FLUID FLOW AND HEAT TRANSFER IN DUCT FAN FLOWS WITH TOP-WALL RECTANGULAR-WING TURBULATORS

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ABSTRACT

Fluid flow and heat transfer in duct fan flows with a 90° rectangular-wing turbulator, mounted on the top duct wall, were experimentally studied and compared with the bottom-wall turbulator results. Three-component velocities were measured to characterize the flow structures and to obtain near-wall flow parameters. Temperatures on heat transfer surfaces were measured to obtain Nusselt number distributions. Results show that the turbulator has the effect to increase the near-wall axial mean velocity, axial vorticity and turbulent kinetic energy, and, consequently, augment the heat transfer. The axial mean velocity and axial vorticity play an influential role on the heat transfer distributions for the flows across the top-wall and bottom-wall turbulators, respectively.

Keywords : Fan flow, Turbulator, Convective heat transfer.

1. INTRODUCTION

This paper presents results of a continuous study of rectangular-wing turbulator effects on heat transfer and flow characteristics in duct fan flows [1]. Electrical fans are commonly used in electronics, computers and so on as the pumping sources to induce swirling flow motions for heat transfer applications. The needs for faster heat transfer rates are urgent, for example, in CPU cooling in modern computers. Turbulators have been shown to have substantial effects on heat transfer rates in uniform flows. For example, Eibeck and Eaton [2] examined the heat transfer effects of a longitudinal vortex embedded in a turbulent boundary layer. Tiggelbeck *et al.* [3] studied the heat transfer enhancement and induced drag using wing-type turbulators in channel uniform flows. Biswas *et al.* [4] investigated the effects of delta-wing and winglet vortex generators on heat transfer in a channel flow. Ligrani *et al.* [5] studied the flow structure and local Nusselt number variations in a channel with a dimpled surface on one wall, both with and without protrusions on the opposite walls. Carvajal-Mariscal *et al.* [6] investigated the convective coefficient distribution on a pipe surface with inclined fins.

A fan flow involves swirling flow motions. The study of an external turbulator effect on fluid flow and heat transfer in such a flow has not been seen in the literatures. Recently, the author and his groups conducted a series of studies of wing-type turbulator effects on heat transfer and flow characteristics in duct fan flows [1,7-8]. The present research continues the study by a top-wall, rectangular-wing turbulator, and the results are compared with those obtained previously.

2. EXPERIMENTAL SETUP AND METHODS

The experimental setup utilized in this research is shown in Fig. 1. An axial DC fan is used to develop a fan flow inside a $7 \times 7\text{cm}^2$ duct. A rectangular-wing, 7cm in width, 3cm in height (H) and 90° angle of attack, is used as a turbulator, which is mounted on the top wall of the duct (referred to as the top-wall turbulator). Eight $4 \times 2\text{cm}^2$, 0.2mm thick aluminum plates were equally-spaced installed, 2cm apart, on the bottom wall of the test section between $X/H = 0$ and 10 to serve as independent heat transfer surfaces. The studies were performed under constant flow velocity and constant pumping power conditions. For the constant flow velocity tests, the DC power input to the fan was adjusted such that the duct average velocity (U_d) was 2m/s for the investigated flows, where the Reynolds number is 8900 based on the duct hydraulic diameter (D). For the constant pumping power tests, the DC power input to the fan for the flows across the turbulator are equal to that for the 2m/s flow without the turbulator. Since the insertion of the turbulator into the flow causes a pressure loss, the duct average velocity for the flow across the turbulator was reduced to 1.1m/s, where the Reynolds number is 4900.

Three-component, mean and fluctuating velocity measurements were performed using a laser Doppler

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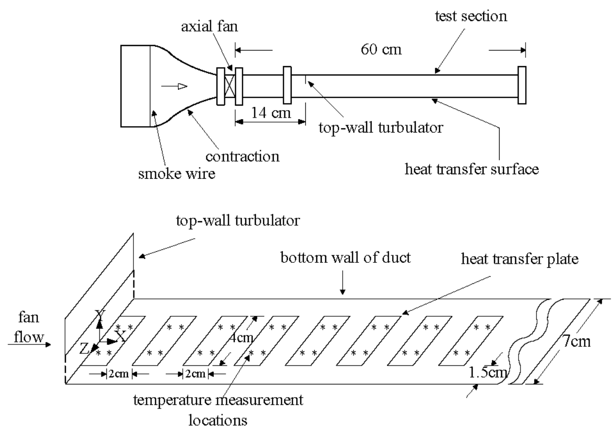

Fig. 1 A Schematic of the experimental setup

velocimetry. The velocity data were repeatedly checked using 4096, 8192 and even 16384 collected velocity samples. The obtained results showed that the uncertainty in the measured mean and fluctuating velocities was less than 5 percent, and the mean velocities do not change with time. Thus, 4096 data points are enough to represent the velocities of the investigated flows. The mean velocity ($\bar{v}$) of each velocity component ($\bar{U}$, $\bar{V}$, $\bar{W}$) at each measurement location is the average value of 4096 velocity samples. The fluctuating velocity of each velocity component (U', V', W') is obtained from $\sqrt{\sum v^2/4096-\bar{v}^2}$, where v is the local instant velocity. The heat transfer coefficient, $h = \dfrac{Q_{\text{in}}-Q_{\text{loss}}}{A(T_w-T_0)}$, was measured on the eight heat transfer surfaces. Only one heat transfer surface was heated during each run, with the power input, Q_{in}, of 2.2 ± 0.08 watts. The heat transfer surface area, A, is 8 ± 0.1cm^2. The temperatures were obtained using type-T, 0.1mm diameter thermocouples, with an uncertainty of ± 0.5°C. The reference temperature, T_0, was the air inlet temperature. Four thermocouples were spotted welded to the back surface of each heated aluminum plate, and the averaged temperature was used to represent that heat-transfer surface temperature, T_w. Heat conduction loss through the back and sides of the heat transfer plate was calculated using Fourier's law by knowing the temperatures drops across the thermally insulated plates [1]. Radiation loss was calculated using Stefan-Boltzmann law. The conduction and radiation losses were estimated to be less than 11 percent and 4 percent of the total power input, respectively. The total uncertainty in heat transfer coefficient was evaluated using the method suggested by Moffat [9], which was estimated to be 8 percent.

3. RESULTS AND DISCUSSION

It has been shown in Ref. [1] that the fan generates a nearly symmetrical, counterclockwise rotated, vortical flow inside the duct. The turbulator is expected to largely disturb the flow, especially in the immediate neighborhood behind the turbulator. Figure 2 presents the cross-sectional flow structures in X-Y planes at Z/D = 0.286 and Z/D = −0.286 as the flow across the turbulator. Due to the counterclockwise rotating flow, the flow is accelerated upward at Z/D = 0.286 and downward at Z/D = −0.286. It is noted that the turbulator is located at X/H = 0, Y/D = 0.572 to 1, Z/D = −0.5 to 0.5. As the flow develops downstream, the fluids at upper regions (large Y/D) are blocked by the turbulator and diverted through the lower regions (small Y/D). As a result, flow reversals and small flow velocities occur at the upper regions, and the flow acceleration occurs at the lower regions downstream of the turbulator to maintain the constant flow rate at each X/H station. Large velocity jumps occurred around the tip of the turbulator. The regions of flow reversal and small flow velocity are much larger at the plane of Z/D = −0.286 than the plane of Z/D = 0.286. The flow acceleration at the lower regions increases the convective effect near the heat transfer surface, which should have a positive effect on heat transfer rate, as discussed below.

Figure 3(a) presents the secondary flow velocities in the Y-Z plane at X/H = 0.333, showing the fan-generated symmetrical, counterclockwise rotated, vortical flow is completely disrupted by the turbulator. Since the fluids at the upper regions are blocked by the turbulator, the secondary flow velocities at the lower regions are, in general, increased. The turbulator effect on the secondary flow velocities becomes less as the flow develops downstream. Figure 3(b) shows that a single, counterclockwise rotated, vortical flow appears again at X/H = 3 though the vortex is distorted and unsymmetrical.

The velocity fields near a heat transfer surface should have major effects on heat transfer process. Three near-wall flow parameters, including the axial mean velocity, axial component of vorticity (axial vorticity) and turbulent kinetic energy, are determined. The velocity measurements were conducted at forty-five locations in each X/H plane between Y/D = 0.0286 and 0.1429 (2 to 10mm from the heat transfer surface), and between Z/D = −0.286 and 0.286 [1]. The obtained data were used to calculate the averaged axial mean velocity, axial vorticity and turbulent kinetic energy. More details on these calculations are included in Ref. [1]. The results of the fan flow and the fan flow with the bottom-wall turbulator [1] are also presented for comparison.

The axial mean velocity is an indication of the convective effect on heat transfer. The measured 45 near-wall axial mean velocities at each X/H station were averaged, and the obtained velocity, U_{av}, is used to represent the overall convective effect on heat transfer at that X/H station. It is noted from Fig. 2 that the fluids are blocked by the turbulator and the flow is accelerated at the lower regions. As a result, Fig. 4 shows that the top-wall turbulator increases U_{av} in the measured X/H

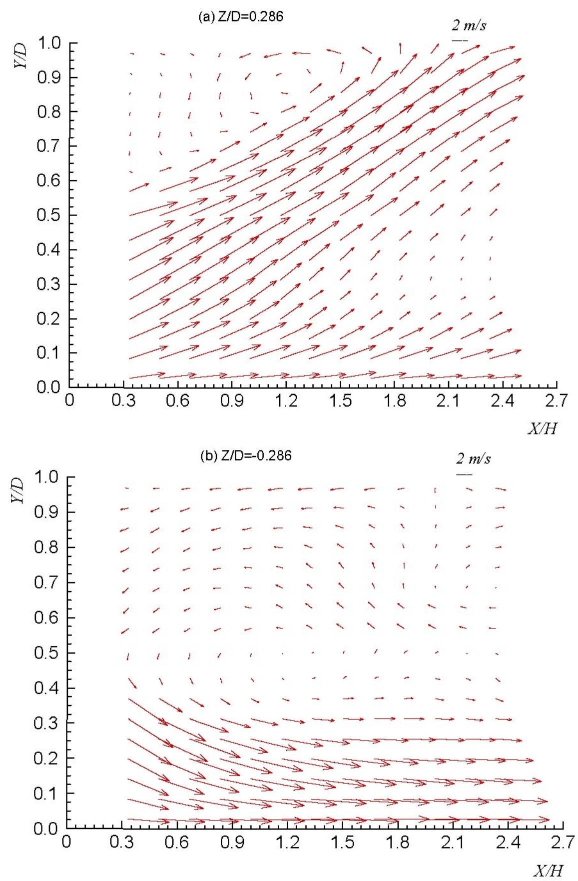

Fig. 2 The mean velocity vectors in the X-Y plane for the flow across the turbulator at (a) Z/D = 0.286, (b) $Z/D = -0.286$

ranges under constant flow velocity condition, and the largest velocity occurs at the measured X/H = 1.667. The flow with the top-wall turbulator apparently has a larger convective effect than the flows without turbulator and with the bottom-wall turbulator due to the flow acceleration described in Fig. 2. The U_{av}/U_d distribution under constant pumping power condition is qualitatively similar to that under constant flow velocity condition, but with smaller values due to the pressure loss caused by the turbulator. The U_{av} for the flow with turbulator under constant pumping power condition are still larger than those for the flow without turbulator at the stations of $X/H < 3.6$.

Large strength of the secondary flow should have a large effect on heat transfer. In this study, the circulation of a near-wall region (in Y-Z plane) between Y/D = 0.0286 and 0.1429, and between $Z/D = -0.286$ and 0.286 was calculated from the measured 45 Y- and 45 Z-component mean velocities [1]. The axial vorticity (η), obtained from the circulation dividing the area of the near-wall region, is used to denote the secondary-flow effect on heat transfer. Figure 5 presents the normalized axial vorticity distributions, $\eta D/U_d$ along X/H for the investigated flows. The axial vorticity is largely increased by the top-wall turbulator at

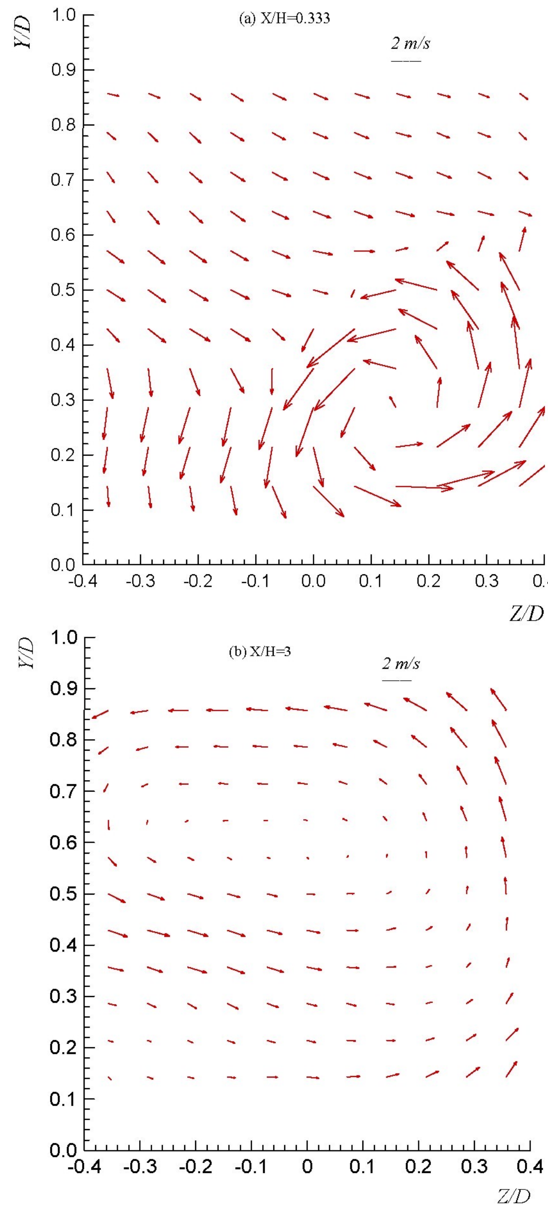

Fig. 3 The secondary velocity vectors in the Y-Z plane for the flow across the turbulator at (a) X/H = 0.333, (b) X/H = 3

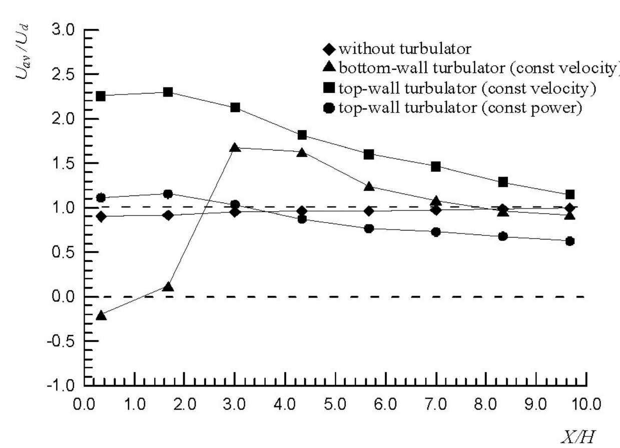

Fig. 4 The near-wall normalized averaged axial mean velocity distributions along X/H for the flows without and with turbulators

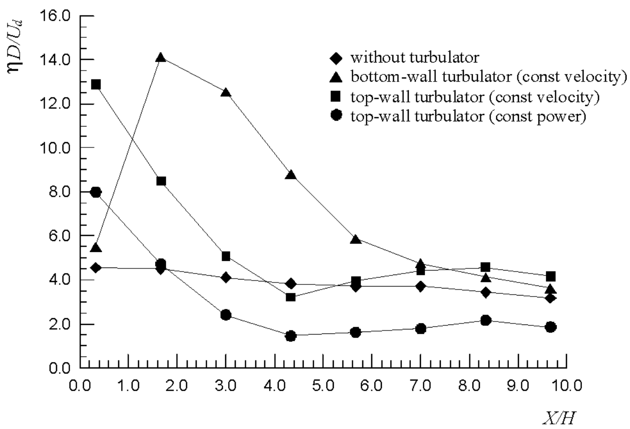

Fig. 5 The near-wall normalized axial velocity distributions along X/H for the flows without and with turbulators

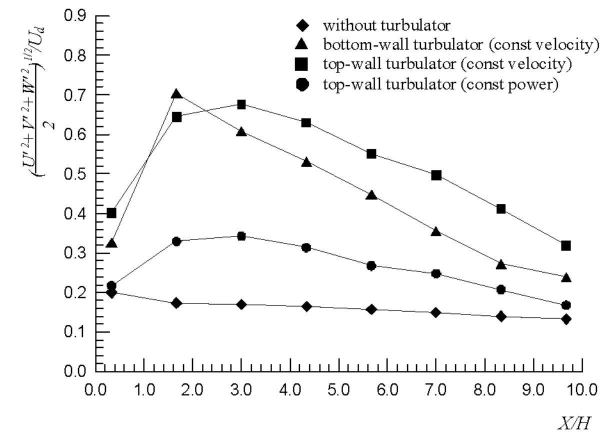

Fig. 6 The near-wall normalized turbulent kinetic energy distributions along X/H for the flows without and with turbulators

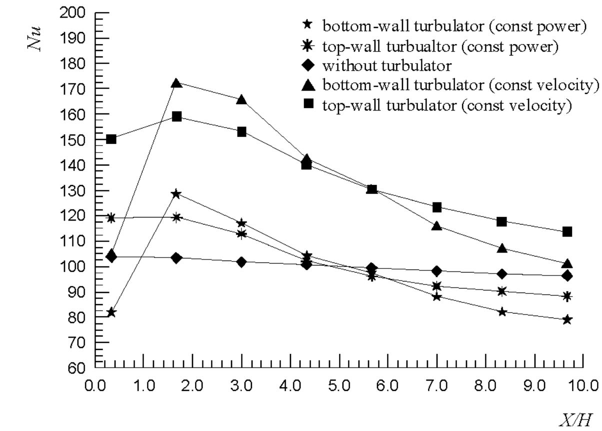

Fig. 7 The Nusselt number distributions along X/H for the flows without and with turbulators

$X/H = 0.333$. It has been shown in Fig. 3(a) that the secondary flow velocities near the heat transfer surface are largely increased at $X/H = 0.333$, and accordingly, the axial vorticity is largely increased. The axial vorticity distribution under constant pumping power condition is qualitatively similar to that under constant flow velocity condition, but with small values as expected. Figure 5 shows that the top-wall turbulator increases the axial vorticity in most of the investigated X/H ranges under constant flow velocity condition, and at the stations of $X/H < 1.667$ under constant pumping power condition. Also, the bottom-wall turbulator generally has a better effect on axial vorticity than the top-wall turbulator.

The turbulent kinetic energy, $(U'^2+V'^2+W'^2)/2$, is obtained from the near-wall fluctuating X-, Y- and Z-component velocities. Large turbulent kinetic energy (turbulence effect) is expected to have a large effect on heat transfer augmentation. Turbulators are generally used to increase the flow unsteadiness. Figure 6 presents the normalized turbulent kinetic energy distributions, showing that the top-wall turbulator increases the flow turbulent kinetic energy in all of the investigated X/H ranges under both constant flow velocity and constant pumping power conditions. In general, the top-wall turbulator has better effect on flow turbulent kinetic energy than the bottom-wall turbulator.

The heat transfer data are presented in a dimensionless form as the Nusselt number, Nu = hD/k, where k is the thermal conductivity of the air, evaluated at the film temperature. Figure 7 presents the effects of top-wall turbulator and bottom-wall turbulator (data were from Ref. [1] on heat transfer under constant flow velocity and constant pumping power conditions. Under constant flow velocity condition, the top-wall turbulator augments the heat transfer at all of the investigated X/H stations, and the largest heat transfer augmentation, up to 55 percent compared with the case of without turbulator, occurs at the measured $X/H = 1.667$. The augmentation becomes less as the flow develops downstream. This figure also shows that the bottom-wall turbulator causes higher local heat transfer augmentation than the top-wall turbulator, while the heat transfer augmentation by the top-wall turbulator persists further downstream than the bottom-wall turbulator. The overall heat transfer augmentation is obtained from the Nusselt number distributions. Results show that the bottom-wall and top-wall turbulators increase the Nusselt number in the investigated X/H ranges by 30 and 36 percents, respectively. The Nusselt number distribution under constant pumping power condition shows qualitatively similar to that under constant flow velocity condition. The turbulator effect on heat transfer augmentation persists approximately to 5 turbulator heights under constant pumping power condition.

It has been shown above that the turbulator have the effect to increase the near-wall averaged axial mean velocity, axial vorticity, turbulent kinetic energy, and to augment heat transfer in fan flows. Larger and smaller heat transfer augmentation occur in the vicinity of the measured $X/H = 1.667$ and 9.667, where the near-wall flow parameters are, in general, largely increased and less affected by the turbulator, respectively. The turbulators cause the decrease in Nusselt number at the stations beyond $X/H = 5$ under constant pumping power condition, where the averaged axial mean velocity and axial vorticity are decreased, and the turbulent kinetic energy is a little increased. It was shown in Figs. 4 and 7 that the averaged axial mean velocity distributions are very similar to the Nusselt number distributions for the flows across the top-wall turbulator. The maximum averaged axial mean velocity occurs at the measured X/H = 1.667, where the largest heat transfer augmentation is observed. The turbulent kinetic energy distribution is also similar to the Nusselt number distribution, as shown in Figs. 6 and 7. However, the Nusselt numbers for the flow across the top-wall turbulator at the stations of $X/H > 5$ are smaller than those for the flow without the turbulator under constant pumping power condition, where the turbulent kinetic energy is still increased. Figures 5 and 7 indicate that the axial vorticity distribution is different from the Nusselt number distribution. The maximum axial vorticity occurs at X/H = 0.333, where the Nusselt number is not the maximum. Also, the heat transfer is still largely augmented by the top-wall turbulator in the vicinity of X/H = 4.333 under constant flow velocity condition, where the axial vorticities are little affected. These results suggest that the convective effect plays a more influential role than the turbulence effect and the secondary-flow effect in determining the heat transfer distribution for the fan flow across the top-wall turbulator. The results shown in Chen and Chen [1] indicated that the secondary-flow effect is the main flow dynamic factor affecting the heat transfer distribution for the fan flow across the bottom-wall turbulator.

4. CONCLUSIONS

Results of this research indicate that the top-wall turbulator in fan flows has the effect to increase the near-wall flow axial mean velocity, axial vorticity and turbulent kinetic energy, and thus, augments heat transfer. The augmentation is more than 10 turbulator heights under constant flow velocity condition, and up to 5 turbulator heights under constant pumping power condition. The maximum heat transfer augmentation occurs in the neighborhood behind the turbulator. The convective effect plays an important role on heat transfer distribution for the flow across the top-wall turbulator. The top-wall turbulator in fan flows causes better convective and turbulence effects, but worse secondary-flow effect than the bottom-wall turbulator does. In overall, the top-wall turbulator has a comparable heat transfer augmentation as the bottom-wall turbulator in the investigated X/H ranges.

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