

INVESTIGATION OF HEAT TRANSFER AND FRICTION CHARACTERISTICS IN A
CIRCULAR TUBE FITTED WITH CONICAL RINGS HAVING INTERNAL FINS

دراسة خصائص انتقال الحرارة والاحتكاك في انبوب دائرى مزود من الداخل بحلقات
مخروطية ذات زعانف داخلية

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ملخص البحث

البحث يقدم دراسة معملية لتأثير حلقات مخروطية مزودة بزعانف داخلية داخل انبوب دائرى على انتقال الحرارة والاحتكاك. فى هذا البحث الحلقات المخروطية ذات الزعانف الداخلية موضوعة داخل الانبوب الدائرى عند خمس نسب مختلفة لاقطار الحلقات المخروطية ($d/D = 0.5, 0.6, 0.7$) وعند كل نسبة يتم ترتيب الحلقات المخروطية عند ثلاثة اوضاع مختلفة وهى كالاتى:

1- Converging conical ring 2- Diverging conical ring 3- Converging-diverging conical ring
فى التجارب المعملية يتم تسخين الانبوب الدائرى من الخارج عن طريق سخان كهربى بينما استخدم الهواء البارد داخل الانبوب الدائرى عند عدد رينولدز مختلف يتراوح من 8000 الى 32464. واطهرت النتائج ان تأثير انعكاس تدفق المائع ونسبة الاقطار المختلفة وترتيب الحلقات المخروطية يحسن فى انتقال الحرارة. واتضح من التجارب ان استعمال الحلقات المخروطية ذات الزعانف الداخلية يودى الى زيادة انتقال الحرارة عن الانبوبة الملساء عند كل الاوضاع. واطهرت النتائج ايضا ان وضع الحلقات المخروطية فى حالة diverging conical ring تودى الى تحسين انتقال الحرارة عن الاوضاع الاخرى. ومن نتائج التجارب تم الحصول على معادلات معملية لانتقال الحرارة والاحتكاك كدالة فى نسبة الاقطار عند كل وضع من الحلقات المخروطية ذات الزعانف الداخلية.

Abstract:

The paper deals with an experimental study of the influence of conical rings with internal fins inserts on heat transfer and friction characteristics in a circular tube. In the present work, the conical rings with internal fins are placed in the test tube section with five different diameter ratios of the ring to tube diameter ($d/D = 0.5, 0.6, 0.7$) and for each ratio, the rings are placed with three different arrangements, converging conical ring, CR array, diverging conical ring, DR array and converging-diverging conical ring, CDR array. In the experiment, cold air at ambient condition is passed through the uniform heat flux circular tube for Reynolds

numbers in a range of 8000 to 32464. The results indicate that the effect of the reverse flows, ring to tube diameter ratio and the ring arrays can improve the heat transfer rate in the tube. Also, the experimental results demonstrate that the use of conical ring with internal fins inserts leads to a higher heat transfer rate than that of the plain surface tube, and the diverging conical ring array yields a better heat transfer than the others. The results are also correlated in the form of Nusselt number and friction factor as a function of Reynolds number, and diameter ratio.

Keywords: Enhancement of heat transfer, circular tube, conical ring.

Nomenclature

A : Heat transfer surface area, m^2
 C_p : Specific heat of air, J/kg K
d : Small end diameter of conical ring, m
D : Inside diameter of the test tube, m
f : Friction factor
h : Heat transfer coefficient, $W/m^2 K$
I : Current, A
k : Thermal conductivity of air, $W/m K$
L : Length of the test section, m
m : Mass flow rate, kg/s
Nu : Nusselt number
 ΔP : Pressure drop, Pa
P : Spacing between the rings, m
Q : Heat transfer rate, W
Re : Reynolds number
t : Temperature, $^{\circ}C$
 t_f : Mean temperature, $^{\circ}C$
U : Average axial velocity inside the test section, m/s
V : Voltage, V
 ν : Kinematic viscosity, m^2/s
 ρ : Density, kg/m^3
 η : The overall enhancement efficiency

a : Air
c : Convection
i : Inlet
o : Outlet
w : Wall
r : ring inserted tube
s : smooth tube

Introduction

A heat exchanger is a device facilitating the convective heat transfer of fluid inside the tubes and is extensively used in many engineering applications; In general, heat transfer augmentations can be divided into two groups. One is the passive method without stimulation by external power such as rough surfaces, extended surfaces and swirl flow devices (twisted tape or coiled wire, sometimes called "turbulators"). The other is the active method, which requires extra external power sources, for example, fluid vibration, fluid injection and suction and the use of electrostatic fields. A passive method of heat transfer enhancement has been of interest for many researchers because additional external power is not needed. Turbulators

Subscripts

are inserted into the flow to provide redevelopment of the boundary layer, to increase the heat transfer surface area and to cause enhancement of heat transfer by increasing the turbulence or by rapid fluid mixing. Therefore, a more compact and economic heat exchanger with lower operation cost can be obtained. The reverse flow devices or the turbulators are widely employed in thermal engineering applications. The reverse flow is sometimes called "re-circulation flow". The effect of reverse flow and boundary layer disruption (dissipation) is to enhance the heat transfer coefficient and momentum transfers. The reverse flow with high turbulence can improve convection of the tube wall by increasing the effective axial Reynolds number, decreasing the flow cross-section area, and increasing the mean velocity and temperature gradient. It can help to produce the higher heat fluxes and momentum transfer due to the large effective driving potential force but also higher pressure drop. The strength of reverse flow and the reattached position are the main interest in many heat transfer applications such as heat exchangers, combustion chambers, gas turbine blades, and electronic devices.

A heat transfer enhancement concept in which swirl was introduced in the flow was proposed by Kreith and Margolis [1]. Gambill and Bundy [2] claimed that twisted-tapes are also effective in high Prandtl number fluids because for such fluids it provides high heat transfer rate with less pressure drop compared with other inserts. Zozulya and Shkuratov. [3] reported that a smooth decrease in

pitch of a twisted-tape results in an improved heat transfer rate.

Van Rooyen and Kroeger [4] found that, for laminar swirl flow heat transfer in a smooth tube subjected to axially constant tube wall temperature, the heat transfer rate increases considerably for a moderate increase in pressure drop. Bergles [5] considered all these effects and developed laminar flow correlations for the friction factor and Nusselt number, including the swirl parameter, which defines the interaction between viscous, convective inertia and centrifugal forces. Saha and Dutta [6] observed that on the basis of constant pumping power and constant heat flux boundary condition short length twisted-tapes are found to perform better than full length twisted tapes for tighter twists.

Al-Fahed and Chakroun [7] found that there is an optimum tape width, depending on the twist ratio and Reynolds number, for the best thermo-hydraulic characteristics and the tight-fit tape yields a better performance over the loose-fit one. Suresh et al. [8] presented the thermo-hydraulic performance of twisted-tape inserts in a large hydraulic diameter annulus. Manglik and Bergles [9] investigated numerically the laminar convection heat transfer in circular-segment, uniform wall temperature ducts with a straight tape insert.

Junkhan et al. [10] reported the heat transfer with several types of turbulators located in a tube to provide energy savings. Promvongse and Eiamsa-ard [11] experimentally studied the effects of the conical turbulator combined with the snail entrance on heat transfer rate and flow friction in a heat exchanger.

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Eiamsa-ard and Promvonge [12] investigated the enhancement efficiency in a tube by using V-nozzle turbulators at different pitch ratios. Ayhan et al. [13] studied numerically and experimentally the heat transfer in a tube by means of truncated hollow cone inserts. Durmus [14] also examined the effect of cutting out conical turbulators, placed in a heat exchange tube, on the heat transfer rates with four different types of turbulators and four conical-angles (5° , 10° , 15° and 20°). Yakut et al. [15] experimentally investigated the effect of conical-ring turbulators on the turbulent heat transfer, pressure drop and flow induced vibrations. Their experimental data were analyzed and presented in terms of the thermal performances of the heat transfer promoters with respect to the heat-transfer enhancement efficiencies for a constant pumping power. Yakut and Sahin [16] reported the flow-induced vibration characteristics of conical-ring turbulators used for heat transfer enhancement in heat exchangers. They pointed out that the Nusselt number increased with the rising Reynolds number and the maximum heat transfer was obtained for the smallest pitch arrangement.

The effects of the conical ring turbulator inserts on the heat transfer rate and friction factor are experimentally investigated by Promvonge [17]. The experimental results demonstrate that the use of conical ring inserts leads to a higher heat transfer rate than that of the plain surface tube, and the diverging conical ring (DR) array yields a better heat transfer than the others. Heat transfer in flat tubes by means of helical screw-tape inserts studied by

Ibrahim [18]. The study shows that, the Nusslet number (Nu) and friction factor (f) decrease with the increase of spacer length or twist ratio for flat tube.

As shown from the above literature review, most of the conical-ring turbulators are in a truncated hollow cone shape with constant thickness throughout and diverging arrangements. None of them are in conical ring with internal fins geometry with cylindrical outer surface. So, the objective of this research is investigating the use of conical ring with internal fins turbulators with different ring to tube diameter ratios from both the heat transfer and friction factor points of view. The present work also investigates the effect of the conical ring with internal fins arrangements (DR, CDR and CR arrays) on the enhancement of heat transfer by comparing the results of different ring to tube diameter ratios ($d/D = 0.5, 0.6, 0.7$) with those of the plain tubes. All of the experiments are conducted at the same inlet conditions with the Reynolds number, based on the tube diameter, in a range of 8000 to 32464. In addition, the results of the heat transfer enhancement efficiency are also reported for performance evaluation of turbulators in this work.

Experimental apparatus:

The experiments were carried out in an open-loop experimental facility as shown in Fig.1a. The loop consisted of a 3 kW blower, orifice meter to measure the flow rate, and the heat transfer test section. The outer surface of the circular tube was covered with an electric insulating tape on which nickel-chrome wire of 0.4 mm was uniformly wound

to form the main heater as shown in Fig 1b. The main heater was covered with an asbestos layer of 55 mm thickness, on which another nickel-chrome wire of 0.4 mm was uniformly wound to form a guard heater. The Guard heater was covered with a 40 mm thick asbestos layer. Two pairs of thermocouples were installed in the asbestos layer between the main heater and the guard heater. The thermocouples of each pair were fixed on the same radial line. The input to the guard heater was adjusted so that, at the steady state, the readings of the thermocouples of each pair became practically the same. Then all the heat generated by the main heater is flowing inward

unit in conjunction with the thermocouples as can be seen in Fig.1b. Twenty thermocouples were taped on the local wall of the tube, and the thermocouples were placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The mean wall temperature was determined by means of calculations based on the reading of the copper-constantan thermocouples. In the experiments the conical ring arrays used in the present work are depicted in Fig.2a and the section of conical ring is shown in Fig.2b. Each conical ring was made of copper with 48mm in length and its small end diameters for three sizes were 24mm (0.5D), 28.8mm (0.6D), 33.6mm (0.7D), with 2mm uniform thickness throughout. In each test run, the conical rings were placed in three type arrays (1) diverging conical ring(DR), (2) converging

to the circular tube. The terminals of the wire were connected to a variac AC transformer, which was used to vary the voltage of the AC current passing through the heating coil. The test-section was circular tube and made of copper having a 51mm outer diameter, was 1250 mm long (L) and 1.5 mm thick (t) as depicted in Fig. 2.a and Fig.2.b shows the conical rings insert inside circular tube and conical ring with internal fins. The inner and outer temperatures of the bulk air were measured at certain points with a multi-channel temperature measurement

conical ring (CR) and (3) converging-diverging conical ring (CDR), respectively.

Uncertainty analysis:

The various characteristics of the flow, the Nusselt number, and the Reynolds numbers were based on the average of tube wall temperature and outlet air temperature. The local wall temperature, inlet and outlet air temperature, the pressure drops across the test section and airflow velocity was measured for heat transfer of the heated tube. The average Nusselt numbers were calculated and discussed where all fluid properties determined at the overall bulk mean temperature. In order to quantify the uncertainties, the reduced data obtained experimentally were determined. The uncertainty in the data calculation was based on Ref. [19]. The maximum uncertainties of

non-dimensional parameters are $\pm 3\%$ for Reynolds number, $\pm 14\%$ for Nusselt number and $\pm 15\%$ for friction. The uncertainty in the axial velocity measurement was estimated to be less than $\pm 6\%$, and pressure has a corresponding estimated uncertainty of $\pm 4\%$, whereas the uncertainty in temperature measurement at the tube wall was about $\pm 0.6\%$. The experimental results were reproducible within these uncertainty ranges.

Data reduction:

In the present work, air is used as the tested fluid and flowed through a uniform heat flux and insulated tube. The steady state of the heat transfer rate assumed to be equal to the heat loss from the test section, which can express as:

$$Q_a = Q_c \quad (1)$$

Where:

$$Q_a = mC_{pa}(t_o - t_i) = VI \quad (2)$$

The heat supplied by the electrical winding in the test tube is found to be 5–8% higher than the heat absorbed by the fluid for the thermal equilibrium test due to convection and radiation heat losses from the test section to the surroundings. Thus, only the heat transfer rate absorbed by the fluid taken for the internal convective heat transfer coefficient.

The rate of heat transfer coefficient from the test section calculated as:

$$h_c = Q_a / A(t_w - t_a) \quad (3)$$

Where:

$$t_a = (t_o + t_i) / 2 \quad (4)$$

$$t_w = \sum t_w / 20 \quad (5)$$

In which t_w is the local surface temperature at the outer wall of the inner tube. The average surface temperature t_w calculated from 20 points of t_w lined between the inlet and the exit of the test tube. The average Nusselt number, Nu estimated as follows:

$$Nu = h_c d_i / k \quad (6)$$

The Reynolds number of the airflow inside the tube is given

$$Re = U d_i / \nu \quad (7)$$

The friction factor, can be written as

$$f = \Delta P / [(L/d_i) (\rho U^2 / 2)] \quad (8)$$

Where, U is the mean air velocity in the tube. All of the thermo physical properties of the air are determined at the overall bulk air temperature from Eq. (4).

The overall enhancement efficiency (η) which had written as follows:

$$\eta = (Nu_r / Nu_s) \cdot (f_s / f_r)^{1/3} \quad (9)$$

Results and discussions:

Figs. 3 and 4 show comparison between the present experimental work and correlations from the literature. The present plain tube data is found to be in good agreement with the previous correlations of Dittus-Boelter, Kay and Petukhov [20] for both the Nusselt number and the friction factor. In the figures, results of the present work agree well with the available correlations within $\pm 12\%$ in comparison with Nusselt number correlations and within $\pm 9\%$ in comparison with friction factor correlation. The effect of using the conical ring with internal fins arrays (DR/CDR/DR) and the diameter ratios ($d/D = 0.5, 0.6, \text{ and } 0.7$) on the heat transfer coefficient across the test tube are presented in Figs. 5, 6 and 7 respectively. The variation of the heat transfer coefficient is shown in terms of Nusslet number with Reynolds number. In the figures, it is apparent that the presence of the use of conical ring with internal fins inserts leads to considerably higher heat transfer rates than the plain tube for all arrangements. The mean increases in Nusslet number of using the conical ring with internal fins arrays (DR/CDR/CR type) inserts with diameter ratio, $d/D=0.5$ are about 6 to 8; 5 to 6 and 3.5 to 5 times that of the smooth tube, respectively. The increase in heat transfer rates with the reverse turbulent flow is higher

than that with the axial flow (smooth tube) and is also much larger than that in Nusselt number. On the other hand, the boundary layer disruption causes a better chaotic mixing between the core and the wall regions. This indicates that the effects of reverse flow and boundary layer disruption can help to enhance the convection heat transfer and momentum processes. In addition, the use of the DR array provides better heat transfer than that of the CDR or CR.

The variation of friction factor with Reynolds number for the conical ring with internal fins arrays (DR/CDR/CR) and the diameter ratios ($d/D = 0.5, 0.6, \text{ and } 0.7$) are shown in Figs.8, 9 and 10. The friction factor decreases with increasing Reynolds number and diameter ratios for all conical ring arrays. Conical ring arrays (DR/CDR/CR) a considerable increase in friction factor is observed in comparison with the smooth tube at all diameter ratios. This is because the reverse flow causes the dissipation of dynamic pressure of the fluid due to high viscous losses near the pipe wall and to the forces exerted by the ring blockage. Moreover, the friction increase is probably due to the secondary flows occurring as a result of the interaction of pressure forces with inertial forces in the boundary layer. The mean increases in friction factor of using the Conical ring with internal

fins arrays (DR/CDR/CR type) inserts with diameter ratio, $d/D=0.5$ are about 144 to 168, 95 to 112 and 68 to 73 times that of the smooth tube, respectively. For the DR array, it is observed that the friction factor, having a peak value at lower diameter ratio, shows a rapid reduction as the Reynolds number increases. As expected, the DR arrangement yields a higher pressure loss across the test tube than those of the CDR and CR arrays at similar diameter ratios.

The variation of overall enhancement efficiency (η) is plotted against Reynolds number for all cases in Figs. 11, 12 and 13. Heat transfer enhancement is provided with the expense of rising friction losses caused by conical ring with internal fins. So, a performance analysis is essential for the evaluation of the net energy gain to make a decision if the technique applied to increase the heat transfer is efficient from energy point of view. The comparison is made based on the same pumping power in order to determine the net final gain. There will be a net energy gain only if η is greater than unity. In the figures, it is worth noting that the enhancement efficiency reduces with the increase of Reynolds numbers and the decrease of diameter ratio. Also, it is worth noting that the enhancement efficiency of the DR array is found to be the highest compared to those of

the CDR and CR arrays for all similar diameter ratios.

Correlation of the results

The general correlation of the Nu as a function of Re and d/D of the experimental results are expressed as the following:

$$Nu = a Re^b (d/D)^c \quad (10)$$

The experimental data is fitted to get the constants and the following correlation can be obtained:

For diverging conical ring (DR):

$$Nu = 0.34 Re^{0.603} (d/D)^{-1.142} \quad (11)$$

For converging-diverging conical ring (CDR):

$$Nu = 0.24 Re^{0.61} (d/D)^{-1.199} \quad (12)$$

For converging conical ring (CR):

$$Nu = 0.29 Re^{0.55} (d/D)^{-1.461} \quad (13)$$

The general correlation of the f as a function of Re and P/L of the experimental results is expressed as the following:

$$f = c Re^m (d/D)^n \quad (14)$$

The experimental data is fitted to get the constants and the following correlation can be obtained:

For diverging conical ring (DR):

$$f = 17.52 Re^{-0.368} (d/D)^{-3.174} \quad (15)$$

For converging diverging conical ring (CDR):

$$F = 7.133 \text{ Re}^{-0.27} (d/D)^{-2.518} \quad (16)$$

For converging conical ring (CR):

$$F = 6.479 \text{ Re}^{-0.367} (d/D)^{-3.412} \quad (17)$$

$$32464 \leq \text{Re} \leq 8000 \text{ and } 0.5 \leq d/D \leq 0.7$$

The calculated Nusselt number (Nu_{cal}) and (f_{cal}) from Eq. (12) through (17) is plotted versus experimental Nusselt number (Nu_{exp}) in Fig. 14 to 16 and (f_{exp}) in Fig.17 to 19. In these figures, the results of the present work agree well with the fitted correlations within $\pm 5\%$ to 10% in comparison with experimental data for the Nusselt number, and within $\pm 7\%$ to 10% for the friction factor

Comparison with the previous work:

Comparisons between the overall enhancement efficiency obtained from the present measurement with reference [17] are portrayed in Figs. 20, 21 and 22, respectively. In these figures, the overall enhancement efficiency for circular tube fitted conical rings with inserted internal fins increases than a circular tube fitted conical rings without fins inserted inside conical rings at $d/D = 0.5$.

Concluding remarks

Experimental investigations have been conducted to examine the effect of conical rings with internal fins turbulators on heat transfer rate and flow friction characteristics in a uniform heat flux tube using air as a test

fluid. The conical rings with internal fins were placed in three type arrays, diverging conical ring (DR), converging conical ring (CR) and converging– diverging conical ring (CDR) at d/D are 0.5, 0.6 and 0.7. It is found that the insertion of conical rings with internal fins turbulators has significant effect on the enhancement of heat transfer which the turbulator is used as a reverse/turbulence flow generator. The variations of the enhancement efficiency for Reynolds number ranging from 8000 to 32464. This study indicates that the maximum improvements of heat transfer rate over the corresponding plain tube are found to be about between 497% and 668%, 403% and 478%, 296% and 404% for DR, CDR, and CR at $d/D = 0.5$, respectively. In addition the heat transfer rate in the test tube can be promoted by fitting with conical rings with internal fins-turbulators. Despite very high friction, the turbulators can be applied effectively in places where pumping power is not significantly taken into account but the compact size including ease of manufacture installation is required. Correlations for the Nusselt number and the friction factor based on the present experimental data are introduced for practical use.

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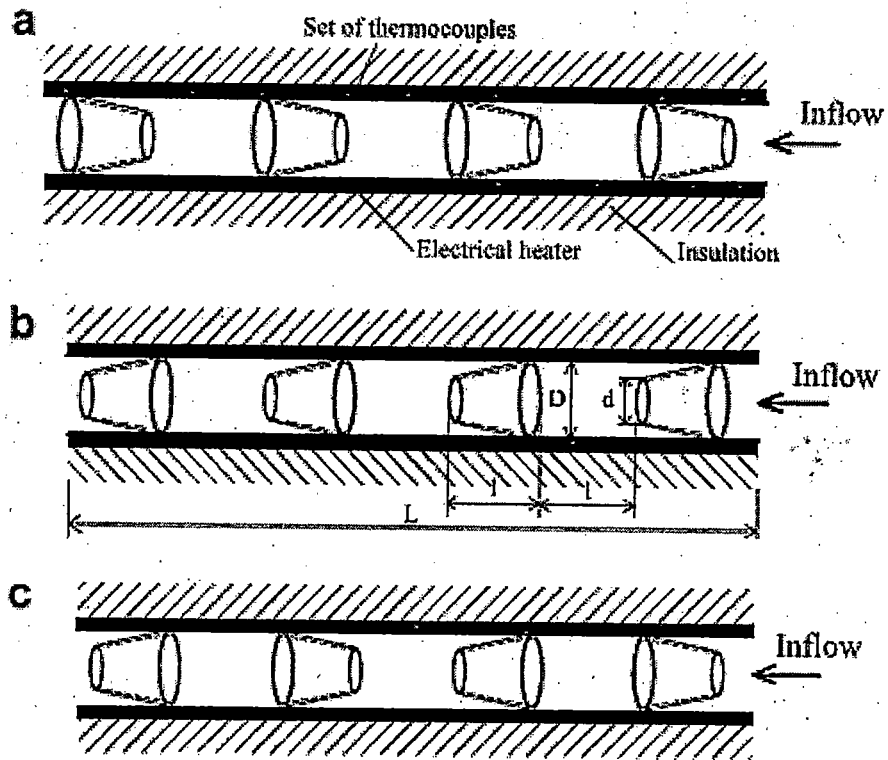


Fig. 2.a Test tube fitted with varies conical rings arrays,
 (a) DR array, (b) CR array and (c) CDR array.

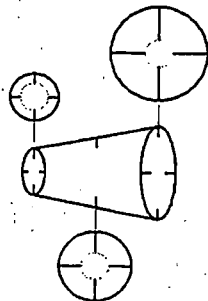


Fig.2.b shows the section of conical ring with internal fins

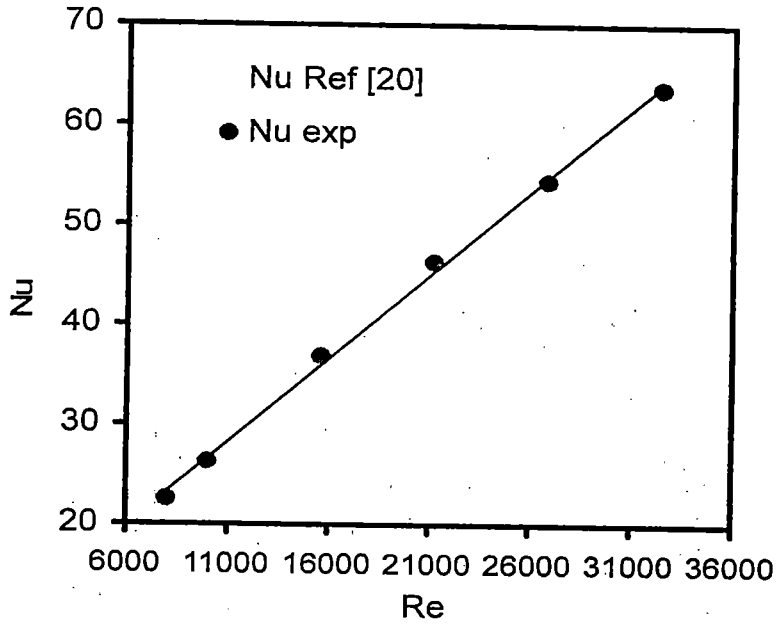


Fig. 3. Verification of Nusselt number of plain tube.

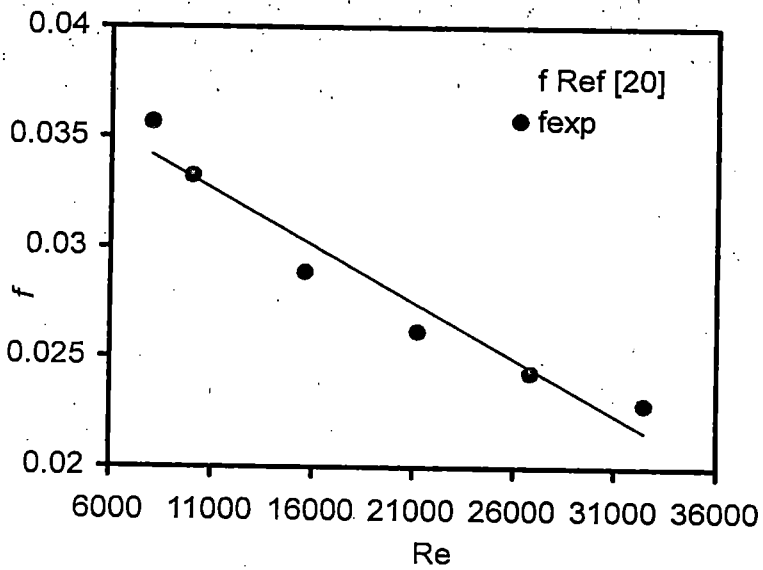


Fig. 4. Verification of friction factor of plain tube.

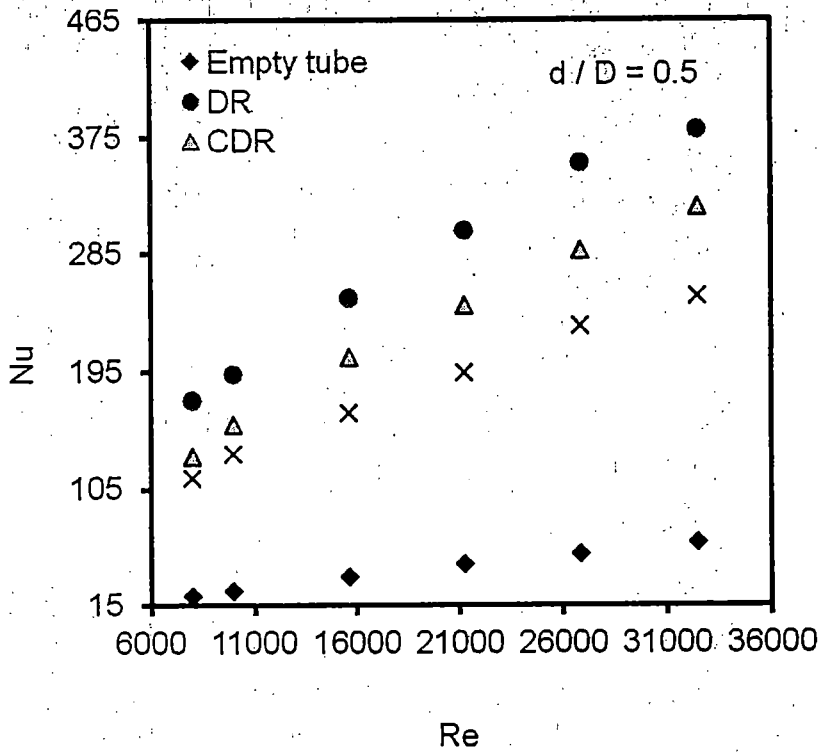


Fig. 5. Variation of Nusselt number with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.5$.

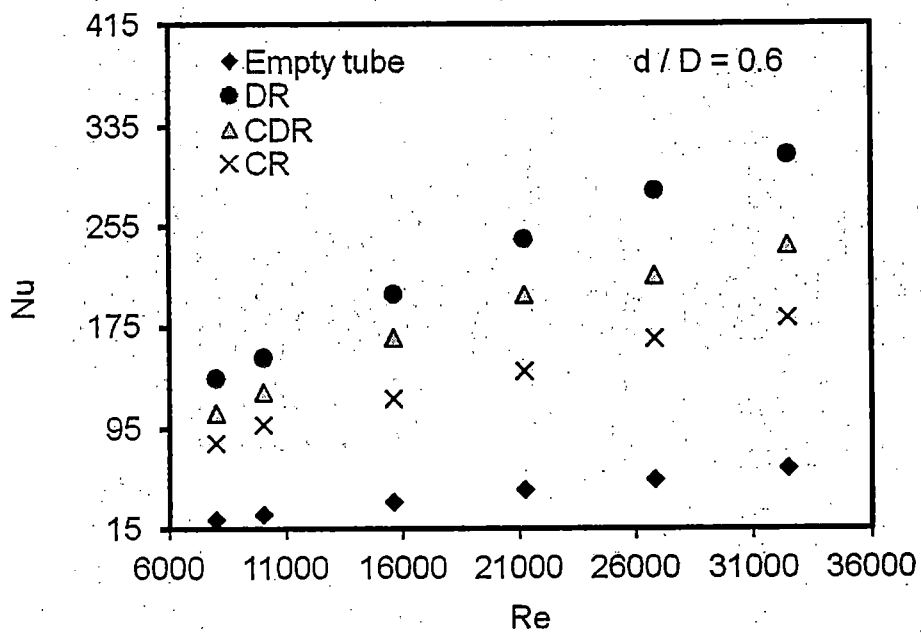


Fig. 6. Variation of Nusselt number with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.6$.

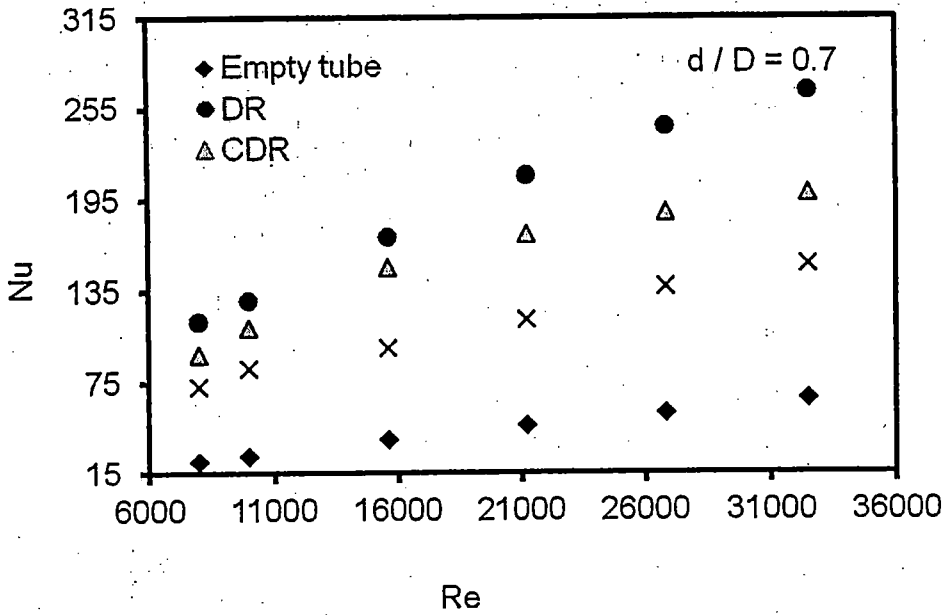


Fig. 7. Variation of Nusselt number with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.7$.

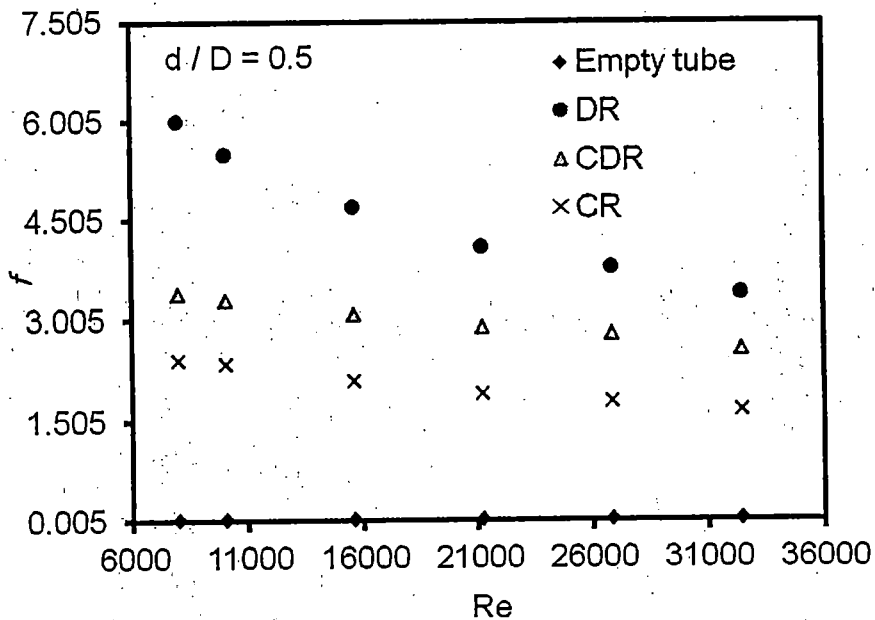


Fig. 8. Variation of friction factor with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.5$.

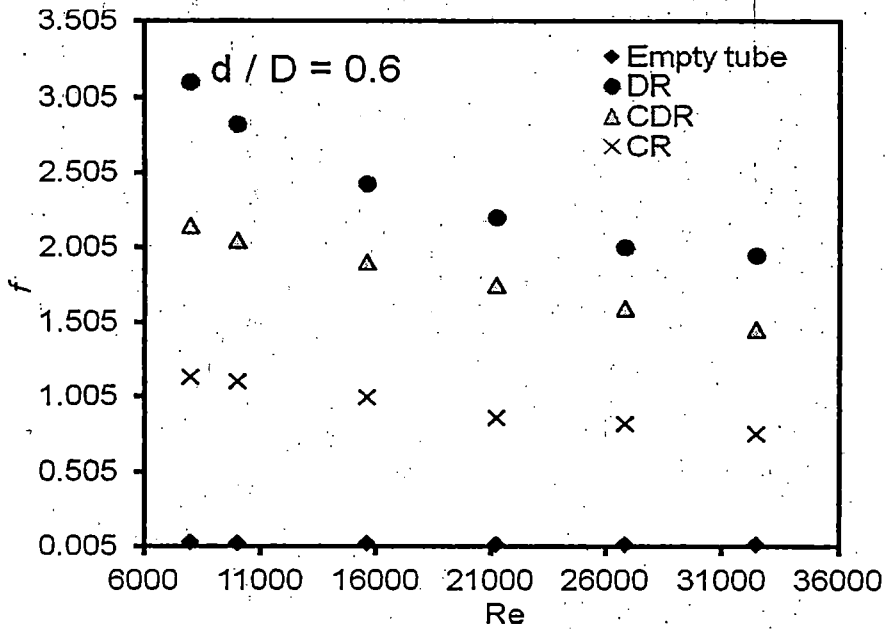


Fig. 9. Variation of friction factor with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.6$.

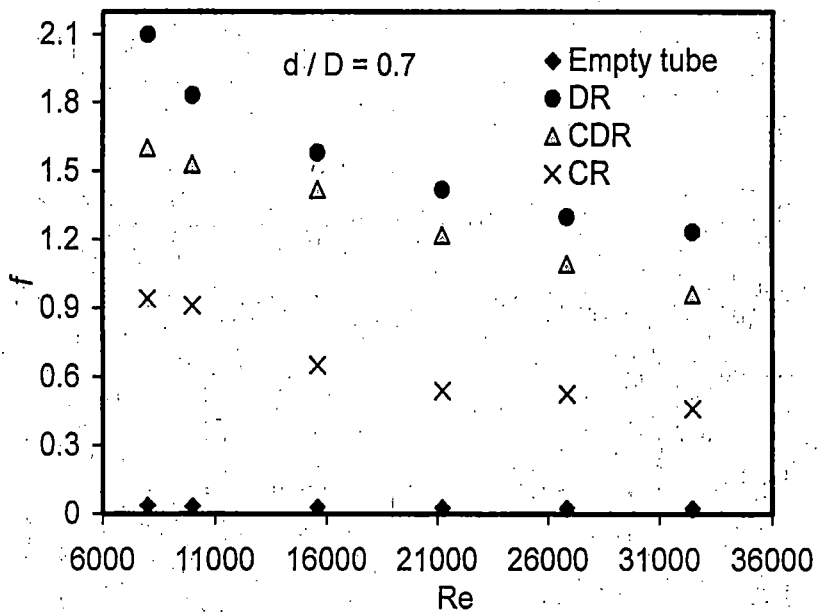


Fig. 10. Variation of friction factor with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.7$.

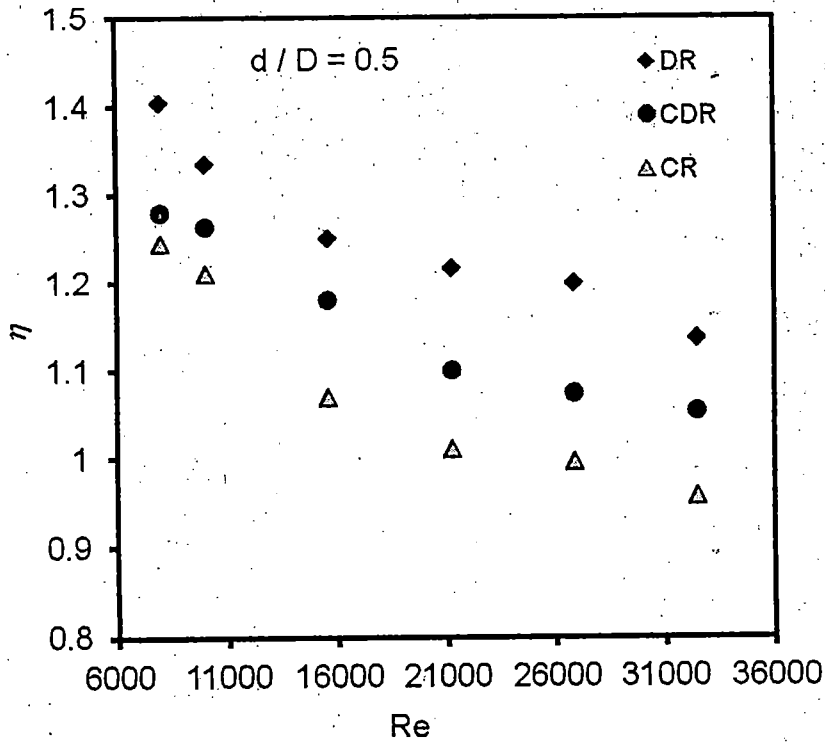


Fig. 11. Variation of the overall enhancement efficiency with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.5$.

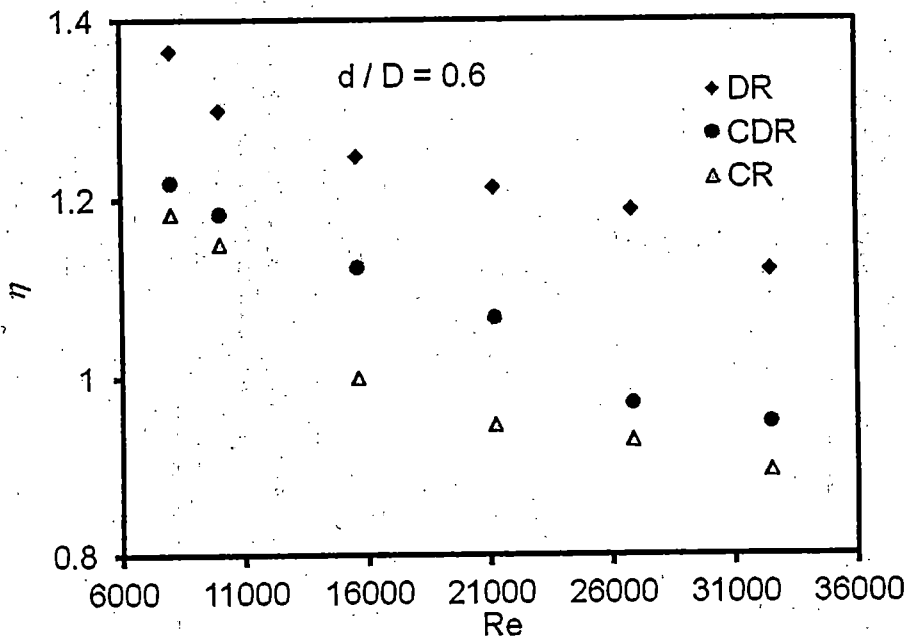


Fig. 12. Variation of the overall enhancement efficiency with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.6$.

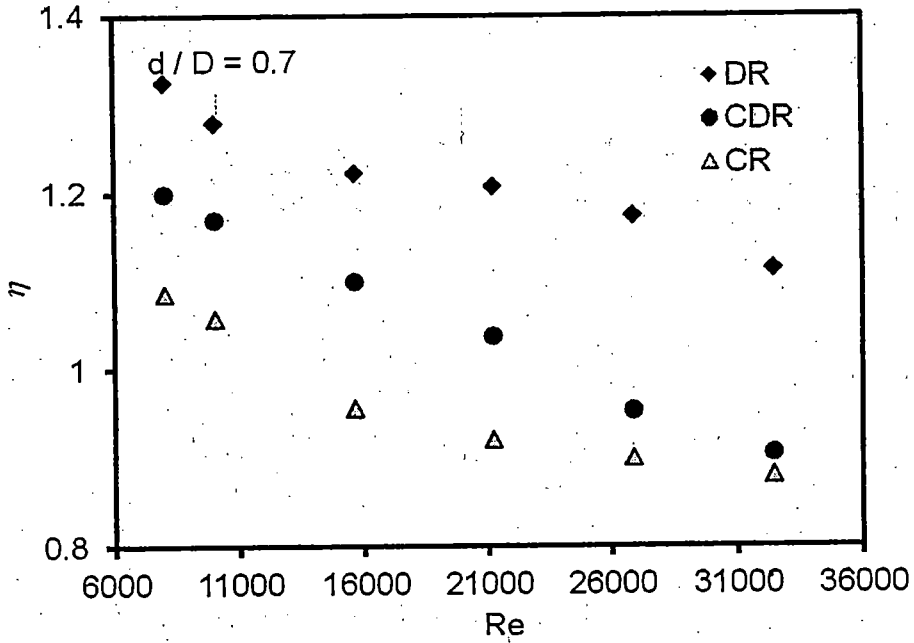


Fig. 13. Variation of the overall enhancement efficiency with Reynolds number for various conical ring arrays (CR/DR/CDR) at the diameter ratio $d/D = 0.7$.

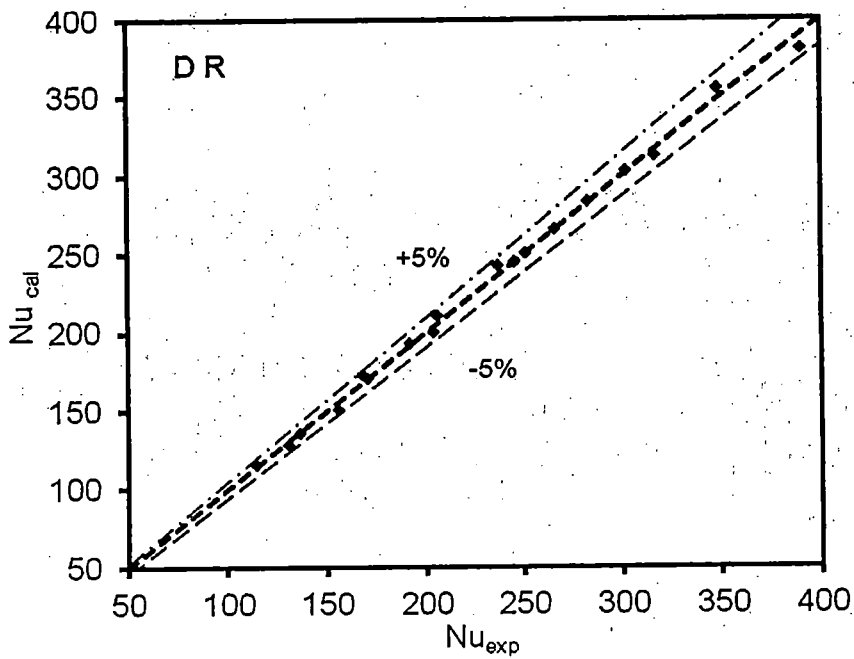


Fig. 14 Nusselt numbers obtained by the present correlation and experimental data for Diverging conical ring (DR)

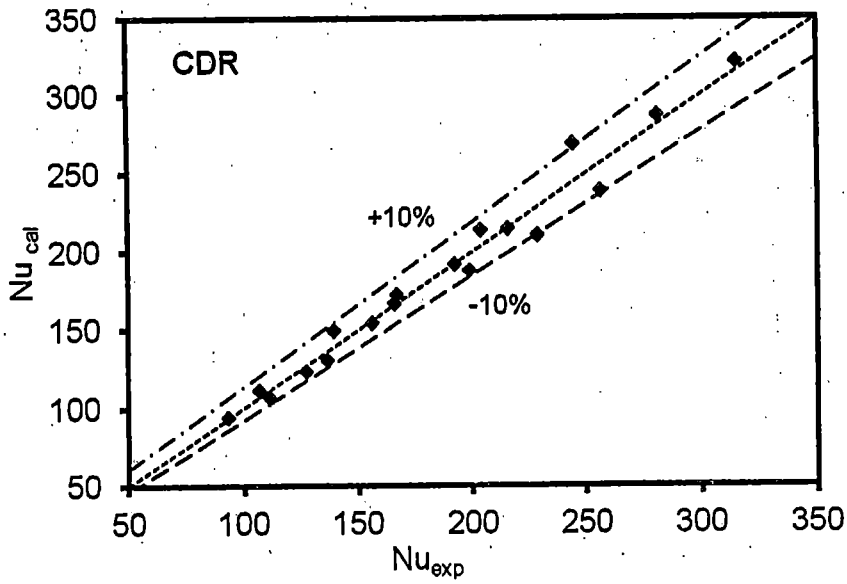


Fig. 15. Nusselt numbers obtained by the present correlation and experimental data for Converging-diverging conical ring (CDR)

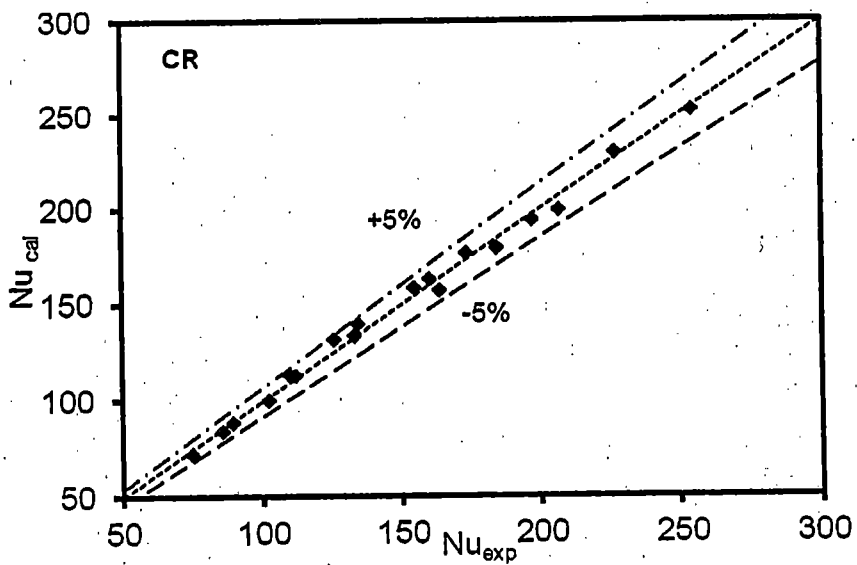


Fig. 16 Nusselt numbers obtained by the present correlation and experimental data for Converging conical ring (CR).

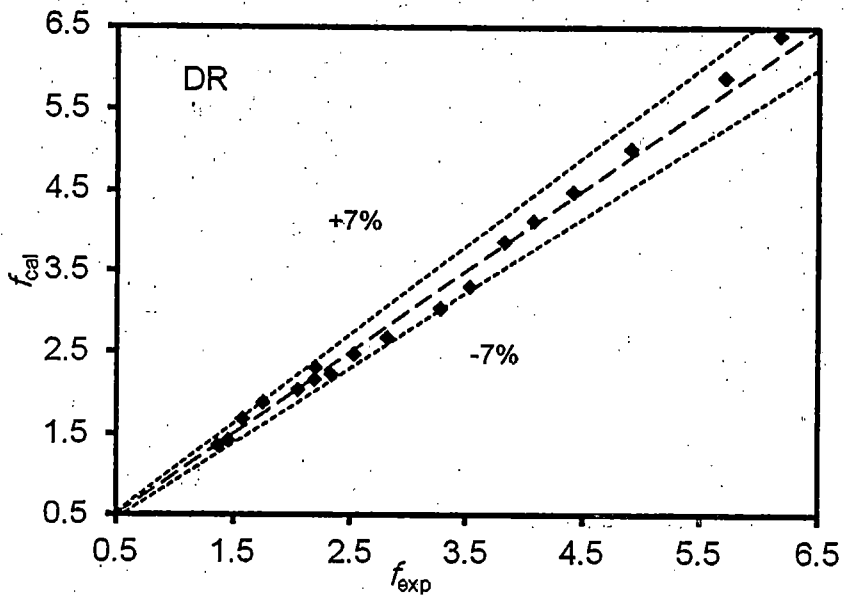


Fig. 17. Friction losses obtained by the present correlation and experimental data for Converging conical ring (DR).

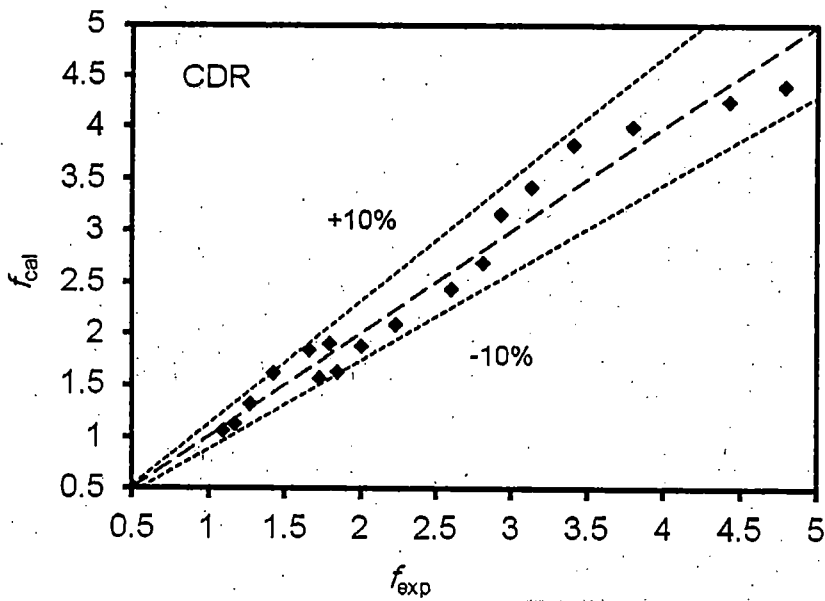


Fig. 18. Nusselt numbers obtained by the present correlation and experimental data for Converging-diverging conical ring (CDR)

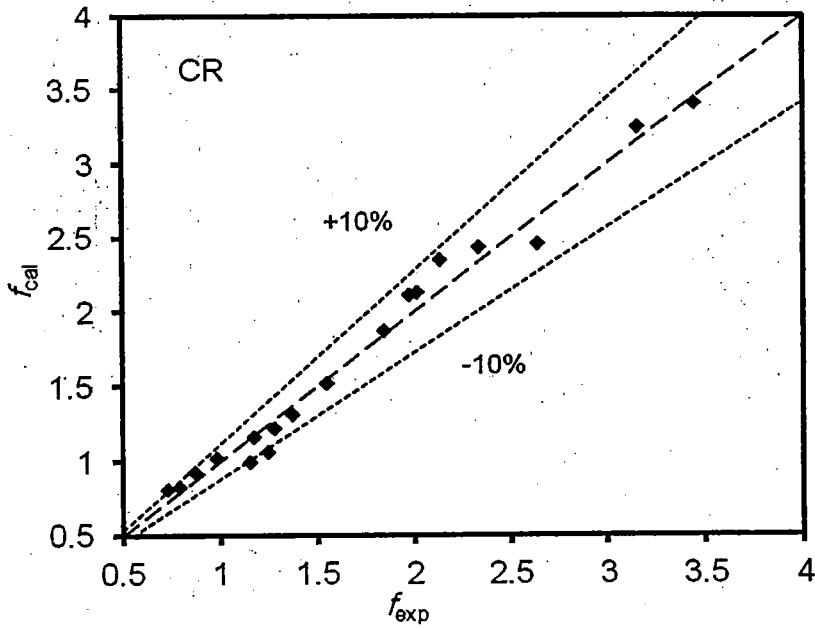


Fig. 19. Friction losses obtained by the present correlation and experimental data for Converging conical ring (CR).

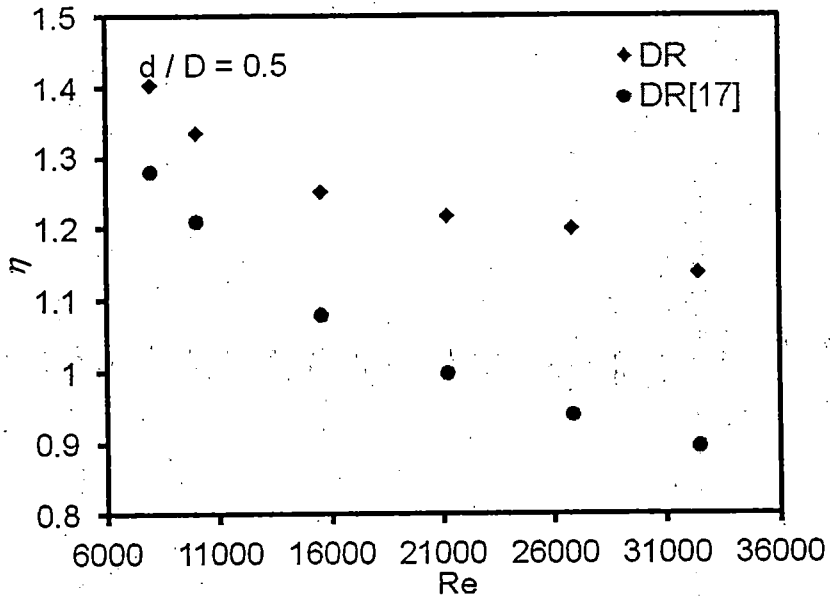


Fig. 20. Comparison of circular reference [17] and experimental value of the overall enhancement efficiency of conical rings with internal fins (DR) arrays at diameter ratio $d/D=0.5$.

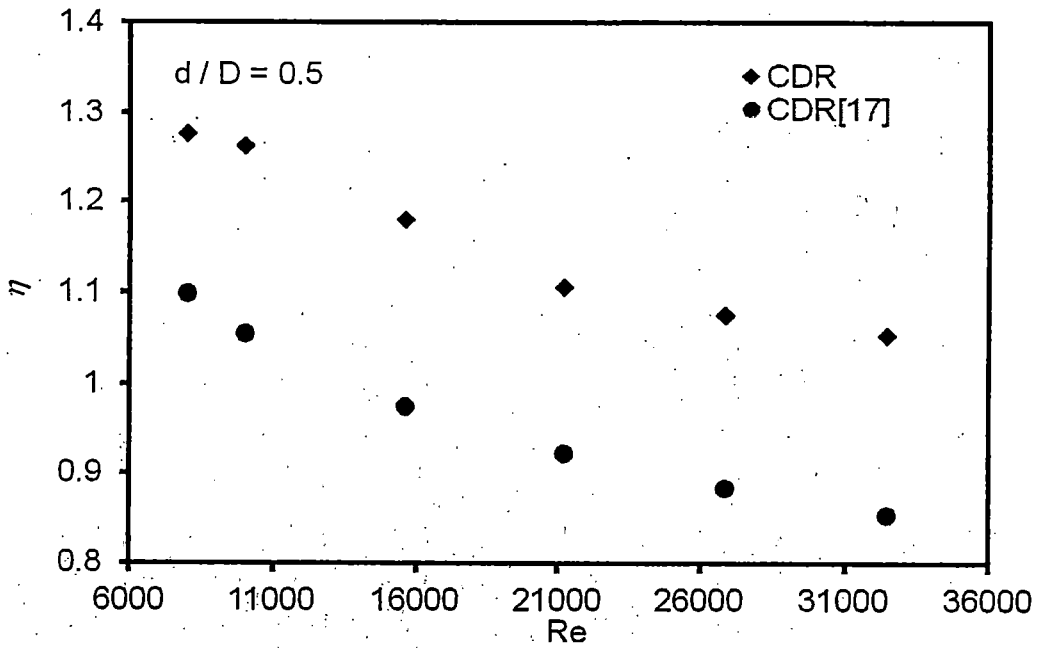


Fig. 21: Comparison of circular reference [17] and experimental value of the overall enhancement efficiency of conical rings with internal fins (DR) arrays at diameter ratio $d/D=0.5$.

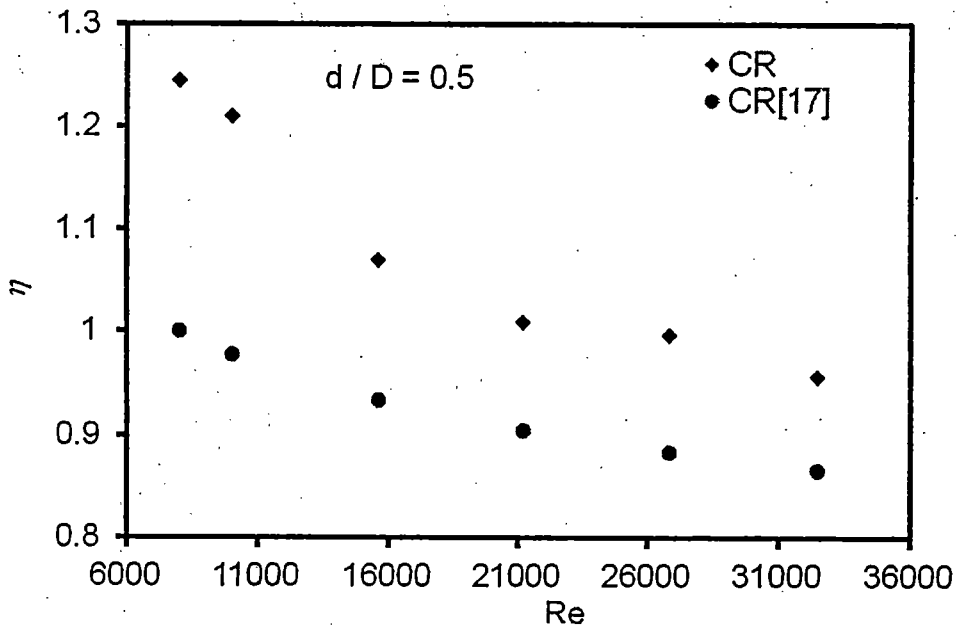


Fig. 22. Comparison of circular reference [17] and experimental value of the overall enhancement efficiency of conical rings with internal fins (DR) arrays at diameter ratio $d/D=0.5$.