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Directed co-flow effects on local entropy generation in turbulent heated round jets

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ABSTRACT

In the present paper, we have studied numerically the directed co-flow effects on local entropy generation rate in turbulent and heated round jets. The first order closure model was used and compared to existing experimental findings. The numerical results of the mean and turbulent quantities of the entropy generation rate and of the Merit number have been presented and discussed. The results obtained show that the directed co-flow with a positive angle enhances the mixing. Consequently, the local entropy generation rate increases progressively with the deviation angle.

1. Introduction

The turbulent jets are widely used in various industrial applications for mixing two streams of fluid such as in combustion process, where a fuel jet flow is commonly injected into a co-flowing stream of air. In many combustion devices, a slow co-flow or a swirling flow is used to stabilize the flame through a recirculation zone [1]. Moreover, the heat transfer characteristics in the jet flow change considerably once the co-flowing stream air is introduced. The entropy generation in the system enables to identify the friction losses and heat transfer rates in the system [2,3]. Consequently, investigating flow and heat transfer characteristics as well as investigating the entropy generation in co-flowing jet flow is essential.

Considerable research studies were carried out to explore the effect of the nozzle exit of the jet and their surrounding characteristic in the entropy generation rate. The recent work of Lucia [4] has analysed the maximum and the minimum of entropy generation in open systems. It has suggested that there are two approaches which are interrelated: The first is associated to the system, while the second is related to the interaction between the system and the environment. The entropy generation of a fully developed laminar flow in a hexagonal duct has been investigated by Jarungthammachote [5]. In addition, he has been compared the entropy generation found by the hexagonal duct with that found by rectangular and circular ducts as well. Cervantes and Solorio [6] have investigated the entropy production with and without oscillations in plane turbulent jet. Their results have confirmed that the entropy generation grows along the flow direction and depend directly on the entrainment of the still ambient fluid. Shuja et al. [7] have studied a confined laminar swirling jet and entropy generation with different jet exit velocity profiles. They have found that the swirling expands the jet in the radial direction and reduces the jet length in the axial direction. Accordingly, when the swirl velocity increases the entropy generation rate decreases. Moreover, Shuja et al. [8] have examined the effects of the outer cone angle of the annular nozzle on the entropy generation in an impinging jet. They have come to a conclusion that the volumetric entropy generation rate increases at the nozzle exit due to the velocity distribution of the flow emerging from the nozzle. Chu and Liu [9] have analysed the entropy generation in a two-dimensional high-temperature confined jet flow. They observed that the total entropy generation number decreases when the jet Reynolds number and Boltzmann number increase. Yapici et al. [10] have investigated the local entropy generation in a compressible flow through the expanding pipe for different expansion ratios. Results show that contraction of the radius of the throat increases the volumetric entropy generation rate maximum value. Gazzah and Belmabrouk [11], have studied the effects of a co-flow and inlet jet temperature on local entropy generation in turbulent round jets. Their results show that the total entropy generation rate decreases when the co-flowing jet increases and when the inlet hot jet temperature decreases.

The effect of turbulence model on entropy generation has attracted attention of many researchers. Shuja et al. [12] have

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Nomenclature

D	nozzle diameter (m)
Γ_Φ	diffusion coefficient ($m^2\,s^{-1}$)
$\dot{I}$	rate of total irreversibility (W)
k	turbulent kinetic energy ($m^2\,s^{-2}$)
L_T	temperature halfwidth (m)
L_U	velocity halfwidth (m)
M	Merit number
$\dot{Q}$	rate of exergy transfer (W)
r	radial distance (m)
σ_Φ	turbulent Schmidt number
$\dot{S}_{gen}$	entropy generation rate ($W\,K^{-1}$)
T	mean temperature (K)
u	fluctuation velocity ($m\,s^{-1}$)
U	mean axial velocity ($m\,s^{-1}$)
x	axial distance (m)
Greek	
α	deviation angle (°)
β	volumetric expansion coefficient (K^{-1})
μ	dynamic viscosity ($kg\,m^{-1}\,s^{-1}$)
ε	dissipation rate of turbulent kinetic energy ($m^2\,s^{-3}$)
ε_θ	temperature dissipation rate ($K^2\,s^{-1}$)
θ	temperature fluctuation (K)
θ^2	temperature variance (K^2)
ρ	density ($kg\,m^{-3}$)
Φ	generalized turbulent parameter
Subscript	
a	ambient fluid
c	centerline
co	coflow
j	jet fluid

evaluated numerically the effect of turbulence models on the local entropy generation rate in an impinging jet. Their results showed that, close to the stagnation region, the low-Reynolds number $k-\varepsilon$ model predicts a smaller value of the volumetric entropy generation than Reynolds Stress Models (RSTM). In addition, Ghasemi et al. [13] investigated the entropy generation process in the bypass transition for a flat plate boundary layer. In their study, the Reynolds-Averaged Navier–Stokes (RANS) models and Direct Numerical Simulations (DNS) are implemented and all the RANS models overpredict the integral entropy generation rate.

In order to predict the effects of directed co-flow on local entropy generation in turbulent heated round jet, we have used first order closure model based on three transport equations: the turbulence kinetic energy equation, the kinetic energy dissipation rate equation and the temperature variance equation. We have also used the Finite Volume Method to solve numerically the differential equations. First, we will compare our numerical findings with Djeridane experimental results [14]. Then, we will study a turbulent heated round jet into co-flowing air in various directions between −10° and +10°. Moreover, we will analyze the influence of the directed co-flow on various physical parameters such as the mean and the turbulent quantities, entropy generation rate and Merit number.

2. Turbulence model

The turbulence flow is modelled using Favre-averaged quantities with $k-\varepsilon-\theta^2$ turbulence model. The Favre-averaged quantity is denoted by $\widetilde{\Phi}$ and is defined as

$$\Phi = \widetilde{\Phi} + \Phi'' \quad \text{with} \quad \bar{\rho}\widetilde{\Phi} = \overline{\rho\Phi} \tag{1}$$

where the bar $\overline{()}$ denotes the Reynolds average. All variables are Favre averaged except the pressure and density that are conventionally time averaged and denoted by $\bar{p}$ and $\bar{\rho}$ respectively.

The governing equations constitute a set of coupled partial differential equations that can be written in the cylindrical coordinate system in general form as:

$$\frac{\partial}{\partial x}(\rho U\Phi)+\frac{1}{r}\frac{\partial}{\partial r}(r\rho V\Phi)=\frac{\partial}{\partial x}\left(\Gamma_\Phi\frac{\partial\Phi}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\Gamma_\Phi\frac{\partial\Phi}{\partial r}\right)+S_\Phi \tag{2}$$

where Φ, Γ_Φ and S_Φ are respectively, the transportable quantity, the diffusion coefficient and the source term and are presented in Table 1.

2.1. First order closure turbulence model

The turbulence flow is modelled by $k-\varepsilon-\theta^2$ turbulence closure model using Favre-averaged quantities. A turbulent viscosity μ_t is used to relate the Reynolds stress $(\overline{\rho u_i u_j})$ to the average velocity gradient. In the same way, the fluxes of the turbulent temperature $(\overline{\rho u_i \theta})$ can be connected to the average gradients of the temperature using the suitable diffusion coefficients. In this model, the turbulent viscosity μ_t is given by

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{3}$$

where C_μ is a constant of the model.

The three transport equations of the model are written as the same form as Eq. (2). Table 2 shows the diffusion coefficients and source terms related to the variables k, ε and θ^2.

The constants used in this work are shown in Table 3.

The effective viscosity is

$$\mu_{eff} = \mu + \mu_t \tag{4}$$

where μ and μ_t are the molecular and turbulent viscosities, respectively.

Here P_k is the production term of turbulent energy which reads

$$P_k = \mu_t\left(2\left[\left(\frac{\partial U}{\partial x}\right)^2+\left(\frac{\partial V}{\partial r}\right)^2+\left(\frac{V}{r}\right)^2\right]+\left(\frac{\partial U}{\partial r}+\frac{\partial V}{\partial x}\right)^2\right) - \frac{2}{3}\mu_t\left(\frac{1}{r}\frac{\partial}{\partial r}(rV)+\frac{\partial U}{\partial x}\right)^2 \tag{5}$$

and P_θ is the production term of temperature fluctuations variance.

Table 1
Terms of Γ_Φ and S_Φ in the conservation equations.

Φ	Γ_Φ	S_Φ
1	0	0
U	μ_{eff}	$-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial U}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_{eff}\frac{\partial U}{\partial x}\right)$
V	μ_{eff}	$-\frac{\partial p}{\partial r}+\frac{\partial}{\partial x}\left(\mu_{eff}\frac{\partial V}{\partial x}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_{eff}\frac{\partial V}{\partial x}\right)-2\mu_{eff}\frac{V}{r^2}$
T	μ_{eff}/σ_t	0