A HEATED SWIRLING JET EMANATING FROM A FULLY DEVELOPED TURBULENT PIPE FLOW

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ABSTRACT

The flow field of a heated swirling jet emanating from a fully developed rotating pipe flow has been experimentally investigated in order to analyse the effect of rotation on the spreading of a passive scalar. Measurements of the instantaneous axial and azimuthal velocity as well as the temperature were performed simultaneously with a combined X–wire and cold–wire probe.

***K*eywords:** *Heated jet, swirling jet, rotating pipe flow, hot–wire/cold–wire sensor*

1. INTRODUCTION

The swirling jets are used in many engineering applications. Among other things because it is known that rotating jets spread and mix with the ambient fluid faster than the non-swirling ones. Rotation is also used to increase heat transfer in impingement cooling.

Despite of the importance of this type of flows, there are only few theoretical and numerical analysis which can be used in practical application [2], [3]. The limitations are due to the high Reynolds number required for Direct Numerical Simulations (DNS) and due to the lack of reliable turbulence models for Reynolds Averaged Navier-Stokes equations (RANS). Therefore there is still the need of an experimental analysis, both to provide direct informations on the physics of these flows and to create a data-base that can be used to validate the numerical procedures and calibrate turbulence models.

Many investigations have been performed in different laboratories (see the Review in [1]). However comparisons between these studies are hindered due to the fact that the swirl profile depends strongly on how the swirl is generated. This can can be done with e.g. vanes, a rotating honeycomb, azimuthal injection of secondary flow upstream of the jet exit etc. All of these methods leave traces in the flow and make impossible any comparison or general conclusions.

For this reason a special experimental set-up where a swirling jet emanates from a long (100 diameters), axially rotating pipe, has been developed. This gives both well defined streamwise and azimuthal velocity profiles at the pipe exit making possible, under well-controlled conditions, to investigate the characteristics and the dynamics of the jet with and without rotation.

Preliminary studies have been performed on a cold jet to assess different characteristics, viz. axial decay, spreading, turbulence statististics. In this paper the set-up is used to study the effect of rotation on the spreading of a heated jet obtained by warming up the air above the ambient temperature before it enters the pipe. The analysis is limited to small differencies so that the density can be considered constant. Thus, the temperature acts as a tracer which allows the turbulent mixing of the jet to be studied.

A fully developed axially rotating pipe flow is defined through two parameters, the Reynolds number

$$Re = \frac{U_b\,2R}{\nu} \qquad (1)$$

and the swirl number

$$S = \frac{V_w}{U_b} \qquad (2)$$

where U_b is the bulk velocity, R the pipe radius, ν the kinematic viscosity and V_w the azimuthal velocity of the rotating pipe. S=0 corresponds to the non-rotating case. In this first experimental analysis S is limited to S=0 and S=0.5 well below S=0.6-0.7 where it is known that the jet may be subjected to a phenomenon called vortex breakdown and where it is not possible to apply the measurement techniques adopted in this study.

The Reynolds number for the measurements presented here is approximately $Re = 24000$. Previous tests [1] show that only small changes in the flow field can be observed by varying this parameter.

In this paper, axial and azimuthal velocity profiles as well as temperature profiles along the radius of the jet are measured in the near exit region of the pipe from $x/D = 0$ to 6 pipe diameters from the exit.

2. EXPERIMENTAL SET-UP

2.1 Rotating pipe apparatus

The experiments are performed at the laboratories of KTH-Mechanics in a specially designed set-up, consisting of a 6 meter long axially rotating pipe (see figure 1). The air is provided by a centrifugal fan (A), downstream of which a flow meter (B) monitors the flow rate. After the flow meter a flow distribution chamber (D) distributes the flow into three different pipes, which are symmetrically fed into the stagnation chamber (E). A bell mouth shaped entrance first feeds it into a one meter long stationary section, which is connected to the rotating pipe (K) through a rotating coupling (F). In the first part of the rotating part of the pipe a honeycomb (G) is mounted which brings the flow into more or less solid body rotation. Thereafter the flow develops along the 6 meter long pipe before it emanates as a free jet (M). The pipe

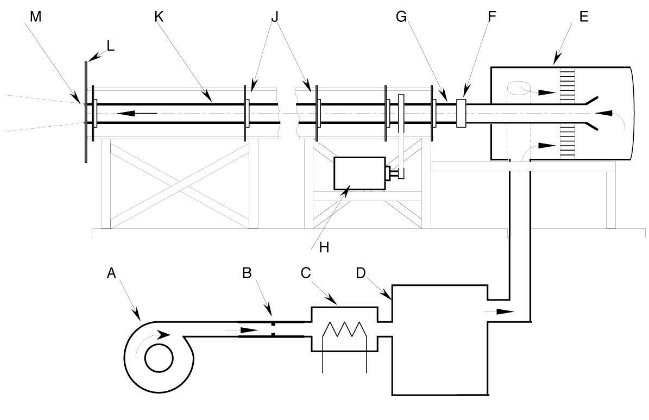

Figure 1: Experimental set-up

is made of seamless steel and has a honed inner surface. It is supported along its full length by 5 roller bearings (J), which are mounted within a rigid triangular shaped framework, and it is belt driven via a feed back controlled DC motor (H). In the present measurements the pipe ends with a 12 cm diameter circular end plate (L). A heater (C) placed in the flow upstream of the distribution chamber provides the heating of the air. The heater power can be regulated and is typically around 1000 W. The outer pipe surface is insulated with a 15 mm foam material in order to establish a constant radial temperature. Typical temperature difference between the flow in the pipe and the ambient air is T=10 K.

2.2 Measurement technique

In order to get simoultaneous acquisition of velocity and temperature, a specific home made probe has been designed and built. Two X-wires, operated in the constant temperature (CTA) mode with an overheat ratio of 30%, are fixed on the probe to measure the two in-plane velocity components while a cold wire, operated in the constant current (CCA) mode with a constant current of 0.3 mA, is positioned on the same support for temperature measurements. The position of the cold wire was chosen in order to minimize the thermal and wake interference. Platinum wire with a diameter of 1.27 μm is used for all sensors. The velocity sensors are 0.8 mm long while the temperature sensor is 1.1 mm.

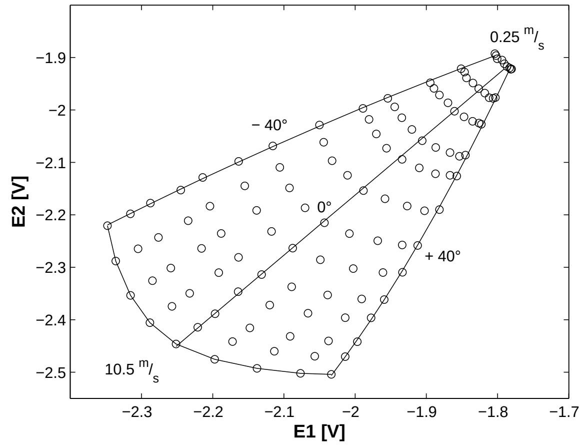

Figure 2: Example of calibration plot for the X-wire probe

Since the probe cannot be calibrated in situ a special designed calibration facility was used. The probe was positioned in the potential core of a contraction jet and calibrated, at a constant temperature, against a prandtl tube for velocities between 0.25 m/s and 10.5 m/s and different flow angles (see figure 2). Conversely, the temperature calibration took place in the axis at the pipe exit and the cold wire was calibrated against an thermocouple with a measurement resolution of 0.1 K. Due to the small diameter and the low current through the cold wire the anemometer output becomes a linear function of the fluid temperature.

While the fluctuating temperature can be measured directly by the cold wire, the voltage output for the hot-wires must be corrected against changes in fluid temperature as well as any drift in the ambient temperature. The hot-wire response was compensated for temperature changes through the well known relation

$$E_{out}(T_0) = E_{out}(T)\sqrt{\frac{T_{hot} - T_0}{T_{hot} - T}}\,. \tag{3}$$

Here $E_{out}(T_0)$ is the hot-wire response obtained for the same velocity under isothermal ($T = T_0$) conditions and $E_{out}(T)$ is the measured response from the anemometer output. Here the wire temperature T_{hot} is estimated by using the applied overheat ratio a and the temperature coefficient of the electrical resistivity α through the following relation

$$T_{hot} = T_{ref} + \frac{a}{\alpha}\,. \tag{4}$$

Thus the problem of finding the temperature of operation was reduced to find the temperature coefficient of electrical resistivity of the wire's material, which turned out to be a crucial step, as well reported in [4]. In this experiment an iterative approach was used in order to determine the value of this coefficient with enough accuracy and a quite smaller value than the ones reported in the literature was found. This is also confirmed in [4]. From previous analysis performed on non rotating pipe flow, it was found that even for this case a small radial temperature gradient occurs, affecting the hot-wire response particularly in the near wall region. Therefore, it is either recommended to compensate the influence of the temperature drift or to insulate the pipe wall to keep the temperature constant over a wide range. The signals from the CTA and CCA channels of an AN-1003 hot-wire anemometry system were digitised on a PC using a 16−bit A/D converter at a sampling frequency of 4 kHz and a sampling duration between 30 s and 60 s depending on the downstream position of the probe.

In order to check the accuracy of the measurement techniques the streamwise evolution of the axial flux of axial momentum $M_x = 2\pi\rho\int_0^\infty r(U^2 - 0.5V^2)\,dr$ and heat $\Theta_x = 2\pi\rho\frac{U_b}{\Delta T}\int_0^\infty rUT\,dr$ has been computed. Figure 3 shows that the both are conserved as predicted by the theory.

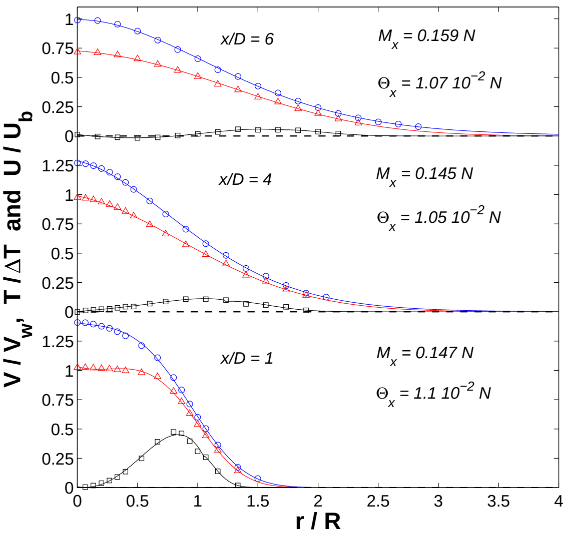

Figure 3: Axial fluxes of axial momentum and heat for $x/D = 1$, 4 and 6. $V/V_w(\square)$, $T/\Delta T(\triangle)$ and $U/U_b(\circ)$

3. RESULTS AND DISCUSSION

Mean axial velocity and mean temperature radial profiles at the jet exit plane ($x/D = 0$) and at 6 pipe diameters downstream are shown in figure 4 for non-swirling and swirling case. The velocity and the temperature for both the non-swirling and swirling jet are normalized by the bulk velocity U_b and the temperature difference ΔT, respectively. Hereby ΔT denotes the difference between the temperature at the center of the jet at the pipe outlet and the ambient temperature for the non-swirling case. The figure points out the effect of rotation in the turbulent pipe flow at the exit. The pipe rotation influences the mean streamwise velocity component such that the maximum velocity in the center of the pipe increases while the velocity close to the wall decreases. This effect results in a sligthly decrease of the shear stresses at the wall and an overall decrease of the pressure-drop. Moving downstream the rotation decreases the velocity at the jet axis and conversely increases the one in the outer part of the mixing layer, i.e. for radial distances larger then the pipe radius ($|r/R| > 1$). The figure shows also that the profiles have a common crossing point at about $|r/R| = 1$, a characteristic that is also found for all other streamwise positions, and which confirms results from previous investigations using the same experimental set-up [1]. The decay of the mean axial velocity and temperature at the center of the jet along the streamwise direction is shown in figure 5. It is clearly shown that the rotation increases the decay rate moving downstream from the jet exit.

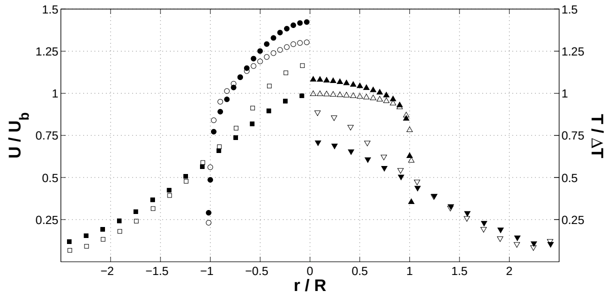

Figure 4: Mean axial velocity ($r/R < 0$) and temperature ($r/R > 0$) across the heated non-swirled (open markers) and swirled (filled markers) jet for two different downstream positions. $x/D = 0\,(\circ\ \triangle)$, $x/D = 6\,(\square\,\triangledown)$

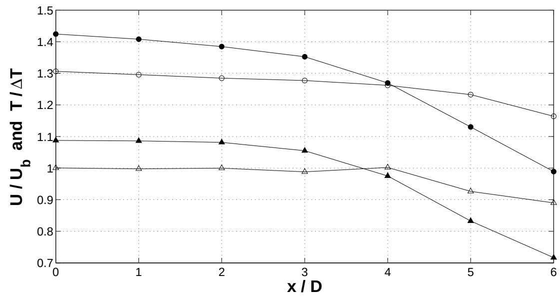

Figure 5: Downstream evolutuion of the mean axial velocity ($\circ$) and temperature ($\triangle$) along the centerline for the heated non-swirled (open markers) and swirled (filled markers) jet.

Figure 6 shows the radial distribution of the rms ($u' =< u^2 >^{1/2}$) values for the axial velocity and the temperature fluctuation. The downstream evolution, measured at the jet axis, from the pipe outlet for the same fluctuations is shown in figure 7. In this case, close to the pipe outlet, while the mean axial velocity fluctuations decrease due to the rotation, even if it is only a small effect, an increase of the temperature fluctuations is measured. Nevertheless, the overall behaviour of the temperature fluctuations follows the same trend as the axial velocity fluctuations. In the non rotating pipe, the turbulence level remains constant at the jet axis for a distance of about 3 diameters from the jet exit, similarly to what happen in the potential core of a top-hat jet profile. When the pipe is rotated, the swirl has the effect of increase progressively the turbulence level with a strong increase at ($|x/D| = 2$). The effect of the swirl in the spreading of the jet is shown in figure 8. The figure shows the radial position where the velocity and the temperature reach 50% and 10% of their value of the centerline. The 0.1 lines show how the external mixing layer spreads. It is clear that the rotation increase the spreading. No clear differences can be seen by comparing velocity or temperature. This is probably linked to the diffusion coefficient for momentum and scalar which for fully turbulent flows are very similar. However, if the 0.5 value is picked up (this is a typical value used to define the jet width), then a different behaviour is obtained. While the non rotating jet remains almost constant, the rotating one seems to slightly shrink reaching a minimum at ($|x/D| = 2$). In this region the differences between the spreading of momentum and the spreading of temperature are somewhat different. According to the Prandtl number of this flow the temperature seems to spread more rapidly than the velocity. The effect of rotation can be also assessed by looking at the entrainment coefficients presented in figure 9. As can be seen from the figure the increase in the mass flow is mostly obtained at the exit where the rotation is strong. Moving downstream the azimuthal velocity decreases and the entrainment rate, i.e. the slope of the curve, seems to become the same.

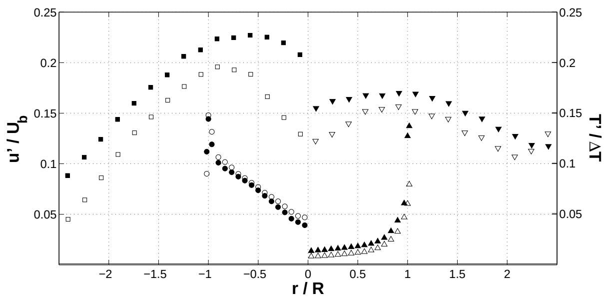

Figure 6: Axial ($r/R < 0$) and temperature ($r/R > 0$) turbulence intensity across the heated non-swirled (open markers) and swirled (filled markers) jet for two different downstream positions. $x/D = 0$ ($\circ$ $\triangle$), $x/D = 6$ ($\square$ $\triangledown$)

The correlation coefficient of the axial velocity and temperature ($\rho_{u'T'} = \frac{<\overline{u'T'}>}{u'T'}$) for all measured downstream positions across the heated jet is shown in figure 10.

In the figure it is possible to appreciate their evolution and at the same time the effect of rotation. To better appreciate the swirl effects the streamwise development of the same quantity in the jet axis is shown in figure 11.

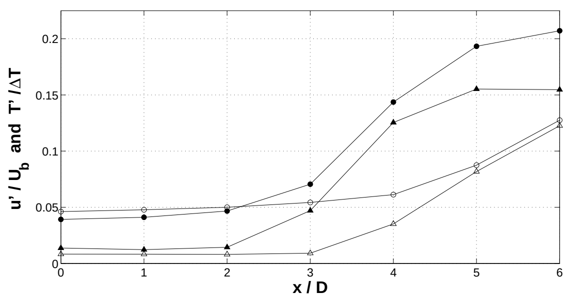

Figure 7: Downstream evolution of the turbulence intensity of axial velocity ($\circ$) and temperature ($\triangle$) along the centerline for the heated non-swirled (open markers) and swirled (filled markers) jet.

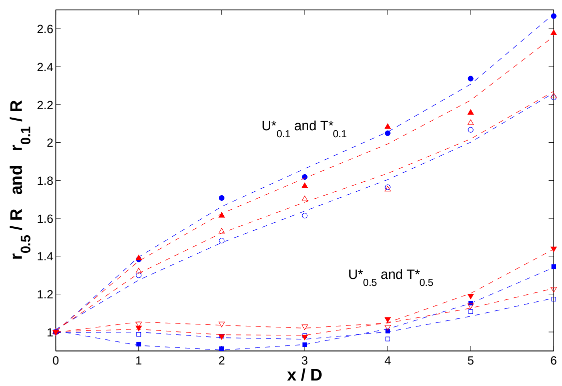

Figure 8: Jet half width and external jet mixing layer width for velocity ($\square$ and $\circ$) and temperature ($\triangledown$ and $\triangle$) along the centerline for the heated non-swirled (open markers) and swirled (filled markers) jet.

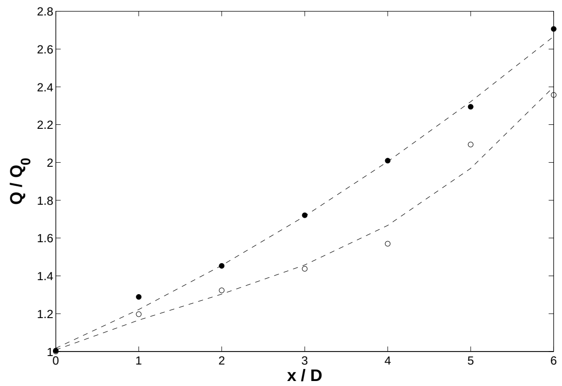

Figure 9: Entrainment coefficient (Q/Q_0) for the heated non-swirled (open markers) and swirled (filled markers) jet along the streamwise direction.

4. CONCLUDING REMARKS

In this paper a preliminar experimental analysis of the axial velocity and thermal field characteristics produced by

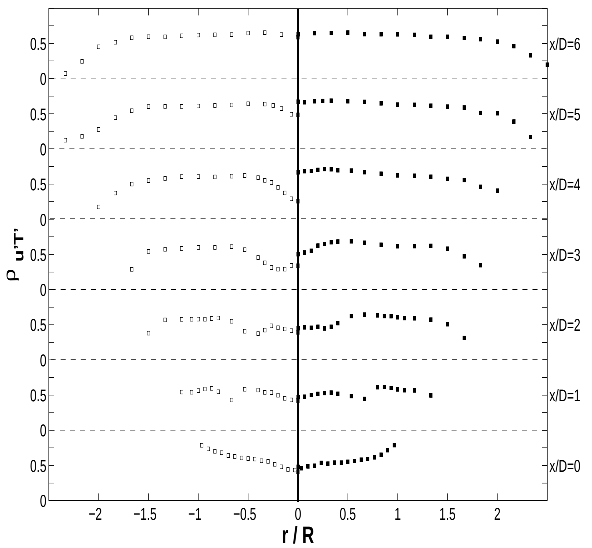

Figure 10: Correlation coefficient of axial velocity and temperature fluctuations across the jet for the heated non-swirled (open marker) and swirled (filled marker) jet at different downstream positions.

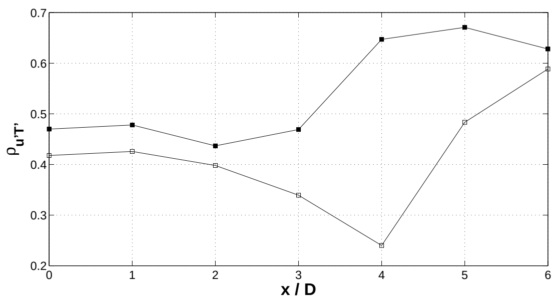

Figure 11: Downstream evolution of the correlation coefficient of axial velocity and temperature fluctuations along the centerline for the heated non-swirled (open marker) and swirled (filled marker) jet.

a fully developed turbulent rotating pipe flow is presented. There are, to the author's knowledge, no previous works on heated turbulent swirling jets issuing from a fully developed rotating pipe flow. The implemented combined X-wire and cold-wire measurement technique with temperature compensation seems to provide reliable data. Mean and fluctuating velocity components and temperature distribution result to be in good agreement with previous experimental investigations on swirling jets [1] and heated round jets [5] emanating from a fully developed pipe flow.

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