Numerical study on turbulent characteristics of wall jet in a quiescent environment over a plate in motion

V M Behera¹ and S K Rathore²
¹National Institute of Technology Rourkela, Odisha 769008, India
²National Institute of Technology Rourkela, Odisha 769008, India
Corresponding author E-mail: isushilrathore@gmail.com;

Abstract. The present article reports the numerical analysis of a two dimensional turbulent wall jet flow over a horizontal moving plate in a quiescent environment. The simulation is done using the low-Reynolds number turbulence model proposed by Yang and Shih [21]. The turbulent characteristics are observed for various velocity ratio of plate with respect to jet ($U_p$) from 0 – 2 keeping Reynolds number at jet inlet to be constant at 15000. The Reynolds averaged Navier Stokes (RANS) equations are discretized using the finite volume method and solved in steady state condition. The pressure correction technique applied to couple the velocity and pressure terms is SIMPLEC (semi-implicit pressure linked equation - consistent). The variation of jet half-width and the decaying local maximum axial velocity ($U_{max}$) for the bottom plate moving with high velocity ($U_p > 1$) shows dominancy of plate motion over the jet stream in the flow field. The velocity contour diagram shows the velocity distribution in the flow field. The Reynolds stresses induced in the wall proximate regions are directly proportional to the relative motion of the flow field but the magnitude of Reynolds stress away from wall decreases by increasing the plate velocity. The plate motion significantly affects the turbulence characteristics and the flow field in the viscous laminar zone of the wall region.

Keywords. turbulence, low-Reynolds model, wall jet, near wall region, plate velocity

1. Introduction
Turbulent jets have enormous area of applications that includes cooling of electronic devices, cooling in a continuous casting process and hot rolled sheets in steel industries, prevention of icing problem in aircraft at higher altitude [1-3] and so on. One such type of turbulent jet which has numerous applications is the wall jet. If the jet stream coming out of the nozzle flows tangentially over the flat surface, such that boundary layer effect is observed in the fluid layer at the wall, it is called as wall jet flow. Many experimental work have been performed in past to study turbulent wall jet flow behaviour. The flow parameters like jet half-width, local maximum axial velocity and Reynolds shear stresses variations are studied by Rostamy et al. [4] in an experiment and later by Tang et al. [5]. A large number of researches have been reported on turbulent impinging jet flow over moving surface. The plate motion strongly influences the mean velocity of the fluid stream along with the turbulent parameters like the Reynolds stresses and turbulence kinetic energy as reported by Chattopadhyaya

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and Saha [6]. Sharif and Banerjee [7] investigated the impinging jet on a moving plate by varying Reynolds number and found that skin friction coefficient does not respond to variation in Reynolds number however its magnitude appeared to be higher and more stable at lower plate velocity and is lower for higher plate velocity for a given Reynolds number. Kadiyala and Chattopadhyay [8] using transient SST model studied the slot jet impinging over moving plate case and found that plate motion has significantly influenced the flow field as well as the heat transfer rate rather than the Reynolds number.

Numerous computational work with various types of turbulence models such as - Anderson and Spall [9] conducted a numerical investigation using standard k-ε model along with Reynolds stress model, SST model by Shivankar et al. [10], standard k-ε turbulence model by Vishnuvardhanrao and Das [11], Singh et al. [12] worked with wall jet on wavy surface, and so on has been reported. The high Reynolds number turbulent models use wall functions in wall proximity regions and perform poorly in case of flow separation regions. To overcome these drawbacks some researchers developed low-Reynolds number (LRN) turbulence model by modifying the k-ε two-equation. Later, Kechiche et al [13] investigated turbulent wall jet flow by applying LRN turbulence models of Herrero et al. [14], Nagano and Hishida [15], and Chien [16] and concluded that LRN models can predict the turbulence parameters in near wall regions accurately. Rathore and Das [17-19] worked on turbulent offset and wall jet flow using different turbulence models like standard k-ε, two LRN models (namely LS model [20] and the YS model [21]) and later on included k-ω SST model and concluded that YS model could readily capture the flow field parameters in the viscous sub-layer region very accurately.

The above literature reveals that many investigations have been performed in past years on turbulent wall jets using various turbulent models. But so far no investigation on turbulent wall jet flow over a moving plate was reported. This paper aims at characterizing the turbulent wall jet flow on a moving plate.

2. Mathematical formulation

2.1. Problem definition

A two-dimensional, steady incompressible wall jet stream in a quiescent atmosphere is considered as shown in Fig. 1. The fluid properties are taken constant and effect of gravity is neglected. The jet inlet velocity is non-dimensionally taken as $U_0 = 1$, and the inlet Reynolds number is 15000. The plate is moving with a constant velocity which can be expressed in the non-dimensional form ($U_p$) as the ratio of plate to jet velocity. The turbulent flow behavior in the near wall regions are observed for the moving plate and the stationary plate.

![Figure 1. Layout of wall jet on a moving wall](image-url)
2.2. Numerical methodology
The RANS equations are discretized using FVM. The diffusion terms in RANS equations are
discretized by central differential scheme whereas for convection terms, power law scheme is applied.
The mean velocity gradients and turbulent Reynolds stresses are related by applying Boussinesq
model. The eddy viscosity equation as given by Kolmogorov-Prandtl used expresses the turbulent
viscosity ($\nu_t$) in terms of turbulent kinetic energy ($k_n$) and rate of dissipation ($\epsilon_n$). The pressure
and velocity terms are related by adopting SIMPLEC (semi-implicit pressure linked equation - consistent)
by Doormaal and Raithby [22]. Algebraic equations are solved by applying Tri-diagonal matrix
algorithm by Malalasekhar [23].

2.3. Governing equations
The equations used are given in a dimensionless form:

**Continuity equation:**
$$\frac{\partial U_i}{\partial X_i} = 0 \quad (1)$$

**Momentum equation:**
$$U_j \frac{\partial U_i}{\partial X_j} = -\frac{\partial (P + \frac{\nu_t^2}{\rho})}{\partial X_i} + \frac{1}{2} \frac{\partial}{\partial X_j} \left[ (1 + \frac{\nu_t}{\nu}) \left( \frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i} \right) \right] \quad (2)$$

**Turbulent Kinetic energy equation:**
$$U_j \frac{\partial k_n}{\partial X_j} = \frac{1}{Re} \frac{\partial}{\partial X_j} \left[ (1 + \frac{\nu_t}{\sigma_k}) \frac{\partial k_n}{\partial X_j} \right] + G_n - \epsilon_n \quad (3)$$

**Rate of Dissipation equation:**
$$U_j \frac{\partial \epsilon_n}{\partial X_j} = \frac{1}{Re} \frac{\partial}{\partial X_j} \left[ (1 + \frac{\nu_t}{\sigma_k}) \frac{\partial \epsilon_n}{\partial X_j} \right] + G_n \frac{C_{1f}}{k_n} \frac{\epsilon_n}{k_n} - C_{2f} \frac{\epsilon_n}{k_n}^2 \frac{\epsilon_n^2}{k_n} + E_n \quad (4)$$

Where, $E_n = (2\nu \frac{v_n}{Re^2})(\frac{\partial^2 U_i}{\partial X_j^2})$

**Production of shear:**
$$G_n = \frac{\nu_t}{Re} \frac{\partial^2 U_i}{\partial X_j^2} \frac{\partial U_i}{\partial X_j}$$

**Eddy viscosity:**
$$\nu_t = C_{\mu}f_{\mu}Re \frac{k_n^2}{\epsilon_n} \quad (5)$$

**YS model functions:**
$$f_1 = (1 + 1/Ret) \left[ 1 - \exp \left( -aRe - bRe^3 - cRe^5 \right) \right]$$
$$f = f_2 = \sqrt(Re \left( 1 + Re \right))$$

Where, $a = 1.5 \times 10^{-4}$, $b = 5.0 \times 10^{-7}$, $c = 1 \times 10^{-10}$.

2.4. Boundary conditions
The jet inlet velocity in the non-dimensional form is taken as $U = 1$, $V = 0$ and the turbulent kinetic
energy at jet inlet, $k_n = 1.5I^2$ where $I$ is the turbulent intensity taken to be 0.05 and the dissipation
rate, $\epsilon_n = (k_n^{3/2}C_{\mu}^{3/4})/0.07$. No-slip condition of $U = U_p$ where $U_p = u_p/U_0$ and $V = 0$
is imposed on horizontal moving plate surface at the bottom of fluid domain and. At the vertical wall,$U = 0$ and $V = 0$ is imposed as the boundary condition. At walls, $k_n = 0$ whereas rate of dissipation
is given as $\epsilon_n = \frac{2}{Re} \left( \frac{\partial^2 U_i}{\partial n^2} \right)^2$, where $n$ represents dimensionless coordinate in normal direction. At the
entrainment side (top side) of the domain, Neumann boundary condition ($\frac{\partial \Phi}{\partial n} = 0$) is imposed where,$\Phi = U, V, k_n, \epsilon_n$ and at the right side exit boundary out flow condition is set which is expressed
as $\frac{\partial \Phi}{\partial X} = 0$. 
3. Grid independent test and validation

A non-uniform grid cluster is formed such that highly dense smaller size grid is generated within the jet inlet region as well as in the near wall area and as moved away the grid size increases. The first grid size is chosen such that, the $y^+ \leq 1.5$ in all cases. The fluid domain size is $90 \times 50$. The grid sizes considered are $221 \times 221, 313 \times 313$ and $441 \times 441$. From Fig 2, it is found that the grid size $313 \times 313$ would be the most suitable grid size from the computational cost view. The variation of $u^+ - y^+$ in the stationary plate case is validated with experimental result of Nizou and Tida [24]. The $u^+ - y^+$ profile is in good agreement in near wall region for $y^+ \leq 11$ but it deviates from the experimental result as moved away in logarithmic region.

![Figure 2. Validation with the experimental result (Nizou and Tida [24])](image)

![Figure 3. Zoomed view of U-component velocity contour in flow field](image)
4. Results and discussion

4.1. Velocity contour diagram

The velocity contours are shown for $U$ – component of velocity, in $X$ – direction for stationary and moving plate with various plate to jet velocity ratios ($U_p = 1, 1.5, \text{ and } 2$) in Fig. 3. The enlarged view of velocity contour illustrates the effect of moving plate in $X$ – direction. It can be observed from the contour diagram that $U$ – component of velocity is maximum at the jet inlet in the stationary plate case whereas for the moving plate the maximum velocity is identified at the surface for the moving plate with $U_p = 1.5, \text{ and } 2$.

![Velocity contour diagram](image)

Figure 4. Axial variation of jet half-width for various cases of plate velocity

4.2. Spreading of jet half-width ($Y_{0.5}$)

Spreading of jet can be traced by jet half-width ($Y_{0.5}$) which gives the location of local velocity in vertical direction where, $U = U_{\text{max}}/2$. Figure 4, represents the variation $Y_{0.5}$ in the axial direction. It is observed that in a stationary plate, the jet half-width increase monotonically along the downstream. But for the cases of higher plate velocity where, $U_p \geq 1$, different patterns are observed. In case of moving plate ($U_p = 1$), the jet half-width increases to a peak point and then it decreases slowly at a distance far away in X-direction. For $U_p = 1$, the jet half-width decreases gradually in axial direction to a minimum point after which the rate of decrement becomes very slow. Whereas, for the moving plate ($U_p = 2$), the $Y_{0.5}$ shows a sudden fall near jet inlet. It then increases to a peak point and
decreases in axial direction. The combined effect of plate velocity and jet velocity decides the profile of axial spreading of jet.

4.3. Distribution of local maximum velocity ($U_{\text{max}}$) in axial direction

The variation of ($U_{\text{max}}$) velocity in axial direction is shown in Fig. 5. For a stationary plate the velocity possessed at the jet inlet is highest but on moving away from the inlet in axial direction, the velocity decreases gradually due to interaction with the surrounding fluid. But for higher plate velocity cases where $U_p > 0.5$ the velocity traced in the profile is same as that of plate velocity. It is because velocity of fluid is highest at the plate surface due to no-slip condition imposed on the fluid layer in direct contact with the plate surface. Hence, for the plate to jet velocity ratio ($U_p \geq 1$), the magnitude of velocity possessed by the fluid layer at the moving plate surface is maximum. It is observed further that the profile remains invariant while moving away in axial direction.

![Figure 5. Axial variation of local maximum velocity](image)

![Figure 6. Reynolds shear stress variation](image)

4.4. Distribution of turbulent Reynolds shear stress

The variation of Reynolds shear stress ($\overline{u'v'}/U_0^2$) along $Y$ axis at axial location of $X = 30$ is shown in Fig. 6. At walls, the fluctuating velocity components ceases and hence the Reynolds shear stress ($\overline{u'v'}/U_0^2$) gets zero. A negative stress is developed near the plate surface in case of stationary plate. A similar negative profile can be seen for plate moving with $U_p = 0.5$ for the axial positions. It then increases and changes sign as moved away from the wall in $Y-$ direction, and becomes maximum in the region of local maximum mean velocity. Whereas, for the other cases of moving plate $U_p = 1, 1.5$ and $2$, the Reynolds shear stress developed has a positive profile closer to wall which follows similar trend as that of stationary plate when moved away in vertical direction. When moved away towards the upper free shear layer, the stresses decreases and becomes minimum above the upper shear layer.

5. Conclusions

The turbulent flow characteristics for a wall jet stream flowing over a flat plate moving in same direction as that of jet stream have been investigated numerically using YS turbulence model proposed by Yang and Shih. The consequences of moving plate on the flow field are observed from the variation of velocity contours, jet half-width, local maximum velocity and distribution of Reynolds shear stress.

From the velocity contour, jet half-width ($Y_{0.5}$) and the maximum local velocity diagram ($U_{\text{max}}$) it was found that the maximum velocity is developed in the flow field in the fluid layer next to the
moving wall in case of higher plate to jet velocity ratio cases. It is because of the no slip condition of $U = U_p$ imposed on the fluid layer next to the solid moving plate. The jet half width diverges away from the wall in a stationary plate and for plate moving with lower velocity ratio. Whereas for higher plate velocity cases ($U_p > 1$), the jet half-width appears closer to wall which indicates that fluid speed is maximum at the wall. This indicates that flow field is greatly influenced by the velocity of the horizontal moving plate.

The Reynolds shear stress profile suggests that it is dominant in the flow field away from wall and in the flow region closer to wall surface it gets weaker and is zero at the wall for all cases. However, the Reynolds shear stress profile appears to possess a positive value closer to wall region in a moving plate. The magnitude of Reynolds shear stress decreases with increase in the velocity ratio($U_p$). It may be because in stationary plate case and for $U_p = 0.5$, the mean velocity is maximum in the flow field away from the wall as verified by the jet half-width curve but for the cases of $U_p \geq 1$ the maximum velocity lies at the plate surface due to the no-slip condition hence the magnitude of $(\overline{u'\nu'/U_p^2})$ decreases with the increase in velocity of plate. As the Reynolds shear stress is weaker within the thin viscous sub-layer region, the opposing shear force developed in high magnitude velocity field in this region is less in magnitude.

It is concluded from the observation of the jet half-width, local maximum velocity distribution and the turbulent Reynolds shear stress distribution that the flow field and turbulent characteristics are dominated by motion of the plate.

ACKNOWLEDGEMENT

We gratefully acknowledge Science and Engineering Research Board (SERB), Department of Science and Technology, Government of India, for providing financial support to our research.

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