Chuang Wen
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Advances in Heat Transfer and Thermal Engineering

Proceedings of 16th UK Heat Transfer Conference (UKHTC2019)

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Preface

Nowadays, we are facing ever-severe crisis of conventional energy resources and pollutions, increasing demand for new energy resources and applications, and significant challenges for the technologies of efficient cooling, heat and mass transfer enhancement, and effective thermal management, etc. These have become crucially important for almost all engineering area and industries such as mechanical, aerospace, civil and building, chemical and process, electric and electronic, pharmaceutical and medical, as well as power industries.

Over the more than 100 years development, heat transfer has now become a cross-disciplinary subject. The study on micro-nano scale heat transfer has played an important role in the research progress of material science, and the development of biomedical engineering, etc. The 16th UK Heat Transfer Conference (UKHTC2019) addressed these challenges. The conference gathered the UK active and leading researchers typically many young and new academic colleagues, as well as the researchers from heat transfer communities of Europe, North Americans, Australia, South Africa, Kuwait, Japan, and China, etc.

The conference was aimed at a closer collaboration and cooperation between the UK and international scholars in the field of heat transfer. The UK National Heat Transfer Committee organised this conference biennially to provide an innovative platform for scholars in thermal engineering to share and exchange new ideas and solutions. Six plenary and three keynote lectures and more than 190 papers were presented at UKHTC2019 throughout two days in four sets of seven parallel oral sessions. The proceedings in the title of *Advances in Heat Transfer and Thermal Engineering* contains selected and the authors agreed extended abstracts or papers that cover almost all the topics in heat transfer and thermal engineering.

We would like to thank all authors for their contributions to UKHTC2019 and thank the staff members of the University of Nottingham to provide active assistance during the preparation stage of this conference. We send our sincere gratitude to the dedicated reviewers for their time and contribution to improve the scientific quality of the manuscripts. We also would like to acknowledge the support received from the sponsors. We hope that the emerging solutions described in the conference proceedings will inspire our academic and industrial communities to create, innovate, and build a more energy-efficient world.

Nottingham, UK

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Convection

Natural Convection from Heated Surface-Mounted Circular Cylinder

H. Malah, Y. S. Chumakov, and S. Ramzani Movafagh

1 Introduction

In recent years, there are more efforts on natural convection heat transfer from a horizontal cylinder, because of its practical applications. However, unconfined cylinder is well studied; the effect of introducing end-walls on the heat transfer rate of cylinder is considerably less investigated [1]. By development of computers and enhancement of advanced computational techniques, many studies of flow over a bluff body relevant to solid wall have been performed numerically [2], although experimental studies keep their place among researchers' efforts because of their advantages [3]. All numerical and experimental studies confirmed the expected arise on the heat transfer rate in the upstream region of the cylinder. However, the flow configuration, bluff body geometry and applied conditions on solid walls affect the arising flow [4, 5]. In this study, a numerical model of a heated horizontal circular cylinder mounted on vertical isothermal plate is employed to evaluate the natural convection heat transfer. To quantify the effect of vertical plate on the heat transfer from the cylinder surface, the aspect ratio of the cylinder (H/D) is selected equal to 0.6, in order to immerse in the arisen boundary layer on the vertical plate entirely. This geometrical configuration is evaluated on the vertical plate at fixed Grashof number equals 3×10^8 that represents laminar Grashof number. As a result, we describe the three-dimensional characteristics of natural convection heat transfer, which affect flow around the circular cylinder mounted on vertical heated plate. The results proved the significant effect of height of cylinder on the heat transfer rate from circular cylinder surface in the case of

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laminar natural convection flow. This study improves fundamental understanding of the buoyancy-induced flows around three-dimensional obstacles in different industrial applications to address the anticipated needs to enhance the rate of heat transfer and safety simultaneously.

2 Computational Methodology

In this work, in order to develop a laminar boundary layer on the heated vertical plate, the cylinder was mounted on an isothermal rectangular plate, which its dimensions considered equal to 100D in vertical direction (Y) and 7D in lateral direction (X). The vertical plate temperature is set to 333.15 K. In order to achieve a developed laminar incoming flow around the cylinder, the vertical position of cylinder from leading edge of rectangular plate is equal 30D, which provides Grashof number for laminar flow equals to 3×10^8. In addition, the computational domain was extended 9D from leading and trailing edge of plate and 10D normal to the plate (Z-direction) in order to ensure impermeability and slip in these regions.

In the analysed case of high aspect ratio cylinder, which performed in the similar conditions as present study, the computed thickness of incoming boundary layer on the vertical plate was equal to 0.9D [4]. The cylinder diameter (D) was equal to 0.02 m with fixed surface temperature at 353.15 K. The cylinder height (H) is fixed to 0.6D, in order to immerse in the laminar boundary layer entirely.

The schematic configuration of problem, its dimensions and imposed boundary conditions are shown in Fig. 1a. In Fig. 1a, the solid walls were applied no-slip boundary condition. The boundary condition, which called “Opening” in Fig. 1a, refers to penetrable side of computational domain in constant pressure.

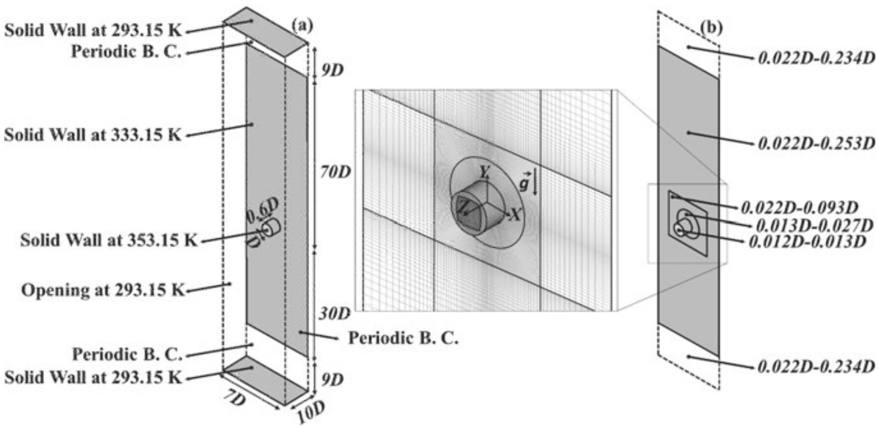

Fig. 1 Schematic of the case geometry and computational details: **a** problem configuration, **b** multi-blocked grid layout

In this work, the case geometry discretized by using body-fitted mesh. The multi-blocked grid in *XY* plane forms two-dimensional grid, and the range of its cells size in different region is shown in Fig. 1b. The two-dimensional grid, which consists of 41 thousand cells, was clustered to the vertical plate over *Z*-axis with a coefficient equals to one and generates the three-dimensional grid layout. The cells' size over *Z* spatial orientation is set to 0.02*D*. The three-dimensional mesh grid consists of approximately 4.8 million hexahedron cells.

In this study, a time-based numerical simulation was performed by using a commercial code (ANSYS FLUENT 16.2). The numerical model is based on the momentum and the energy balance equations, which were coupled by considering the fractional step algorithm and solved by using the Boussinesq approximation. The governing equations were discretized using second-order accurate schemes for all the spatial derivatives. Lastly, the computations were run up to 300 s in physical time with a time step equal to 0.002 s.

3 Results

In this work, in order to survey on characteristics of natural convection heat transfer around surface-mounted circular cylinder, the localized Nusselt number related to angular coordinate (Nu_θ) is determined on the solid surface of circular cylinder. To aim this purpose, the spatial angular coordinates were considered as angles, on which zero angle refers to leading edge of circular cylinder on *YZ* plane. In order to investigate the effect of height of cylinder on the heat transfer rate, the local Nusselt numbers at different *Z* coordinates along height of cylinder within laminar boundary layer thickness were presented in Table 1. Table 1 illustrates the computational results

Table 1 Local Nusselt number (Nu_θ) comparison

Z/*D*	Source	0	30	60	90	120	150	180
0.1	Present	12.39	11.49	8.43	4.45	3.06	3.82	6.21
	[5]	12.56	11.60	8.39	4.31	5.16	10.63	7.58
0.2	Present	21.36	20.50	14.99	6.92	2.06	3.31	5.41
	[5]	21.38	20.46	14.77	6.42	4.59	5.91	3.90
0.3	Present	24.55	23.79	18.13	8.56	2.19	2.62	3.36
	[5]	24.45	23.69	17.89	8.02	3.64	4.83	3.18
0.4	Present	24.95	24.29	18.83	9.44	2.60	2.03	2.03
	[5]	24.36	23.66	18.16	8.71	2.82	3.89	3.32
0.5	Present	25.51	24.94	19.85	10.83	3.75	2.63	1.91
	[5]	23.10	22.47	17.62	9.23	2.67	2.95	3.26
0.6	Present	30.36	30.37	26.01	17.22	9.31	8.50	7.96
	[5]	21.50	20.94	16.78	9.73	3.02	2.61	3.31

at seven discrete angular coordinates on the cylinder surface. In addition, in order to investigate the effect of cylinder aspect ratio on transferred heated flow from cylinder, the computed local Nusselt numbers of the work [3], where the results of high aspect ratio cylinder were provided, are included in Table 1 for comparison.

Based on the presented values in Table 1, the local Nusselt numbers decrease from the leading edge ($\theta = 0°$) to the trailing edge ($\theta = 180°$) of the cylinder for each Z coordinate. In the downstream region of the cylinder ($\theta = 120°$), there is a rapid decline in value of local Nusselt numbers, and after that ($\theta = 150°$) the Nusselt number values experience a slight increase. Since the rate of natural convection heat transfer is proportional to the buoyancy of the fluid, the local Nusselt numbers increase along Z-axis from confined end-wall (rectangular vertical plate) to the unconfined end of circular cylinder.

Although there is a good agreement between the values of local Nusselt number in the present analysis and results of high aspect ratio study [5], Table 1 demonstrates two regions on the cylinder surface, where the rate of convection heat transfer is comparable between low and high aspect ratio cylinder. The first region is at the unconfined end of cylinder (around $Z/D = 0.6$), where incoming flow can bypass the cylinder in the case of low aspect ratio cylinder, so there is anticipated a dramatically increase in the local Nusselt numbers. The second zone is downstream region of cylinder (from $\theta = 120°$ to $\theta = 180°$), where the arisen heated flow from the high aspect ratio cylinder acts as a separate source of heat generation. Since the high aspect ratio cylinder crosses the formed boundary layer entirely, arisen heated flow interacts with the incoming boundary layer and leads to an obvious increase in the rate of convection heat transfer.

An overall view of the data in Table 1 demonstrates the fact that the leading edge of surface-mounted circular cylinder ($\theta = 0°$) is a specific line for this problem. Although the local Nusselt number is practically constant for a long cylinder [5], the local Nusselt number increases for a short cylinder on the leading edge ($\theta = 0°$) along cylinder height (Z-direction). The localized Nusselt number in the wake region of the cylinder is qualitatively similar for different variants, representing a slow, almost monotonic decrease in local Nusselt number with increasing spatial angular coordinates around the cylinder.

4 Conclusions

By comparing the numerical results of arisen convection heat transfer rate in present work (low aspect ratio) with the results of high aspect ratio cylinder [5], the significant effect of cylinder aspect ratio on the Nusselt number in the case of laminar natural convection flow is demonstrated.

Maximum values of local Nusselt number are observed at the confined end of cylinder near the vertical plate. These values are located in laminar incoming boundary layer, which arose on the heated vertical plate. Furthermore, heat transfer coefficient decreases from leading edge of cylinder in upstream region to trailing edge in the downstream of the cylinder.

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Experimental Investigation of Transitional Flow Forced Convection Heat Transfer Through a Smooth Vertical Tube with a Square-Edged Inlet

Abubakar I. Bashir, Marilize Everts, and Josua P. Meyer

1 Introduction

Limited work has been done on forced convection heat transfer in the transitional flow regime, especially in horizontal tubes with higher heat fluxes where the uncertainties are low. Forced convection experiments in horizontal tubes are challenging to perform because of the difference in density between the fluid near the surface (hot) and near the center of the tube (cold) that cause buoyancy effects and lead to mixed convection. Mixed convection can change the heat transfer characteristics in the laminar and transitional flow regimes significantly. For forced convection, the theoretical fully developed laminar flow Nusselt number is 4.36 (for a constant heat flux boundary condition). For mixed convection, the Nusselt numbers can increase up to 180–520% higher than 4.36 [1–3] due to buoyancy effects. In vertical tubes, the buoyancy effects can be reduced as the flow is in the same direction as the buoyancy force and is mostly suppressed at higher Reynolds numbers. Therefore, forced convection conditions can be achieved in the laminar and transitional flow regimes of a smooth vertical tube, even at higher heat fluxes.

Ghajar and Tam [3] found that the boundaries and the heat transfer characteristics of the transitional flow regime were inlet dependent. Everts and Meyer [1] investigated the effect of buoyancy on the heat transfer in the transitional flow regime and found that the transition Reynolds numbers were significantly affected by the buoyancy effects. Furthermore, buoyancy effects increased with increase in heating, and therefore, heating also changes the transition boundaries. However, these analyses

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focused on mixed convection conditions in horizontal tubes. It is important to investigate the heat transfer characteristics for pure forced convection in the transitional flow regime in order to fundamentally understand the behavior of transition heat transfer without the influence of buoyancy. Therefore, the purpose of this study was to experimentally investigate the single-phase forced convection heat transfer characteristics of the transitional flow regime in a smooth vertical tube, with a square-edged inlet, heated at constant heat flux.

2 Experimental Setup

The schematic of the experimental facility is shown in Fig. 1 and water was used as working fluid. A magnetic gear pump was used to pump the water from a storage tank to the flow meters, flow-calming section and inlet section and then to the test section. After the test section, the heated water returned to the storage tank for cooling and recirculation. The flow-calming section was placed prior to the test section to ensure a uniform flow distribution through the inlet section and test section, because transition is inlet dependent. A square-edged inlet geometry was used for all the experiments.

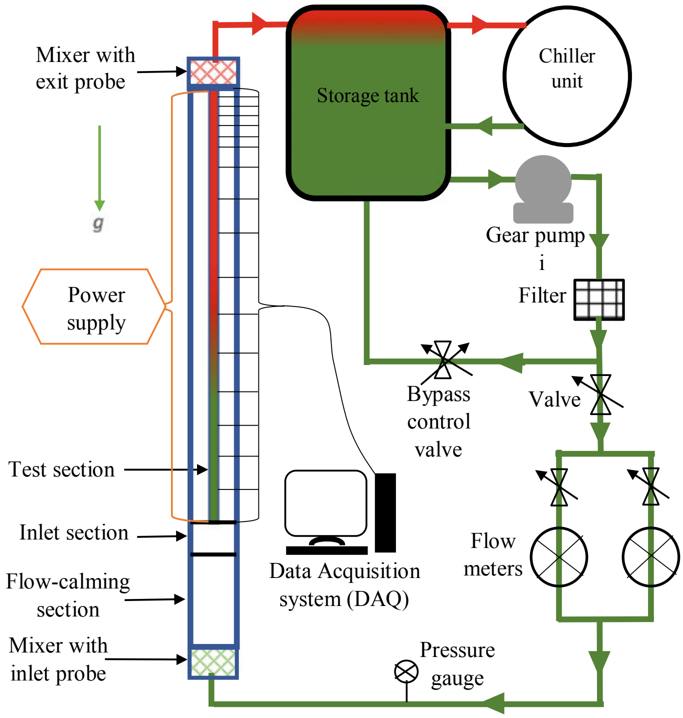

Fig. 1 Schematic of the experimental facility

The test section was a smooth hard drawn copper tube with an inner diameter of 5.1 mm and a heated length of 4.52 m (maximum length-to-diameter, x/D_i of 886). Twenty-one thermocouples were attached to the test section to measure the local wall temperatures. The inlet and exit bulk fluid temperatures were measured using two Pt100 probes placed inside the inlet and exit mixers, respectively. The test section was heated at heat fluxes of 1, 4, 6 and 8 kW/m^2 using a direct current (DC) power supply. At a Reynolds number of 2000, the flow was found to be fully developed from x/D_i = 416, because the heat transfer coefficients became relatively constant along the tube length. Therefore, the local results at x/D_i = 592 were used for the fully developed flow analyses. The test section was set at vertical upward flow direction to avoid the effect of buoyancy or free convection that might cause mixed convection. The experiments were performed for Reynolds numbers between 1000 and 6000 to cover the entire transitional flow regime as well as sufficient parts of the laminar and turbulent flow regimes.

The setup was validated against the literature by comparing laminar and turbulent flow heat transfer coefficients with well-known correlations. The laminar flow heat transfer results were compared using the flow regime map of Metais and Eckert [4] for constant heat flux in vertical tubes. All the heat transfer results fell within the forced convection region of the Metais and Eckert [4] map. Furthermore, at a Reynolds number of 1000, the laminar forced convection Nusselt number was 4.41, which is within 1.1% of 4.36. Thus, the forced convection condition was confirmed in the laminar flow regime up to the start of transition. In the turbulent flow regime, the maximum deviation of the heat transfer coefficients from Gnielinski [5] correlation was 3.9%.

3 Results

Figure 2a compares the local fully developed heat transfer results in terms of Nusselt number (Nu = hD_i/k) as a function of Reynolds number at $x/D_i = 592$. The Nusselt numbers for all the different heat fluxes in the laminar flow regime were approximately the same and approached the theoretical forced convection Nusselt number of 4.36 for a constant heat flux boundary condition. This indicated that there is negligible or no buoyancy effects and confirmed forced convection conditions for all the heat fluxes up to the start of transitional flow regime. However, as the Reynolds number increased and the flow approached the transitional flow regime, the laminar flow Nusselt numbers of all the heat fluxes increased slightly, which might be due to the effect of variable fluid property (viscosity). As expected, there was a negligible difference between the results of the different heat fluxes in the turbulent flow regime, therefore, the flow was also dominated by forced convection conditions. Because both the laminar and turbulent flow regimes were dominated by pure forced convection heat transfer, it confirmed that the entire transitional flow regime was also dominated by forced convection.

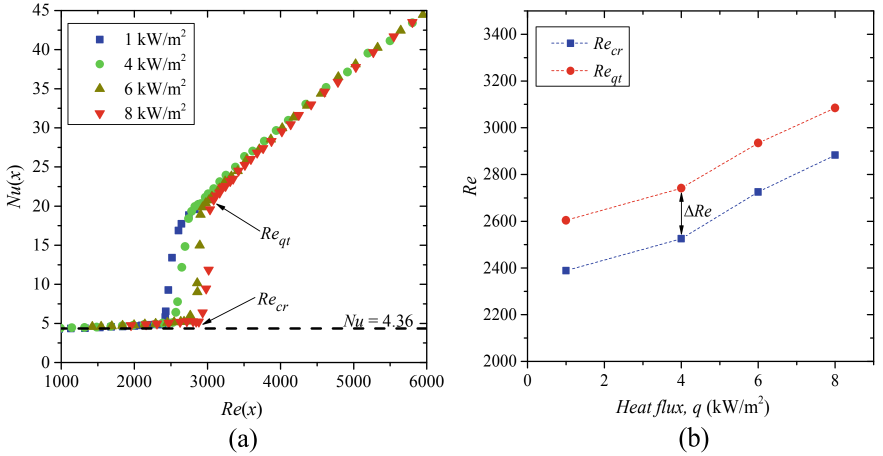

Fig. 2 Comparison of **a** fully developed local Nusselt numbers as a function of Reynolds numbers at x/D = 592 for the different heat fluxes and **b** Reynolds numbers at the start (Re_{cr}) and end (Re_{qt}) of transitional flow regime in **a** as a function heat flux

For all the heat fluxes in Fig. 2a, transition occurred at the same mass flow rate of approximately 0.00890 kg/s, while the critical Reynolds numbers at the start of transition increased with increase in heat flux. As the heat flux was increased for a constant mass flow rate, the increased fluid temperature led to a decreased viscosity which in turn caused the Reynolds numbers to increase. At a heat flux of 1 kW/m^2, transition occurred at a critical Reynolds number of 2388, while at 8 kW/m^2, the critical Reynolds number increased to 2883 for the same mass flow rate. Similarly, transition ended at approximately the mass flow rate but the Reynolds number at end of transition also increased with increased heat fluxes. Figure 2b compares the transition Reynolds numbers for the different heat fluxes in Fig. 2a. It followed that both the Reynolds numbers at the start (Re_{cr}) and end (Re_{qt}) of the transitional flow regime increased simultaneously with increasing heat flux. It also showed that the width of the transitional flow regime, defined by Everts and Meyer [1] as $\Delta Re = Re_{cr} - Re_{qt}$, for all the heat fluxes was approximately equal and ranged between 203 and 219. This is different from that of mixed convection condition as was found by Evert and Meyer [1] in horizontal tubes. For mixed convection condition, the width of the transitional flow regime was significantly affected by free convection effects and therefore decreased with increasing heat flux.

4 Conclusions

Single-phase forced convection heat transfer characteristics of the transitional flow was experimentally investigated using a smooth vertical tube with a square-edged