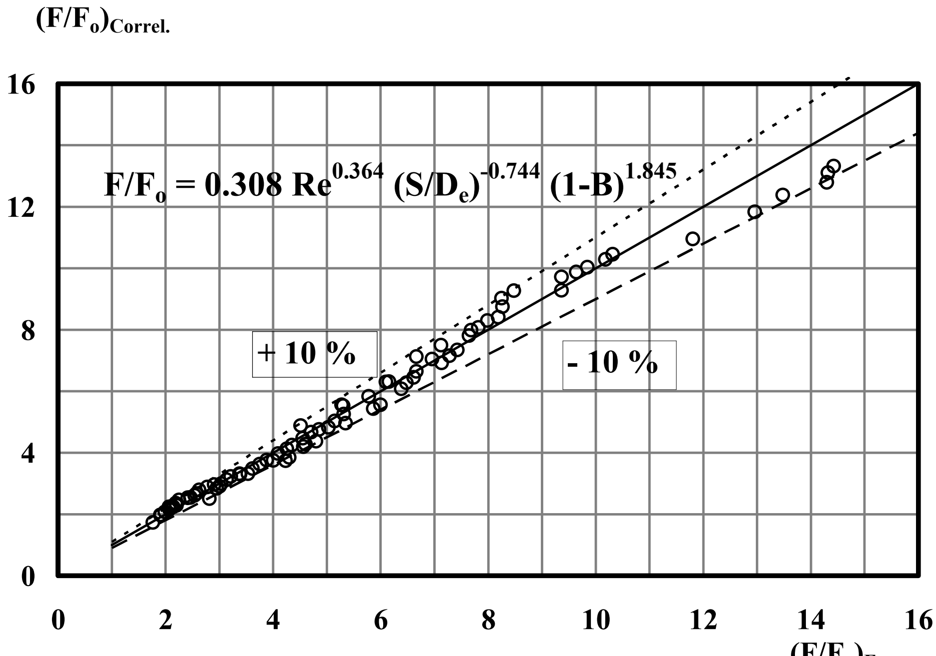


Fig.(8): Efficiency index versus Reynolds number for solid and perforated disc-baffled annulus at different S/D_e ratios.



(a) Nusselt number enhancement ratio



(b) Friction factor ratio

Fig.(9): Comparison of the present correlation with the present experimental data