THERMAL CHARACTERISTICS OF TURBULENT FLOWS UNDER THE EFFECTS OF RIB, CURVATURE AND REYNOLDS NUMBER

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ABSTRACT

An experimental study of convective heat transfer of flows over concave, convex and consecutive ribbed walls in a turbulent boundary layer was presented. The velocity and temperature measurements carried out in a wind tunnel were recorded by a constant-temperature hot wire anemometer and a copper-constant thermocouple, respectively. In the experiments, the Reynolds number varied from $3.1x10^6$ to $4.5x10^6$ which correspond to the turbulent flow. The thermal characteristics of flow over the different walls were compared with the each other and those of the flat plate under the effect of Reynolds number. The measurements indicated that the concave and ribbed wall effectively enhanced the heat transfer performance while the convex wall reduced. The results were also focused on the effect of the Reynolds number on the heat transfer augmentation. The maximum average heat transfer coefficient was obtained over the concave wall and the heat transfer increased with Reynolds number for all the walls.

KEYWORDS: *Heat Transfer Enhancement, Turbulent Flow, Ribbed Wall, Concave Wall, Convex Wall*

I. INTRODUCTION

The thermal characteristics of turbulent flows over ribbed, concave and convex walls are of great importance for engineering applications due to the heat transfer enhancement. There have been several previous investigations on ribbed surfaces such as, Wang and Sunden [1] were experimentally investigated the turbulent heat transfer in a square duct with ribs. A study on the computation of enhanced turbulent heat transfer in a channel with periodic ribs was carried out by Tsai et al. [2]. Leung et al. [3] recorded an increase of 133% with traverse rectangular ribs. Mohammed [4] investigated the air-cooling characteristics of an electronic devices heat sink with various square modules array. An investigation in a blocked channel for heat transfer enhancement was performed by Beig et al. [5]. An experimental study was examined on the forced convective flow over heated blocks by Chen and Wang [6]. Braun et al. [7] carried out an experimental and numerical investigation of turbulent heat transfer in a channel with periodically arranged rib roughness elements. The turbulent flow field and heat transfer enhancement of mixed convection in a horizontal block-heated channel was investigated by Perng and Wu [8]. A numerical investigation considered flow in a channel with a heater was studied by Alves and Altemani [9], who reported that the heater average surface temperature decreased as the airflow rate increased. Tsay and Cheng [10] studied on the two-dimensional forced convection in a channel with heat generating blocks and indicated that heat transfer increased with block height. On the other hand, an increase of more than 100% due to the concave curvature in laminar flows and more than 20% in turbulent flows was recorded by Umur [11] and Thomman [12] respectively. A decrease of around 20% due to convex curvature in turbulent Stanton numbers was observed by Thomman [12] and Mayle et al. [13] and Gibson et al. [14]. Wang and Simon [15] reported 5% to 10% decrease in heat transfer due to convex curvature. Crane and Sabzvari [16] also showed that concave curvature caused higher Stanton number than flat plate values, while Turner et al. [17] and Wright and Schobeiri [18] reported the convex curvature decreased heat transfer as a function of surface curvature and flow rates.
The main objective of this study is to investigate the effects of surface shape and Reynolds number on the heat transfer enhancement in turbulent flows. The all temperature measurements were carried out

at the initial streamwise distance Reynolds number (Re_x) of $3.1x10^6$ and $4.5x10^6$, compassing turbulent flows. The experimental set-ups, measuring equipment and methods, data reductions and uncertainty analysis are discussed in the following section and the experimental results are given in the third section, while the conclusion of the paper and the progressive studies are explained in the fourth and fifth sections, respectively.

II. EXPERIMENTAL TEST SET-UP

The presence of the surface curvature and ribs significantly changes the thermal characteristics of the flows due to the velocity field varies. The concave curvature and ribbed wall destabilized the flow, while the convex wall stabilized. The separations and reattachments over the ribbed wall increase fluid mixing, create flow unsteadiness, interrupt the development of the thermal boundary layer and enhance the heat transfer. In order to understand the heat transfer evolution of this complex flows, experiments were carried out in turbulent flows.

The experimental investigation carried out in a blowing-type, low-speed wind-tunnel, as shown in Fig. 1. The test section is 0.24x0.2 m^2 in cross section. A maximum free stream velocity of 30 m/s with a turbulence level of 0.7% can be attained. A constant temperature hot wire anemometer and copper-constant thermocouples were used to measure the velocity profile through the boundary layer and to obtain flow and wall temperatures, respectively.

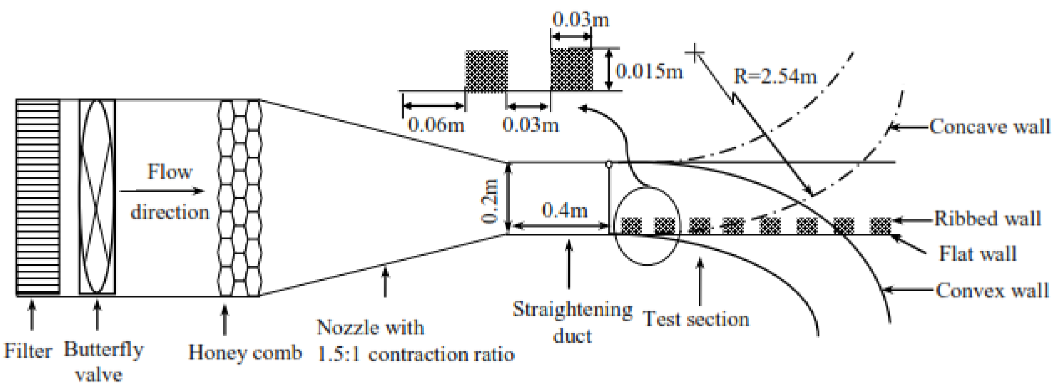

Figure 1. Wind tunnel and test section

Four different copper walls which have 0.24 m spanwise wide and 0.8 mm thick were used in the measurements. The streamwise distance (L) is 0.75 m in flat wall, 1.25 m in curved walls and 0.51 m in ribbed wall. The entire configuration and other dimensions are detailed in Fig. 1. The chrome-nickel resistive wires were furnished on the backsides of the all wall to heat the surfaces and obtain the near constant heat flux condition. The wire knitting was regulated with a variable AC voltage controller and coated with silicon layer. The silicon layer was covered with fiberglass to achieve minimum the heat loss. The thermocouples were embedded backside of the flat, concave and convex walls at x-locations of 0.06, 0.18, 0.30, 0.42, 0.54 and 0.66 m and at interval of 15 mm in the streamwise direction on the ribbed wall.

The boundary layers identified by shape factor ($H=\delta^*/\theta$), intermittency factor ($\gamma=(H-H_L)/(H_T-H_L)$), momentum thickness Reynolds numbers ($Re_\theta=U\theta/\nu$) and streamwise distance Reynolds numbers ($Re_x=Ux/\nu$) in the primarily flat plate measurements, where δ^*, θ, U, ν and x is displacement and momentum thickness, mean free stream velocity, kinematic viscosity and streamwise distance respectively, and subscripts L and T refer to laminar and turbulent flows. The heat transfer coefficient and experimental Nusselt number were calculated by $h=q/(T_w-T_0)$ and $Nu=hx/k$ respectively, where $q=q_f-q_o$, q_f and q_o indicate to flow-on and flow-off powers and T_w, T_0 and k refer to wall temperature, free stream temperature and thermal conductivity, respectively.

The maximum uncertainty in the Reynolds and Nusselt number calculation was estimated to be less than 3% and 4% respectively, using the uncertainty estimation method of Kline and McClintock [19].

III. RESULTS AND DISCUSSION

The flow measurements were carried out at the beginning of the flat plate to identify the inlet boundary layer. The shape factor, intermittency factor and momentum thickness Reynolds numbers changed from 1.32, 0.91 and 1900 to 1.30, 0.94 and 2100 respectively with increasing Reynolds number. It can be said from these results, the flow showed turbulent character at the both Reynolds number, in good harmony with Gostelaw et al. [20] results.

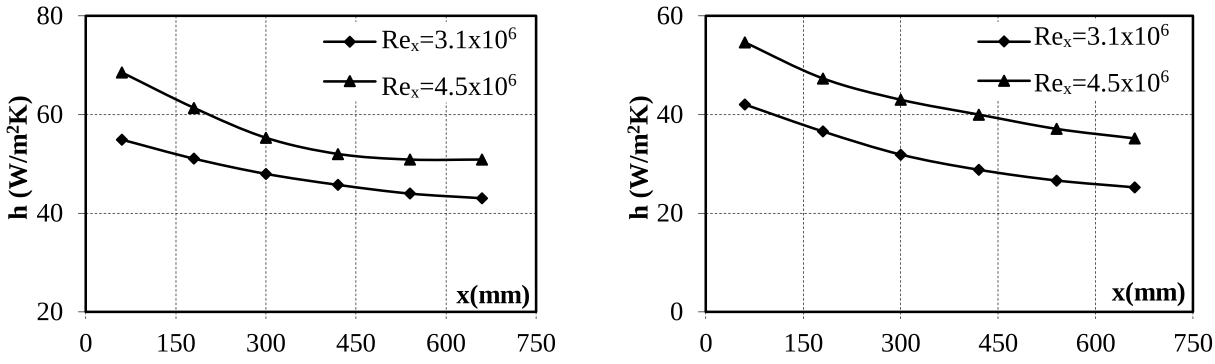

Figure 2. Streamwise variation of the heat transfer coefficient over the concave and convex wall

The streamwise variation of the convective heat transfer coefficients over the concave and convex wall are shown in Fig. 2 and 3, respectively. The heat transfer coefficients showed monotonically decreasing distributions in the streamwise direction and the h values increased with Reynolds number, as expected. The local heat transfer coefficients of the first and last measurement station over the concave wall at $Re_x=4.5x10^6$ were %20 and %15 bigger than those of the values at $Re_x=3.1x10^6$ respectively, while resulted in 26% higher average heat transfer coefficient over the convex wall in good accord with the results of Thomman [12].

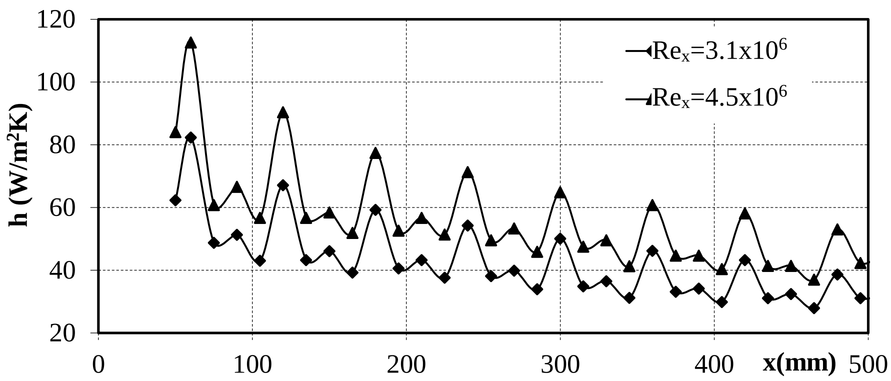

Figure 3. Streamwise variation of the heat transfer coefficient over the ribbed wall

The bigger Reynolds number caused %25 higher average heat transfer coefficient over the ribbed wall, as seen in Fig. 3. This increment can be explained by the augmentation in the turbulent level in the thermal boundary layer and the vortex with greater energy. Due to the strong accelerating and impact effects at the front and rear corners of each rib, the heat transfer coefficients appeared as large and small peaks there respectively, in accordance with Chen and Wang [6]. Besides, the presence of the separating bubbles between ribs caused small heat transfer coefficients which had no apparent contribution to the heat transfer from the ribs.

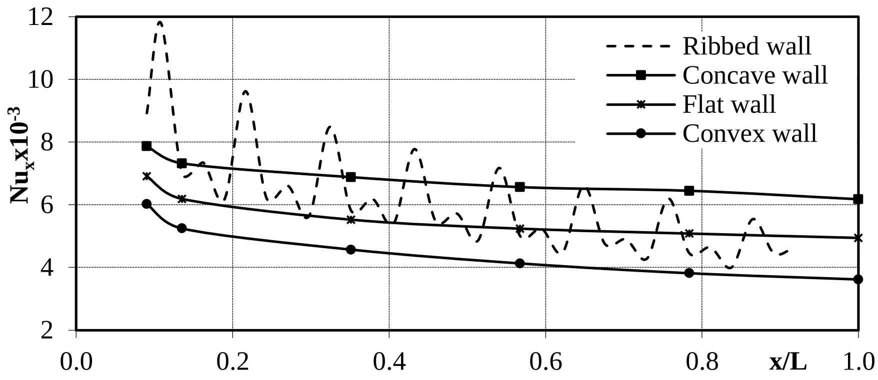

Figure 4. Local Nusselt numbers over the flat, concave, convex and ribbed walls at $Re_x=3.1x10^6$.

The Nusselt number distributions along the flat, concave, convex and ribbed walls for $Re=3.1x10^6$ are shown in Fig. 4. The concave curvature increased the average Nu_x values by 23% which is smaller than that of Umur [11] due to the thinner boundary layer and higher skin friction coefficient, while the thicker boundary layer thickness and lower skin friction on the convex wall caused Nu_x to decrease by 20%, comparing the flat plate values. The ribbed wall raised the average Nu_x by 28% due to higher fluid mixing and turbulence level, which is smaller than that of Perng and Wu [8]. The highest and the lowest Nusselt number was obtained on the beginning corner of the first rib and between the last ribs, respectively.

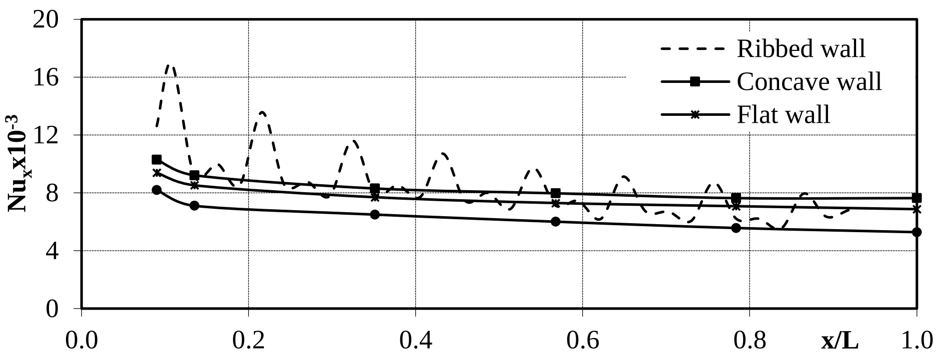

Figure 5. Local Nusselt numbers over the flat, concave, convex and ribbed walls at $Re_x=4.5x10^6$

The local Nusselt numbers over the flat, concave, convex and ribbed walls at $Re_x=4.5x10^6$ in the streamwise direction are given in Fig. 5. The Nusselt number curves of all walls showed similar distribution both at Re_x of $3.1x10^6$ and $4.5x10^6$. The concave wall enhanced the Nu_x values by 9-11% in streamwise direction comparing to those of flat plate, while the values decreased by 12-23% with the convex curvature. The local Nusselt numbers of the ribbed wall were bigger than those of the flat plate 80% and 10% on the first and last ribs. The average Nusselt numbers of the concave and ribbed wall were %9 and %22 higher than the flat plate values respectively, while those of the convex wall was 18% smaller that was similar to those of Mayle et al. [13].

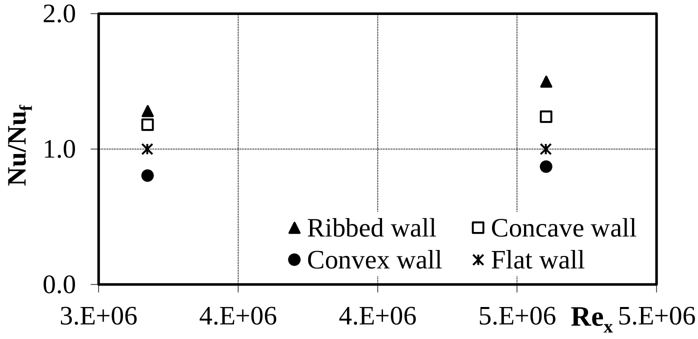

Figure 6. Variations of Nu/Nu_f with Re_x over the flat, concave, convex and ribbed walls.

The ratios of the average Nusselt number (Nu) of the concave, convex and ribbed walls to those of the flat wall (Nu_f) were presented in Fig. 6. The ratios on the ribbed, concave and convex walls were obtained as 1.28, 1.18 and 0.80 at Re_x of $3.1x10^6$ and 1.50, 1.24 and 0.87 at Re_x of $4.5x10^6$, respectively. The ratios of the ribbed wall are within the Yuan [21] findings.

IV. CONCLUSIONS

An experimental study was performed to understand the behaviour of the heat transfer characteristics over the concave, convex and ribbed walls in turbulent boundary layer under the effect of Reynolds number. The findings of the present works can be summarized as follows:
The concave curvature and presence of the ribs increased the turbulent heat transfer while the convex curvature decreased, comparing to the flat plate. The heat transfer enhancement ratios of the ribbed wall were bigger than those of the concave wall. The maximum Nusselt number values of the ribbed wall were obtained at the beginning corner of the first rib, while the minimum values were determined between the last two ribs. The heat transfer coefficients increased with Reynolds number for all the walls and the wall shape (concave, convex or ribbed) was more effective parameter than the Reynolds Number on the heat transfer. The average Nusselt numbers of the concave wall were higher than those of the flat plate by about 23% and 9% at Re_x of $3.1x10^6$ and $4.5x10^6$, respectively. The flat plate average Nusselt numbers were smaller than the concave wall values by nearly 28% at $Re_x=3.1x10^6$ and 22% at $Re_x=4.5x10^6$. The convex wall also decreased the Nu_x by 20% and 18% with increasing Re_x.

V. FUTURE WORK

In this paper the heat transfer characteristics of the concave, convex and ribbed surface are obtained in the turbulent flow. In the forthcoming work, the heat transfer and flow characteristics of the flow surfaces will be investigated together to better understand the complex flow structure. The experiments will be carried out both in laminar and turbulent flows and supported with the numerical analyses.

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