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Effects of duct inclination angle on thermal entrance region of laminar and transition mixed convection

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Abstract

Experimental investigation was performed on the mixed convection heat transfer of thermal entrance region in an inclined rectangular duct for laminar and transition flow. Air flowed upwardly and downwardly with inclination angles from −90° to 90°. The duct was made of duralumin plate and heated with uniform heat flux axially. The experiment was designed for determining the effects of inclination angles on the heat transfer coefficients and friction factors at seven orientations ($\theta = -90°$, −60°, −30°, 0°, 30°, 60° and 90°), six Reynolds numbers ($Re \approx 420$, 840, 1290, 1720, 2190 and 2630) within the range of Grashof numbers from 6.8×10^3 to 4.1×10^4. The optimum inclination angles that yielded the maximum heat transfer coefficients decreased from 30° to −30° with the increase of Reynolds numbers from 420 to 1720. The heat transfer coefficients first increased with inclination angles up to a maximum value and then decreased. With further increase in Reynolds numbers, the heat transfer coefficients were nearly independent of inclination angles. The friction factors decreased with the increase of inclination angles from −90° to 90° when Reynolds numbers ranged from 420 to 1290, and independent of inclination angles with higher Reynolds numbers.

Keywords: Mixed convection; Heat transfer coefficient; Friction factor; Optimum inclination angle

1. Introduction

Mixed convection in ducts of various cross-sections and orientations can be applied to electronic cooling systems, solar energy systems, nuclear power technology, chemical process heat transfer, etc. Because of the numerously possible combinations of the duct shape and orientation, and of wall and entry boundary conditions, fully research of the pressure drops and heat transfer characteristics is necessary for the proper design of a specific system.

Due to the importance of the problem for engineering applications, a great deal of research efforts have been devoted to this topic. The majority of the existing literatures have addressed either the vertical or the horizontal configurations. The interaction of the natural and forced convection currents is very complex and difficult; it depends on many factors, such as parameters determining both forced and natural convection, the relative direction of the natural and forced convection to each other, the duct shape, the velocity profile at tube entrance and the heating surface boundary conditions, etc. [1]. Reviews on mixed convection have been reported by Metais and Eckert [2], Jackson [3], etc. Recently, there have been a variety of experimental, analytical and numerical investigations on mixed convection in horizontal and vertical channels [4–8]. Significant heat transfer enhancement may be realized by mixed convection in some situations, especially in opposed flows.

The investigations on mixed convection in inclined geometries are sparse in comparison with vertical and horizontal configurations, particularly under buoyancy opposed conditions. Early in 1966, Iqbal [9] made an analytical investigation on the influence of inclination angles on laminar mixed convection in an upward circular tube, the heat transfer coefficients increased up to a maximum and then

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Nomenclature

A_{is}	inner surface area of test section (m)
c	specific heat capacity (kJ/(kg K))
D_h	hydraulic diameter of test section (m)
d_1	width of test section (m)
d_2	width of heater unit (m)
f	friction factor
g	acceleration due to gravity (m/s^2)
Gr	Grashof number
Gz	Graetz number, $Gz = RePrD_h/L$
h	height of test section (m)
k	pressure drop increment due to entrance region
L	length of two pressure ports (m)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
ΔP	pressure drop (Pa)
q_{cv}	convective heat flux (kW/m^2)
Q	thermal energy (kW)
Re	Reynolds number
T	temperature (K)
ΔT	temperature difference (K)
u	air velocity (m/s)

Greek symbols

β	volumetric expansion coefficient (1/K)
λ	thermal conductivity (W/(mK))
μ	dynamic viscosity (Pa s)
ν	kinematics viscosity (m^2/s)
θ	inclination angle (deg)
ρ	density (kg/m^3)

Subscripts

a	of air flow
m	mean value
w	at the wall

decreased. The optimum inclination angle corresponding to the maximum heat transfer coefficient appeared to be a function of Rayleigh number, Reynolds number and Prandtl number. In most instances the optimum angle lied between 20° and 60°. Abou-Ellail and Morocs [10] found that the optimum inclination angles were between 30° and 45° for $Re = 50$ and 100, but the heat transfer coefficients monotonically decreased with increasing inclination angles for $Re = 500$. Later, Morcos [11] experimentally investigated the laminar mixed convection of water upwardly through an inclined rectangular channel, and reported that the optimum inclination angles were 30° for $Re = 100$, 250 and 500. Maughan [12] carried out experiment on mixed convection for airflow in a horizontal and upwardly inclined parallel plate channel, and found that the Nusselt numbers increased with inclination angles ranging from 0° to 30°. Lavine [13] demonstrated that the wall friction and Nusselt numbers between inclined parallel plates might vary monotonically or nonmonotonically with inclination angles, depending on the values of the Grashof number, Reynolds number and Prandtl number. An experiment on mixed convection of airflow in upwardly and downwardly inclined rectangular duct was performed by Lin [14]. Their experimental data indicated the Nusselt numbers decreased with inclination angles ranging from −20° to 20°. Busedra [15] made an analysis of laminar mixed convection in inclined semicircular ducts under buoyancy assisted and buoyancy opposed conditions. The heat transfer coefficients experienced a gentle increase with inclination angles up to a maximum and then decreased with further increase in inclination angles when the fluid flowed upwardly. Later, Busedra [16] conducted an experiment on laminar mixed convection of water flow in an inclined semicircular duct, and the Nusselt numbers were found to increase with inclination angles increasing from −20° to 20°. The mixed convection of airflow in a horizontal and upwardly inclined rectangular channel was experimentally investigated by Ozsunar [17]. From their data, the Nusselt numbers decreased with an increase in inclination angles from 0° to 30°. The present authors have also carried out experiments on the mixed convection in inclined rectangular ducts [18,19], with the inclination angles from −60° to 60°. The variations of the heat transfer coefficients and pressure drops with Reynolds numbers were presented. In 2007, Mohammed [20] experimentally investigated the laminar mixed convection in the thermal entrance region of inclined circular ducts, and the Nusselt numbers decreased with the inclination angles increasing from 0° to 90°. The above studies indicate that some aspects of the flow behavior do not always depend monotonically on the inclination angle, the Nusselt numbers may increase or decrease with the increase of the inclination angle under different conditions.

The ducts with various cross-sections, such as circular, parallel plates, semicircular duct, and rectangular duct, have been appeared in Ref. [9–20]. Mixed convection heat transfer, particularly for the laminar and transition flow, is strongly dependent on duct geometry and operation conditions, therefore, results of one duct geometry will not apply to other geometries. Moreover, Poskas [21] demonstrated that the level of heat transfer for assisted flow was closely related to the duct length and diameter in thermal entrance region, which partially gives the explanation why there are big differences on heat transfer data among different investigations.

To the best knowledge of the present authors, experimental investigations on the thermal entrance region of laminar and transition mixed convection in inclined ducts are very limited, particularly under both buoyancy opposed and assisted conditions. The effects of inclination angles on laminar mixed convection in pipes and ducts have received relatively less attention in spite of its importance in practical applications, no experimental investigation has been performed with the inclination angle ranging from $-90°$ to $90°$. The present study was concerned with the experimental investigation of laminar and transition mixed convection in a rectangular duct with air as the working fluid. The rectangular duct was made of duralumin plate and subjected to axial uniform heat flux. The investigation emphasis was placed on the effects of inclination angles on the heat transfer and friction factors with the inclination angle ranging from $-90°$ to $90°$, and also the effects of Reynolds numbers and Grashof numbers were presented.

2. Experimental apparatus and data reduction

2.1. Experimental apparatus

Tests were performed on the apparatus built for the mixed convection investigations at various inclination angles, as shown in Fig. 1. The experimental system consisted of three main parts: the wind tunnel, test section and measurement system. A chassis was used to control the inclination angle.

Air was driven by an induced draught fan and sent into the bellmouth opening, flowed through the filter and entered into the developing channel. The bellmouth, filter and developing channel together were adopted to reduce the fluctuations and achieve smooth flow. Then the air passed through the test section and flowed into the outlet channel, which reduced the effects of the disturbances from the ambient surrounding on the flow in the test section. Finally the air passed through the gas rotameter and was pumped out by the induced draught fan.

The test section consisted of a rectangular duct and a composite heater unit, as shown in Fig. 2. The rectangular duct was constructed of duralumin plate with wall thickness of 4 mm and length of 300 mm. The internal dimensions of the cross section were 46 mm × 20 mm. The developing and outlet channel exactly kept the same internal dimensions as the test section. The composite heater unit, with 6 mm in thickness and 300 mm in length, was composed by a foil and two side duralumin plates, and it was longitudinally mounted in the middle of cross duct section. Therefore the rectangular test section could be treated as two same parts with the aspect ration of 1.0. The foil was serpentine and made of 0.04 mm thick stainless steel, and the foil was electrically insulated by an insulating varnish and a sheet of 0.10 mm thick polytetrafluoroethylene fabric. Silicon gel was rammed into the gap between the heater unit and duct, and good physical contact was checked by a multimeter. Direct electric current was provided by a power supply and transferred to the foil through two copper bus-bars. The copper bus-bars were firmly attached at two ends of the foil, intending to produce the uniform heat flux. Power dissipation was determined by measuring the voltage drop and current across the foil. The voltage difference between the two ends of the foil was measured by a digital multimeter with an accuracy of 6.5 digital resolutions. The current was determined by an ammeter with an accuracy of 0.5%. The test section was thermally insulated from the ambient by the general Durkflex at the exterior surface, and it was thermally insulated from the developing channel and the outlet channel by Teflon filling pieces at two ends.

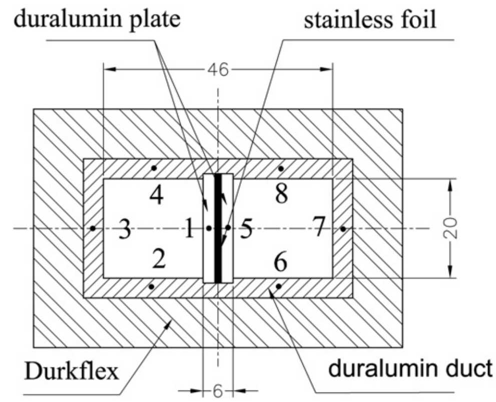
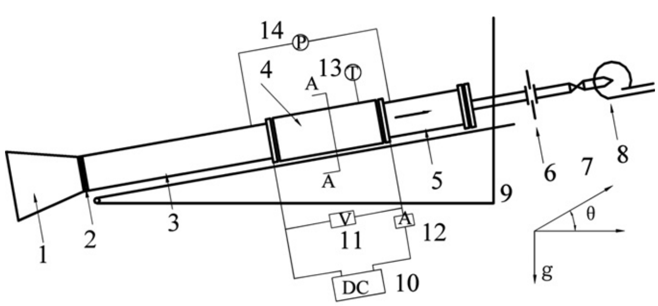


Fig. 2. Cross section of the test section.



1.bellmouth opening; 2.honeycomb; 3.developing duct; 4.test section; 5. outlet duct; 6. float rotometer; 7. air gate; 8. induced draught fan; 9.chassis; 10. DC power supply; 11. digital multimeter; 12. ammeter; 13. thermocouples; 14. difference pressure transducer

Fig. 1. Schematic diagram of the experimental apparatus.

The volume flow rate of air through the test section was measured by the float rotometer with an accuracy of 1.0%, which was regulated by a control valve. Because the pressure drop in the present experiment was as low as less than 10 Pa and hard to be measured exactly, two different pressure transducers were adopted. One was Setra Model 264, with the range from 0 to 25 Pa and an accuracy of 0.4%. The other was LPX9481 (GE Druck), with the range from 0 to 10 Pa and an accuracy of 0.1%. The higher pressure port was located upstream, 40 mm in front of the inlet of the test section. The lower pressure port was located downstream, 40 mm behind the test section.

All temperatures were measured by copper-constantan (T type, 0.2 mm) thermocouples. They were calibrated by the special calibration device and the second-order standard mercury thermometer with an accuracy of 0.1 K. The heater unit temperatures were measured at 14 axial locations downstream from the inlet of the test section. Small holes and shallow grooves were machined to install the thermocouples. The junction of each thermocouple was then affixed to the small hole with soldering tin and epoxy resin. Epoxy resin was also applied to the shallow groove to hold the thermocouple in place. Good physical contact between the thermocouple and the small groove was checked with a multimeter. To measure the circumferential duct wall temperature distribution, 6 thermocouple junctions were welded to the rectangular duct and 2 junctions were welded to the heater unit, as shown in Fig. 2. The inlet air temperature was measured at the beginning of the developing duct, and the outlet air temperature was measured at 20 mm behind the test section. All electric signals from thermocouples and differential pressure transducer were recorded via the automated measuring data acquisition system connected to the personal computer. Generally, after adjusting the desired value of parameters, the experimental system was allowed to run for about 4 h before steady state conditions were achieved.

2.2. Data reduction

The dimensionless parameters of Re and Gr were calculated from Eq. (1),

$$Re = \frac{u_m D_h}{\nu} \text{ and } Gr = \frac{g\beta(T_{w,m} - T_{a,m})D_h^3}{\nu^2} \tag{1}$$

where $T_{w,m}$ was the mean temperature of test section walls, $T_{a,m}$ was the mean air temperature, and D_h was the hydraulic diameter. $T_{w,m}$ was calculated from the axial heater unit temperatures measured by 14 thermocouples, and $T_{a,m}$ was calculated from the inlet air temperatures and outlet air temperatures. Moreover, D_h was given by Eq. (2),

$$D_h = \frac{2(d_2 - d_1)h}{(d_2 - d_1 + h)} \tag{2}$$

According to Fig. 2, $d_2 = 46$ mm, $d_1 = 6$ mm and $h = 20$ mm. Air properties in Eq. (1) were calculated at the average of the inlet and outlet air temperatures. According to Eq. (1), the values of Reynolds numbers and Grashof numbers were correlated with air properties. Therefore, when the air flow rate was fixed at a constant value, the increase of heat flux resulted in the increase of air temperature. Then Reynolds numbers decreased a little instead of keeping constant. In other words, at a fixed air flow rate, Reynolds numbers decreased with the increase of Grashof numbers. In liker manner, Grashof numbers decreased with the increase of Reynolds numbers when the heat flux was fixed.

To calculate the average heat transfer coefficients through the test section, the convective heat flux was required. As the heater unit and rectangular duct were all made of high conductivity materials and in good physical

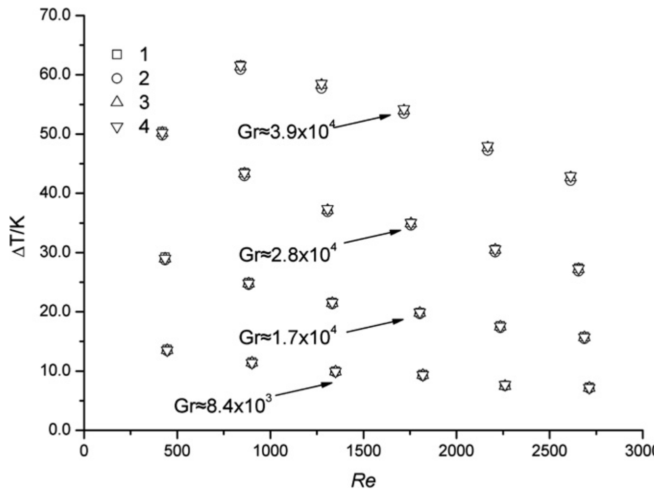


Fig. 3. Circumferential wall temperature distributions for $\theta = 0°$.

contact, the heat generated from the heater unit was rapidly conducted to the duct walls. Results of the wall temperature measurement for $\theta = 0°$ and $Re \approx 420$–2630 are shown in Fig. 3 with Gr about from 8.4×10^3 to 3.9×10^4. The temperature differences, labes of the ordinate, were calculated from the wall temperatures and inlet air temperature. The circumferential variation of wall temperatures was indicated by the readings of the four thermocouples 1, 2, 3 and 4 (seen in Fig. 2 for locations), where 1 represented the heater unit. Examination of Fig. 3 revealed that all the temperature differences between 1, 2, 3 and 4 under different conditions were less than 0.5 K. Thus the results presented in Fig. 3 indicated that the temperature distribution of the heater unit and duct walls was almost uniform. The boundary conditions simulated the electric resistance or nuclear heating for the limiting conditions of highly conductive wall material, known as H1 boundary condition. This has been examined by Shah [22] and Gian [23] for the highly conductive materials (e.g. copper, aluminum). Therefore, the convective heat flux q_c was calculated from Eq. (3),

$$q_{cv} = Q_a / A_{is} \tag{3}$$

where A_{is} was the inner surface area of the rectangular duct, and Q_a was the actual thermal energy gained by air in terms of its temperature rise,

$$Q_a = c_a \dot{m}_a \Delta T_a \tag{4}$$

The convective heat flux could also be obtained by an indirect method according to the energy equation. Comparisons showed that the convective heat fluxes by different methods approximately agreed with each other.

Therefore, the average Nusselt number of the test section was defined as

$$Nu_m = \frac{q_{cv} D_h}{(T_{w,m} - T_{a,m})\lambda_a} \tag{5}$$

The friction factor was defined as

$$f = \Delta P(D_h/L)/\left(\frac{1}{2}\rho u_m^2\right). \tag{6}$$

2.3. Uncertainty analysis

Uncertainty analysis was performed by applying the estimation method proposed by Moffat [24]. The uncertainty estimation of the average Nusselt number (with 95% confidence) took into account the following sources of error: air flow rate (float rotometer reading), air flow and test section temperature (thermocouple readings), and variable errors associated with the lack of repeatability caused by random disturbances and unsteadiness. Overall, the uncertainty in the average Nusselt number was estimated to be ±9.4%. The Reynolds number had a calculated uncertainty of ±6.5% based on the error in air flow rate (float rotometer reading), and Grashof number of ±4.9% based on error in air temperature and heater unit temperature (thermocouple readings). The uncertainty in the friction factor was estimated to be ±19.5% based on the errors in pressure drop (differential pressure transducer reading) and air flow rate (float rotometer reading).

3. Results and discussion

A total of 161 test runs were conducted in this investigation covering the following ranges of the parameters: $Re \approx 420$–2630, $Gr \approx 6.8 \times 10^3$–$4.1 \times 10^4$, $\theta = -90°$ to $90°$. Four different Grashof numbers were taken for each combination of Re and θ except that three different Grashof numbers were taken for $Re \approx 420$. The heat transfer coefficients and friction factors across the test section are presented in the following sections.

3.1. Horizontal ducts

A composite plot showing the effect of Reynolds numbers on the heat transfer coefficients in the horizontal duct is depicted in Fig. 4, for $Re \approx 420$–2630 and $Gr \approx 8.4 \times 10^3$– 4.1×10^4. It can be seen that the present heat transfer coefficients generally increased from 6.2 to 19.2 with the increase of Reynolds number. The data of laminar forced convection [25] and data in the entrance region of turbulent forced convection [26] had the same trend with the present heat transfer coefficients, but the present heat transfer coefficients of mixed convection were higher than data of forced convection. For the laminar mixed convection in horizontal pipes and ducts, Brown and Gauvin [27] have developed a correlation as following

$$Nu = 1.75\left(\frac{\mu_m}{\mu_w}\right)^{0.14}[Gz + 0.012(GzGr^{1/3})^{4/3}]^{1/3} \tag{7}$$

Fig. 4 shows that the present heat transfer coefficients are in satisfactory agreement with the predicted data. The deviations of the present heat transfer coefficients from Eq. (7) were less than ± 15.0% when the present heat

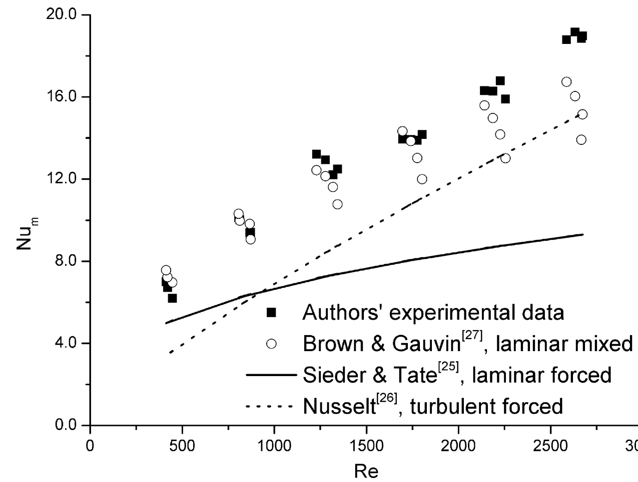


Fig. 4. Comparison of predicted average Nusselt number distribution with measured data for $\theta = 0°$.

transfer coefficients were no more than 14.2, with corresponding Reynolds numbers ranging from 420 to 1800. It could be deduced that the flow in the rectangular duct was laminar flow under these conditions. However, the deviation increased up to −26.4% when the present heat transfer coefficient was 18.9 and corresponding Reynolds number was 2630. The flow was not laminar flow and Eq. (7) could not be applied to these conditions.

Fig. 5 shows a composite plot about the effect of Reynolds numbers on the pressure drops. According to the figure, the variation of the present pressure drops versus Reynolds numbers agreed with that of the heat transfer coefficients in Fig. 4. The present pressure drops generally increased with Reynolds numbers. Moreover, the present pressure drops agreed with the values calculated from Lundgren [28]. The data calculated from Lundgren were based on the entrance region, with the following equation,

$$\Delta P = 0.5(fL/D_h + k)\rho u_m^2 \tag{8}$$

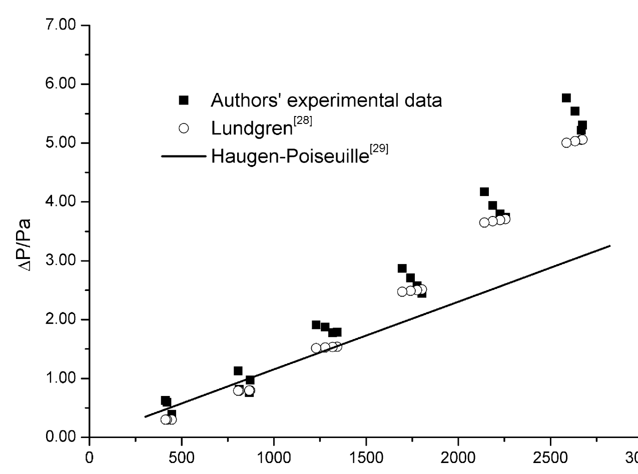


Fig. 5. Comparisons of the predicted pressure drop distribution with the measured data for $\theta = 0°$.

where $fRe = 56.908$ and $k = 1.515$ for square ducts. As mentioned above, the flow in the present experiment was in the thermal entrance region, and there was a heater unit in the middle cross section of the rectangular duct. The two factors resulted that the present pressure drop agreed with data of the entrance region.

The data calculated according to Hagen–Poiseuille equation [29] are also plotted in Fig. 5. The figure showed that the present pressure drops approximately agreed with the values calculated according to Hagen–Poiseuille equation when the Reynolds number was less than 1500. The present pressure drops were larger than the values from Hagen–Poiseuille equation when Reynolds numbers further increased, and the deviations increased with Reynolds numbers. Because the Hagen–Poiseuille equation was based on the fully developed laminar flow, and the flow in the present experiment likely started to transit from laminar flow when Reynolds numbers were larger than 1500.

3.2. Heat transfer coefficients

The effects of inclination angles on the heat transfer coefficients are depicted in Figs. 6–11 for various values of Reynolds numbers, respectively.

From the figures, the effect of inclination angles was progressively more significant with smaller values of Reynolds numbers, and the heat transfer coefficients may vary nonmonotonically or monotonically with inclination angles when the Reynolds numbers increased from 420 to 2630.

According to Figs. 6–9, the heat transfer coefficients firstly increased with inclination angles up to a maximum value and then decreased when Reynolds numbers were no more than 1800. The optimum inclination angles that yielded the maximum values of heat transfer coefficients were 30° for $Re \approx 420$–840 and $Gr \approx 6.8 \times 10^3$–$4.1 \times 10^4$ according to Figs. 6 and 7. Morcos [11] carried out experiments on laminar mixed convection in inclined rectangular

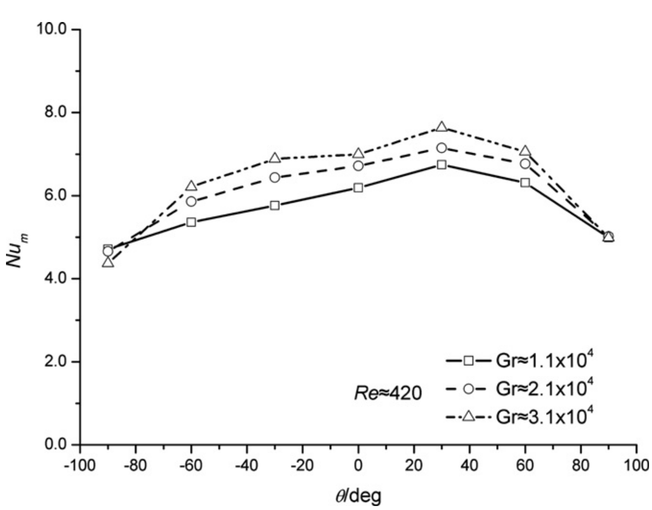


Fig. 6. The Nusselt numbers versus inclination angles for $Re \approx 420$.

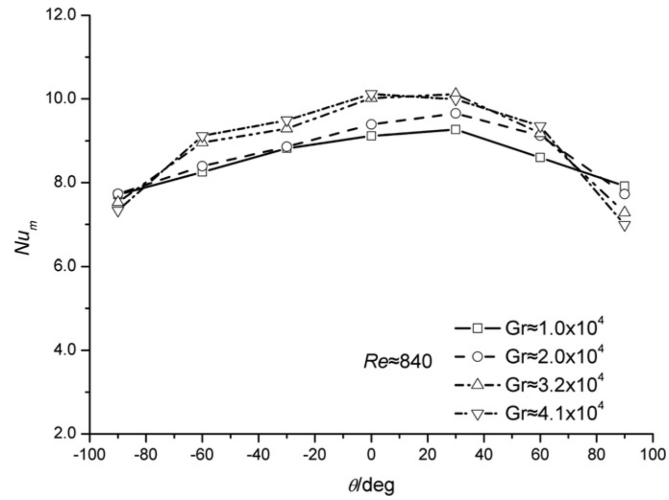


Fig. 7. The Nusselt numbers versus inclination angles for $Re \approx 840$.

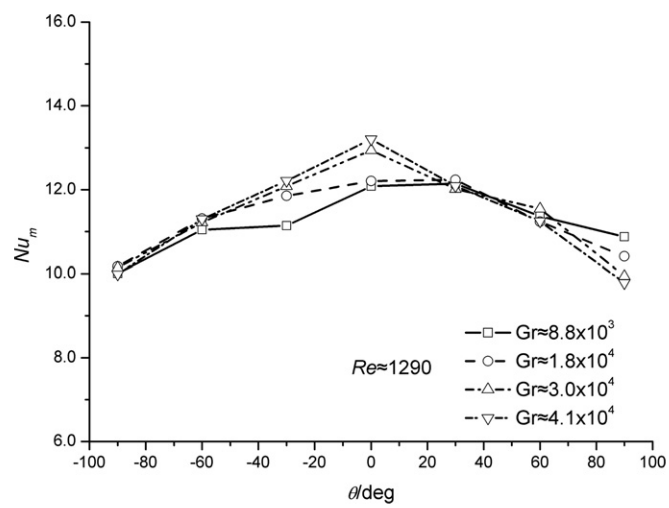


Fig. 8. The Nusselt numbers versus inclination angles for $Re \approx 1290$.

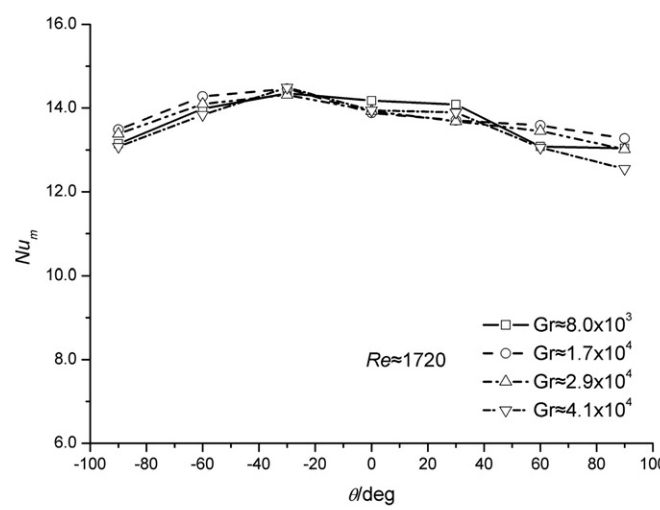


Fig. 9. The Nusselt numbers versus inclination angles for $Re \approx 1720$.

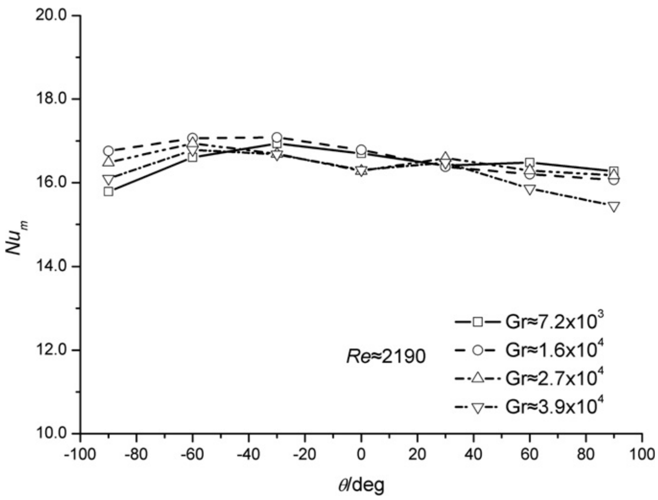


Fig. 10. The Nusselt numbers versus inclination angles for $Re \approx 2190$.

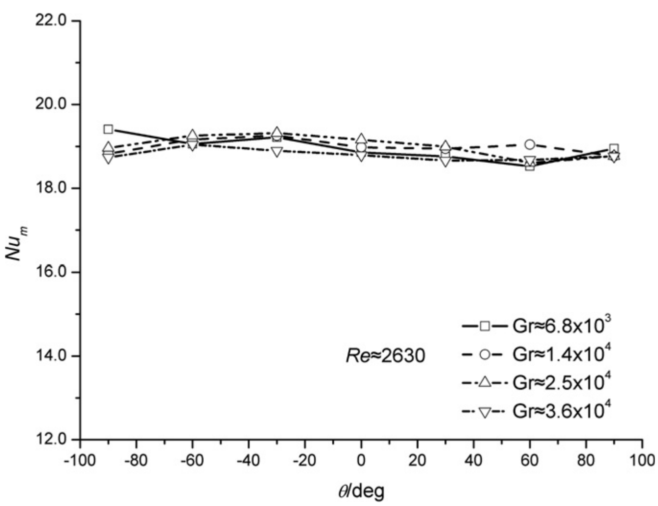


Fig. 11. The Nusselt numbers versus inclination angles for $Re \approx 2630$.

ducts, and their results had the same trends as the variation of the present heat transfer coefficients. They showed that the optimum angles were all 30° when $Re = 100$, 250 and 500, which agreed with the present experimental results.

The optimum angles resulted from different roles that the forced convection and the natural convection played. In the inclined ducts, the buoyancy force due to natural convection was divided into two parts: the forces parallel to the plates and normal to the plates. Therefore the heat transfer in inclined ducts was affected by three forces: the external force by forced convection, parallel force and normal force by natural convection. Different forces played different roles. For the laminar flow, the external force made the boundary layer thinner and the thermal resistance decreased. Therefore, the heat transfer was enhanced when Reynolds numbers increased and other parameters kept constant. The parallel force might bring different influences on the heat transfer for either assisted flow or opposed flow. For the assisted flow, the parallel force was in the same direction to the main flow and speeded up the flow in the boundary layer, and then the flow velocity in the core of ducts decreased due to the mass balance. Thus the boundary layer became thicker and the heat transfer was weakened. For the opposed flow, the parallel force was in the opposite direction to the main flow and disturbed the boundary layer. This resulted in the decrease of thermal resistance and finally induced the enhancement of heat transfer. The normal forces of different plates also brought different influences. The normal force to the lower plate disturbed the boundary layer and enhanced the heat transfer, and the normal force to the upper plate helped to preserve the boundary layer and weakened the heat transfer. When the optimum combination of the above forces was achieved, the heat transfer coefficient increased up to the maximum. Then the optimum inclination angles corresponding to the maximum heat transfer coefficients appeared to be a function of Reynolds numbers and Grashof numbers.

From Fig. 8, when the Reynolds number increased up to about 1290, the optimum inclination angle was 0°. As Reynolds number further increased up to 1720, the optimum inclination angle decreased to −30°. The variation of optimum inclination angles was due to the different roles that above forces played. Abou-Ellail and Morocs [10], and Busedra [15] also demonstrated that the optimum inclination angles varied under different conditions.

However, Figs. 10 and 11 revealed that the heat transfer coefficients were almost independent of inclination angles when Reynolds numbers were larger than 1800. This was due to the fact that the flow was little influenced by natural convection but dominantly influenced by forced convection.

3.3. Friction factors

The pressure drops and friction factors are necessary in the proper design of practical systems. But the

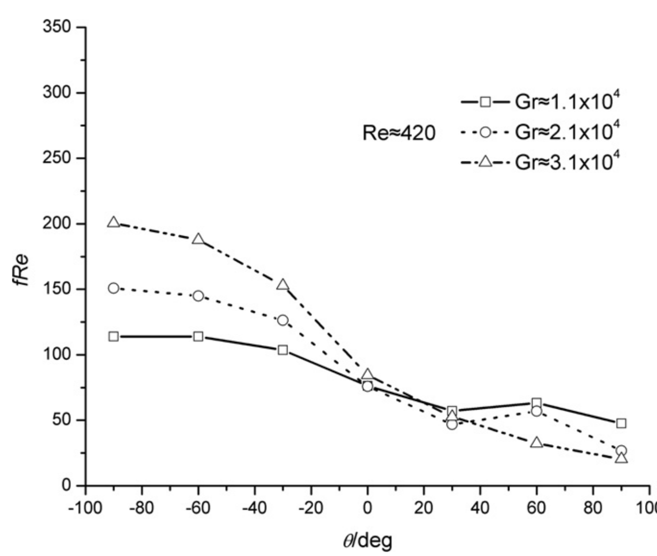


Fig. 12. The friction factors versus inclination angles for $Re \approx 420$.

measurement of pressure drops in mixed convection flow is a very difficult problem, because the actual pressure differences are quite small. Figs. 12–17 show the effects of inclination angles on the friction factors at different Reynolds numbers, with all the pressure drops less than 6.0 Pa. Here, $(fRe)$was used to represent the friction factors.

Figs. 12–14 show that the friction factors decreased with the increase of the inclination angle from $-90°$ to $90°$ when $Re \approx 420$–1290, which was mainly due to the parallel buoyancy force. For the opposed flow ($\theta < 0°$), the parallel force was in the opposite direction flow to the main flow and disturbed the boundary layer, which induced the increase of pressure drops and friction factors. As the absolute values of negative inclination angles increased, the parallel force increased and the boundary layer was further disturbed. This finally resulted that the friction factors became larger when the absolute values of negative inclination angles increased. For the assisted flow ($\theta > 0°$), the parallel force was in the same direction to the forced flow and helpful to preserve the boundary layer. This resulted in the decrease of the friction factors. The larger the positive inclination angles were, the more the friction factors decreased. Therefore, the friction factors decreased with the inclination angle increasing from $-90°$ to $90°$.

When Reynolds numbers were over 1500, the friction factors were little dependent on the inclination angles according to Figs. 15–17. The reason was that the forced convection made a greater impact on the mixed convection with the increase of Reynolds number, and then the variation of buoyancy force due to the variation of inclination

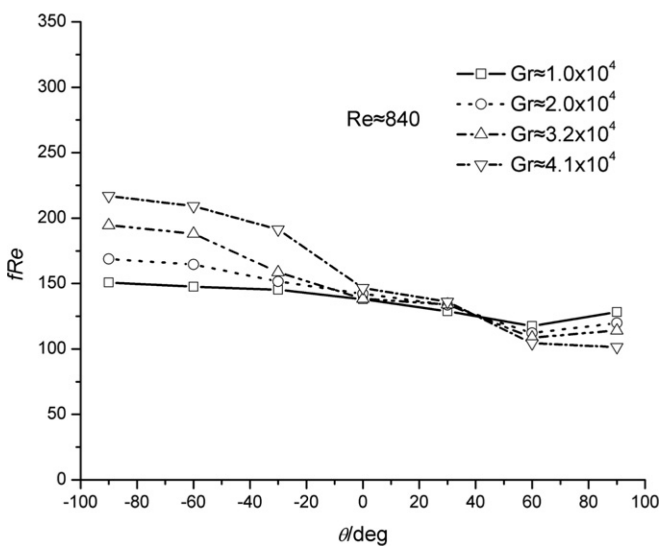


Fig. 13. The friction factors versus inclination angles for $Re \approx 840$.

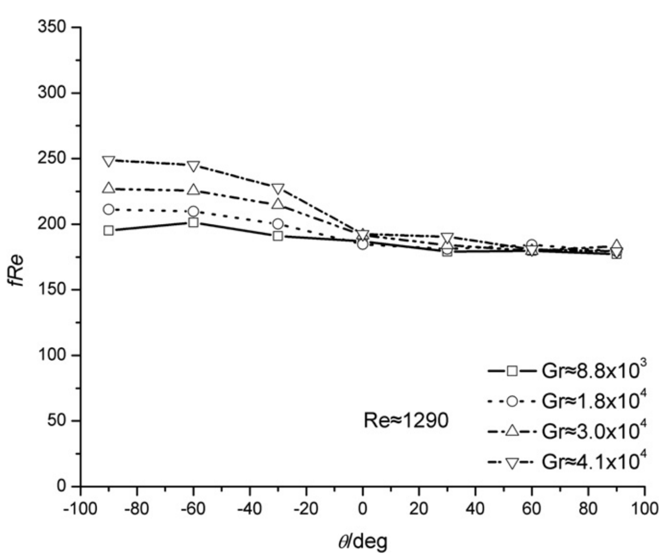


Fig. 14. The friction factors versus inclination angles for $Re \approx 1290$.

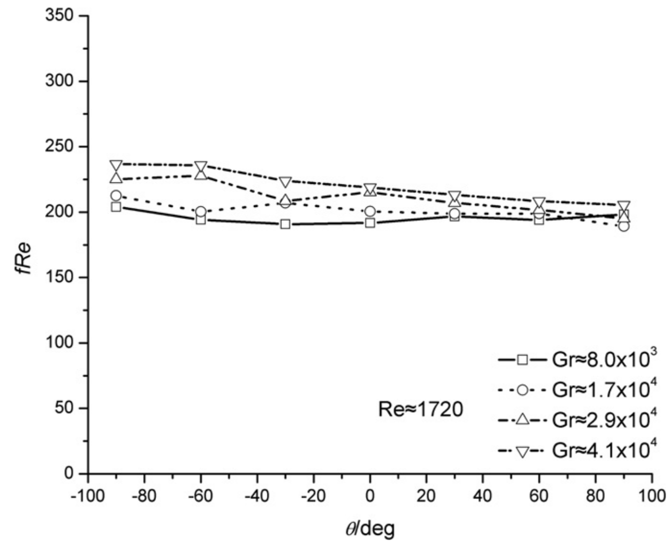


Fig. 15. The friction factors versus inclination angles for $Re \approx 1720$.

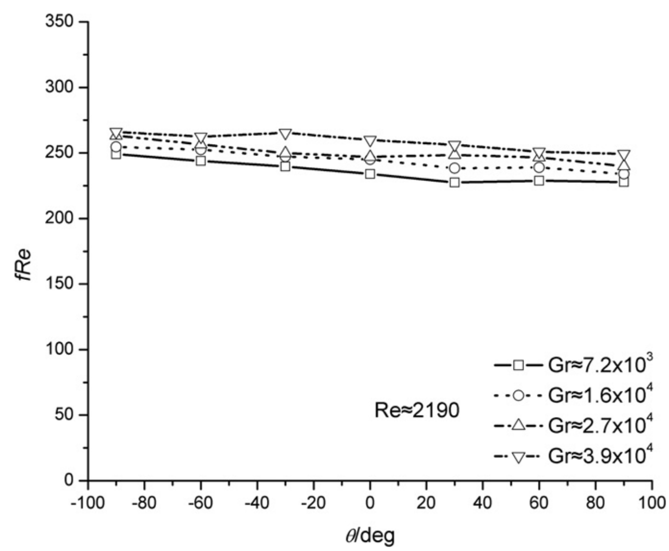


Fig. 16. The friction factors versus inclination angles for $Re \approx 2190$.

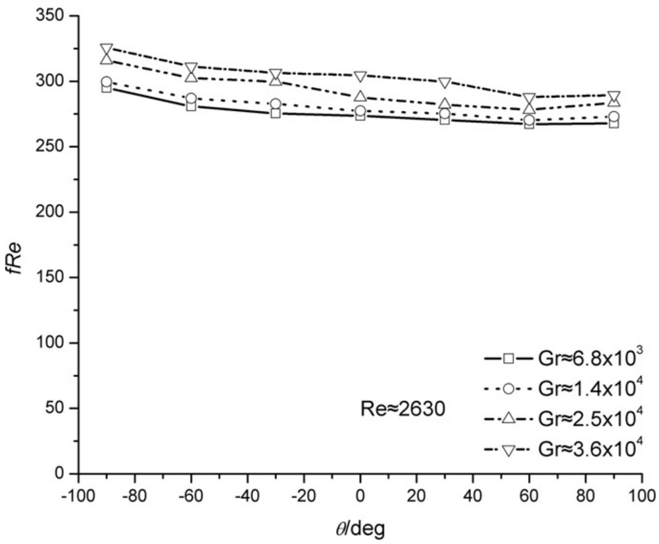


Fig. 17. The friction factors versus inclination angles for $Re \approx 2630$.

angles had little influence on the flows. Therefore the friction factors were almost independent of inclination angles.

From the variation of heat transfer coefficients and friction factors, the flow in the present experiment may start to transit from laminar mixed convection to turbulent mixed convection when Reynolds numbers were about from 1500 to 1800. According to Metais and Eckert [2], transition from laminar forced or mixed convection to turbulent mixed convection through vertical or horizontal tubes was possible at Reynolds numbers as low as 1000, which was in satisfactory agreement with the present experimental results.

4. Conclusions

Experimental investigation was performed to study the effect of inclination angles on the flow heat transfer characteristics of laminar and transition mixed convection in inclined rectangular ducts. The test results were obtained with air as the working fluid included seven orientations ($\theta = -90°$, $-60°$, $-30°$, $0°$, $30°$, $60°$ and $90°$), six Reynolds numbers for each orientation ($Re \approx 420$, 840, 1290, 1720, 2190, and 2630), and Grashof numbers from 6.8×10^3 to 4.1×10^4. The experimental heat transfer coefficients and pressure drops were compared with the values calculated from the literature. From the investigation, the following conclusions can be drawn:

1. The heat transfer coefficients and pressure drops of horizontal orientation generally increased with the Reynolds numbers, and the pressure drops were less than 6.0 Pa. This was due to the enhancement of forced convection when the Reynolds numbers increased.
2. The optimum inclination angles decreased from 30° to −30° when Reynolds numbers increased from 420 to 1720 and Grashof numbers ranged from 6.8×10^3 to 4.1×10^4. The heat transfer coefficients first increased with the inclination angles up to a maximum and then decreased. As Reynolds numbers further increased, the heat transfer coefficients were nearly independent of inclination angles. This was due to the different influences on the mixed convection that the forced convection and natural convection exerted when the parameters varied.
3. The friction factors decreased with the increase of inclination angles from −90° to 90° when Reynolds numbers ranged from 420 to 1290. However, when Reynolds numbers further increased, the friction factors were almost independent of inclination angles.

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