IMPROVEMENT OF THERMAL HOMOGENIZATION USING MULTIPLE SWIRLING JETS

by

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The aim of this study is to examine different blowing configurations of multiple swirling jets for use it in terminal units of ventilation applications. The influence of several parameters such as the inclined vanes of diffuser and the sense of rotation of the single or multiple swirling jets, their number and their arrangement on the flow resulting dynamically and thermally is experimentally investigated. Flow rate was adjusted at Reynolds numbers, Re_0*, ranging from* 10^4 *to* $30{\cdot}10^3$*. The current study is carried out under uniform heat flux condition for each diffuser at Reynolds number of* $30{\cdot}10^3$*, the air being the working fluid. Experiences concerning the fusion of several jets show that the resulting jet is clearly more homogenized under swirling influence. The findings of this study show that the gap between the jets and their sense of rotation relative to the central jet, affects the quality of the homogenization of ambiance. Among the studied different configuration, the one which consists of a swirling central jet controlling the behavior of six swirling jets in counter-rotation is shown to be the most effective in terms of thermal destratification.*

Keywords: *multiple free jets, swirling flow, thermal homogenization, vanes swirler, ventilation improvement, air blowing*

Introduction

Swirling jets are widely encountered in engineering facilities, *e. g.* cyclone combustors, combustion engines, tangentially fired furnaces, and swirl burners. They are often utilized to improve heat transfer in heating, ventilation, and cooling systems. When high heat transfer coefficients are needed over a wide area, swirling jets are usually used for heat transfer devices. These, they presents an interesting flow pattern for practical and theoretical outlooks. Such configurations can be used extensively in many structures and industrial applications, *e. g.* air conditioning, drying of food products, textiles, films, papers, and burners and many others.

The induced air entrainment by the rapid decrease of the average velocity involves a significant transfer of a radial momentum. Experimental results show generally that the swirl increases the spread of the jet, and particularly when the swirl number (S) is high [1].

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The understanding of swirl effects, particularly on the entrainment rate of air and on the stratification of temperature is very important for the efficiency of air conditioning process. However, to our knowledge, these effects have been scarcely investigated, and consequently the fusion of many swirling jets becomes interesting to study. The multiple swirling free jets studies show that the swirling jets will develop more rapidly than the jets without swirl. Note that the number of jets contributes to decrease velocities. Also, the distance between blowing orifices involves a decrease of velocities while delaying jets fusion. Far from the orifice and for great swirl numbers, velocity profiles of multiple jets have a tendency to increase compared with those of the single jet. The interaction between jets allows the distribution of velocities in the mixing zone in which the normal stresses and maximum shear are located. At near origin of the swirler, the profiles are characterized by irregularities due to the swirler geometry and the blowing conditions. The axial temperature for the multijet seems to be an exponential decrease [2-4]. Optimization of these parameters would influence the quality of the thermal stratification induced in the treated atmosphere. That suggests the applicability of the interaction of swirling jets in the ventilation of premises, confined environment as public transport vehicles or drying operations. According to the available literature, there is no advanced research being done, on the multiple swirling jets applied to improve comfort conditions. Most of papers which deal with multiple swirling jets in various geometric or dynamic and thermal conditions are aimed to the improvement of combustion. Swirl flows have been widely used in combustion systems as they enhance mixing between fuel and oxidant [5, 6]. Practically, they appear in aircraft combustors and in a residential/commercial burner where swirl contributes to improve mixing and stabilizes the flame. They are also widely used in industrial burners of power station furnaces or gas turbine combustors to provide stable, high intensity, short flames with wide radial development resulting from good fluid mixing [7]. Therefore, it should be pointed out that, there are still many points to be clarified concerning the understanding of the effect of swirl on heat transfer and behaviour of the induced flow.

Ahmadvand *et al.* [8] studied experimentally and numerically the influences of axial vane swirler on increase of heat transfer and turbulent fluid flow. Their study has been carried out for three blade angles of 30°, 45°, and 60° with uniform heat flux condition of air which is used as the working fluid. These authors confirmed that the use of vane swirler leads to a higher heat transfer compared with those obtained from plain tubes, and the thermal performance increases as vane angle is raised and decreases by growth of the Reynolds number.

The results of the theoretical and experimental investigation of the turbulent mean swirling flows characteristics of the conical diffuser have been obtained by Benišek *et al.* [9]. The swirl flow fields were induced by the axial fan impeller. They found that various swirl parameters were achieved by the impeller opening and rotational speeds.

Lee *et al.* [10] evaluated the influence of factors as cavity ventilation, the slope of the roof, intensity of solar radiation, the size and figure of a cavity, and panel profiles, by investigating airflow and temperature distribution in the cavity, to improve the cooling effect of ventilation in the cavity of the roof. They used thermoanemometer (Velocicalc Plus model 8360) to measure air velocity which is a high-precision multifunctional instrument.

Yin *et al.* [11] experimentally investigated twin jets flow, generated by two identical parallel axisymmetric nozzles. He founded that the twin jets attract each other. With the increasing the Reynolds number, the turbulence energy grows, which indicates that the twin jets attract acutely. He also observed that the jet flow field and the merging process of twin jets vary with the spacing between two nozzles. One of his salient findings is that the width of the twin jets flow spreads linearly downstream and grows with the spacing between two nozzles.

In order to clarify the influence of the flow behaviour on heat transfer fields, Oyakawa *et al.* [12] carried out an experimental study. They measured the time and spatial heat transfer coefficients over the impingement plate using an infrared radiometer when four impinging jets hit a plate for various jets arrangements and separating distances from jet exit to the impingement plate. They founded that the heat and flow behaviour of multiple jets show more complicated characteristics due to the existence of the interactions between adjoining jets, and between jets and spent flows.

Dae *et al.* [13] investigated the heat transfer characteristics using swirling round jet impinging upon the flat plate surface. They noted that the variation of Nusselt numbers for a low swirl number S = 0.77 at a large L/d area (L/d =10, L and d being the nozzle-to-plate distance and the nozzle pipe diameter, respectively) is smallest within ±15-20% of the average Nusselt number in the entire region. This finding allowed them to assert that such configuration can be used for the purpose of the uniform heating or cooling applications.

Valentina *et al.* [14] used 3-D laser Doppler anemometry (LDA) technique and flow visualisation to investigate the burner-burner interaction on an isothermal physical model of a front-wall fired furnace. Four different pitches arranged in a 3×3 array were tested. They observed that, as pitch decreased, interactions between burners became more evident. They also noted that isothermal burner jets were fluctuating arbitrarily in their width, and exhibiting time-dependent features. The LDA results clearly showed differences between flow velocity patterns occurring at different pitches and different configurations. Also, they have observed that both recirculation and jet penetration were more sensitive to the change of the configuration for the smaller pitches. Equally, the sensitivity to the change in pitch was higher in the case of the chequered board configuration.

Yimer *et al.* [15] experimentally studied the development of the flow from multiple-jet burners. They measured the fields of mean velocity and velocity fluctuation intensity with Pitot probes. They noted that beyond the near field, flows are virtually the same and similar to those of a round jet from a single source, and in the near field, individual structures peculiar to each burner are observed.

Akililu *et al.* [16] numerically simulated the mixing process of turbulent streams within a pipeline due to single and multiple transverse jets using the k-ε turbulence model. They founded that multiples jets ensure better mixing in addition to decreasing power requirements with an increasing number of jet.

Nakod *et al.* [17] performed an experimental investigation to study the effect of the fined surfaces and surfaces with vortex generators on the local heat transfer coefficient between impinging circular air jet and flat point. They founded that the heat transfer augmentation in case of vortex generator is as high as 110% for a single row of six vortex generators at a radius of one nozzle diameter as compared to the smooth surface at a given nozzle plate spacing of one nozzle diameter. They concluded the optimum configuration of the vortex generators are single row of six vortex generators.

Felli *et al.* [18] tackled experimentally the dynamic of impinging swirl jet generated by a ducted propeller. They observed that the wall modifies the shape of the swirling jet causing its spread-out and generating a re-circulating zone around which the hub vortex rolls out before breaking down against the wall surface.

Through the literature review, it appears that vortex flows have undeniably some advantages in terms of the power mixing. As mentioned above, all research conducted on such kind of flow are somewhat remote from our present study. Thus, assessing the relevance of integrating the turbulent jets in the air handling and ventilation of living spaces and transport requires a prior study and analysis of multijet swirling over its entire length. For that raison, the choice of a system consisting of blow jets more efficient in terms of mixing may be necessary.

From the above discussion, the main aim of this study is to examine and perhaps control the influence of various parameters such as the sense of rotation of the single or multiple swirling jets, their number and arrangements on the flow resulting both dynamically and thermally. In addition, we expect that optimization of these parameters would influence the thermal stratification of the atmosphere, and help to optimize the choice of the configuration of interest to industry.

Experimental set-up and techniques

The experimental facility is depicted in fig. 1. It consists of a chassis on which is fixed a square plexiglas plate. On the latter, seven devices blowing hot air are fixed and directed downwards, and the lower part of these devices is used to fix different types of diffusers provided with inclined vanes, depending on the studied configuration. Temperatures and velocity of the flow are measured by a thermo-anemometer (type Velocicalc Plus Air velocity Meter, Operation, and Service Manual 1980321, Revision H., June 2006) which is a high-precision multifunctional instrument. The data can be viewed on screen, printed or downloaded to a spreadsheet program allowing us to easily transfer data to a computer for statistical treatment. The accuracy is of order ±0.015 m/s for velocity and ±0.3 °C for temperature from thermal sensor. Note that the thermal sensor is supported by rods which are easily guided vertically and horizontally to sweep the maximum space in the axial and radial directions (fig. 1).

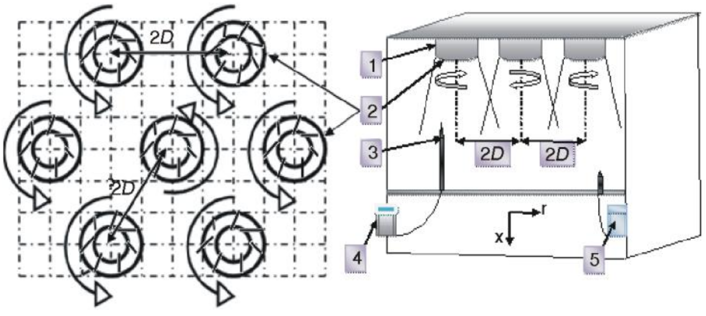

Figure 1. Experimental facility and test assembly
1 – air blowing devices, 2 – diffuser with inclined vanes, 3 – thermoanemometer probe, 4 – thermoanemometer, 5 – thermometer

The swirling free jet considered here is different from the conventional jet because of the existence of a tangential component velocity. To achieve this type of flow, one can either use the axial fan impeller for generating swirl turbulent flow, or use swirling mechanical systems [19]. For example, this system includes inclined vanes, figs. 2(a) and 2(c), which is put in the generating tube jet, fig. 2(b). The application of a tangential velocity component to the flow (W) provides a rotation to flow fluid, which is indicated by a so-called swirl number (S). This number is defined as the ratio of the axial flux of tangential momentum to the product of the axial momentum flux and a characteristic radius [20]. It should be noted that the exact expression of swirl number depends on the injector geometry and flow profiles. Following Gupta *et al.* [20], and for a typical single element injector with a flat vane swirler, the swirl number can be defined as:

$$S = \frac{G_\theta}{RG_x} = \frac{\int_{R_n}^{R_h} UWr^2 \mathrm{d}r}{\int_{R_n}^{R_h} R_n U^2 r \mathrm{d}r} \tag{1}$$

where G_θ is the axial flux of tangential momentum, G_x – the axial momentum flux, and R – a characteristic radius. R_n and R_h are radius of the centre body and the inlet duct, respectively. It is important to note here that if the axial and azimuthally velocities are assumed to be uniform and the vane are very thin, the swirl number can be written as [21]:

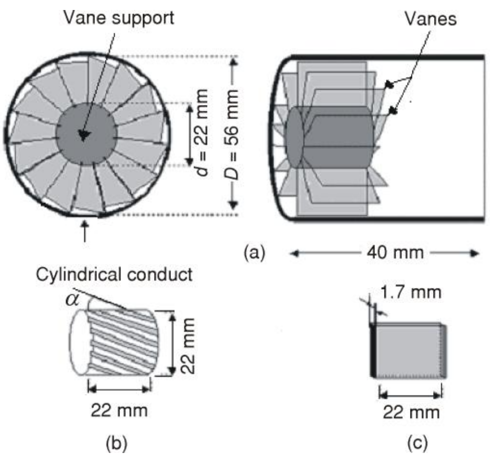

Figure 2. Schematic of swirling generator device
(a) swirling generator, (b) vane support, (c) vanes

$$S = \frac{2}{3}\left[\frac{1-\left(\dfrac{R_\mathrm{h}}{R_\mathrm{n}}\right)^3}{1-\left(\dfrac{R_\mathrm{h}}{R_\mathrm{n}}\right)^2}\right]\mathrm{tg}\alpha \tag{2}$$

where α is the swirler vane angle. Note that in the case of a hubless swirler ($R_h = 0$), the above expression is reduced as:

$$S = \frac{2}{3}\mathrm{tg}\alpha \tag{3}$$

In this study, the axial and tangential velocities U and W, respectively, were measured at the exit of a swirling jet diffuser with a triple probes hot wire anemometer (DISA 55M01). Four swirl numbers values are used in this study. There are $S = 0$ for $\alpha = 0°$, $S = 0.4$ for $\alpha = 30°$, $S = 0.7$ for $\alpha = 45°$, and $S = 1.3$ for $\alpha = 60°$, respectively.

To carry out our experiments, the following operating conditions have been considered: $0 < S < 1.3$, $Q_\mathrm{m} = 0.041$ kg/s, $\mathrm{Re}_0 = 30{\cdot}10^3$, $r/D = 1$ to 8 and $0 \leq x/D \leq 20$. Here it is useful to note that previous studies were based on similar ranges of Reynolds (see [22, 23] to name a few).

In addition, the references having addressed topics close to our work are based on similar ranges of Reynolds.

It is worth recalling that our goal is to identify and study the evolution of temperature profiles of axial and radial multijet swirling in different configurations. This approach allows analyzing the influence of key parameters such as angle of inclination of the vanes, the spacing between diffusers, and the number of peripheral jets which are controlled by a central jet.

Results and discussion

The single swirling jet

Previous studies showed that for a range of experimental conditions, heat transfer enhancement is strongly depending on blade angle and it seems that Re has not significant influence, whereas when the initial angle of the velocity α increases, the jet is more spreaded in the radial direction [8]. Figure 3 shows the profile of the mean flow temperature in the axial direction at different velocity initial angles. As it can be seen, we note that the axial temperature decreases rapidly when α increases. Hence, the ambient temperature is quickly reached and the area under influence is more important.

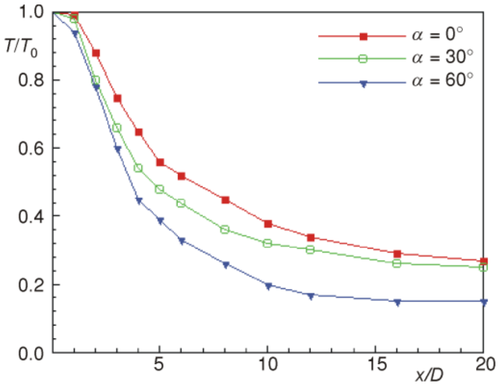

Figure 3. Dimensionless axial temperatures profiles for different vane inclination. Case of a single swirling jet

According to Florent *et al.* [24], the development out of the recirculation zone if it exists, is of the following form:

$$\frac{U}{U_0} = K\left(\frac{x}{D}\right)^{-\gamma} \tag{4}$$

where K and γ are parameters which depends on the initial angle. Thus, a velocity of $0.2U_0$ where U_0 is the axial velocity at the origin of the blowing is obtained at 25D for a free jet without inclination (α = 0°). It is interesting to note that this velocity is equal at 15D and 5D for an angle of 30° and 60°, respectively. D is the diameter of the blowing origin. Thus, by blowing with a jet under limited height, one can get the comfort we seek to achieve by increasing the initial angle of velocity. According the radial direction, the more this angle increases, more the jet expands. Concerning the development of temperature, similar results are obtained. In fact, more the initial angle of the inclination increases, more the axial temperature decreases rapidly and more the area affected is important (fig. 3). It has stipulated that axial and radial developments of the temperatures far enough from the origin of the blowing are of the form [24]:

$$\frac{T}{T_0} = K_1\left(\frac{x}{D}\right)^{-\beta_1} \quad \text{and} \quad \frac{T}{T_0} = K_2\left(\frac{r}{D}\right)^{-\beta_2} \tag{5}$$

where coefficients K_1, K_2, β_1, and β_2 depend on the inclination of the vanes of the swirl generator. There is therefore a radial temperature gradient. Note that, here the mixing of the jets is of particular importance (fig. 6).

Based on a classical analogy between velocity and temperature profiles, it seems appropriate to characterize the jet by using a scalar quantity instead a vector quantity (for details, the reader could refer to [24]). Figures 4 and 5 give the radial comparison between the dimensionless mean velocities and mean dimensionless temperature profiles for a single swirling jet for two values of the vane angle α (α = 30° and α = 60°) at a given station x/D =2, respectively. It is showed that the shape of the dimensionless radial temperature and velocities profiles are quite similar confirming the analogy recounted above.

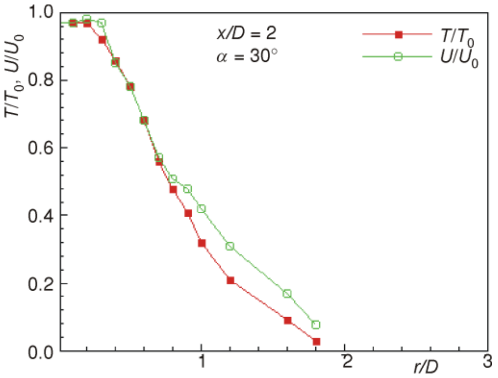

Figure 4. Dimensionless radial velocities and temperatures profiles for a single swirling jet at a swirler vane angle α of 30° and at the station x/D = 2

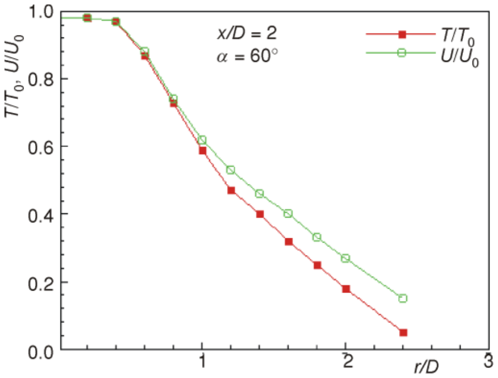

Figure 5. Dimensionless radial velocities and temperatures profiles for a single swirling jet with the swirler vane angle α = 60° at the station x/D = 2

Figure 6 shows the comparison of the dimensionless mean radial temperature for different swirler vane angle's α. It can seen that, only with α = 60°, the temperature profile decreases less rapidly giving an important spreading compared to cases corresponding to α = 0° and α = 30°. Also, we note that the initial angle velocity α = 60° improves significantly a thermal homogenization of the flow. It is this angle which is adopted throughout this work.

Double swirling jet

In order to advance the understanding of the effect of number of jets, we performed the installation of the device with two swirling jets (fig. 7) where blowing orifices are distant at 2-D or 3-D. Dimensionless radial profiles of the temperature are plotted in fig. 8 at the station x/D = 8, which is located far from the blowing orifice. It is noted from this figure that the temperature decreases as the radial distance increases. Obviously, the use of such a device provides fairly regular profiles, and consequently an important thermal homogenization. In the radial direction, the temperature is more important for the double swirling jet than for the single swirling jet. The spacing between blowing orifices gives rise to a spreading of

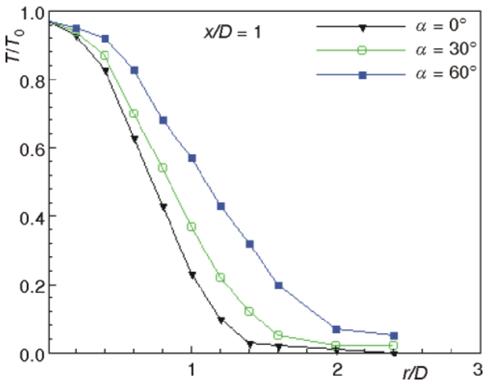

Figure 6. Dimensionless radial temperature profiles for the single swirling jet at different swirler vane angle α and for x/D = 1

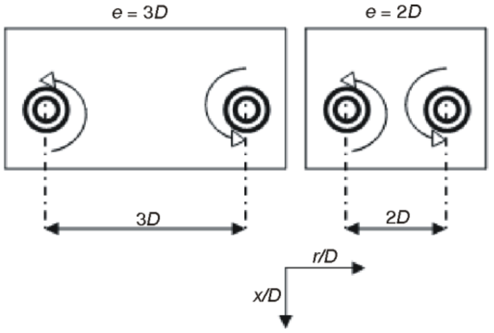

Figure7. Double swirling jets arrangement for two spacing between blowing orifices

radial profiles with a relative decrease of temperature amplitudes in the zone between the blowing orifices confirming results founded by Zhao-Qin [11]. In the case of two swirling radial jets, it is interesting to note that the peak temperature decreases less sharply than that which is achieved in the case of single swirling jet.

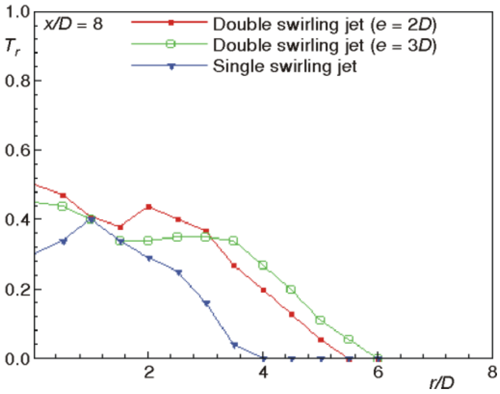

Figure 8. Dimensionless radial temperatures profiles for swirling single and double jets at the station *x/D* = *8* for two spacing between blowing orifices

It is further noted from fig. 8 that the double swirling jets case with $e = 3D$ is found to be quite different from that of the single jet. Moreover, the double swirl jet device seems to provide a better thermal homogenization. This, it can be explained by the fact that the initial kinetic energy is dissipated by viscous stresses. According to this figure, it can be seen that the spacing between blowing orifices allows the reduction of the radial amplitude of the temperature, while providing both a substantial spreading and homogenization. In the dynamic study, the space between blowing orifices causes a diminution of blowing velocities, and delays the fusion of jets while increasing the fluctuation intensities [24].

Plane configuration with three and four swirling jets in a triangle arrangement

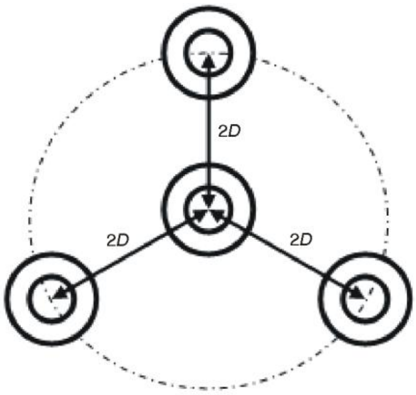

Figure 9. Arrangement of 3 peripherical swirling jets (a swirling four-jet)

Figure 9 shows the equilateral triangular arrangement of the configuration with three swirling jets including a central swirling jet. The spacing e between jet axes is equal to $2D$.

Figure 10 shows the axial evolution of the dimensionless temperature for the single central jet, and for three and four swirling jets including a central jet, respectively. For the last configuration, we note that the temperature decreases sharply. In the case where only the peripheral jets are active, the temperature begins to increase due to less of the mixture. However, where all four jets are active, the temperature decreases less rapidly compared to the single swirling jet case. Indeed, the central jet modifies the adjacent behaviour of peripheral jets and improves the homogenization of flow and the temperature decreases monotonically. Through our experiments, we noted that the evolution of the temperature reveals the importance and the role played by the presence of the central jet surrounded by peripheral jets. In the three-jet configuration, and near the blowing orifice, it is found that the temperature profile increases first and then decrease until it reaches the ambient temperature at a distance of about of $20D$. Moreover, it is observed that the maximum is located three diameters of the orifice blowing, and it indicates the beginning of the mixing zone. However, in the case of four jet configuration including the central jet, we find that the maximum is reached near the hole at a distance almost equal to a diameter. Then the temperature undergoes a rapid decrease up to 4 diameters of the discharge port. This decrease goes beyond 50% from its maximum value. From this station and up to an axial distance of about 10 times the diameter, the temperature profile has a slope much less pronounced than the first. Beyond a distance of about 10 diameters, the intensity decreases slightly and the profile becomes stable throughout the flow. This decrease involves a transfer of energy to the radial direction of the flow (fig. 11). These findings allow us to say that the performance of mixing jets including the central jet is confirmed, and suggest the increase of number of peripheral jets.

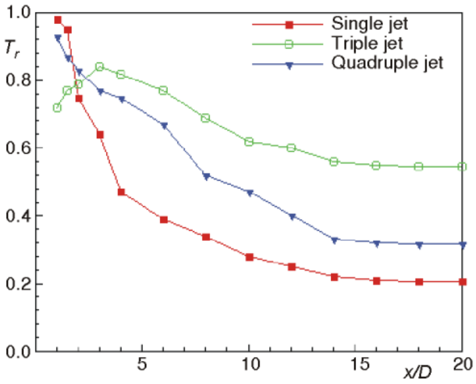

Figure 10. Dimensionless axial temperature profiles for 1, 3, and 4 swirling jets in triangle arrangement

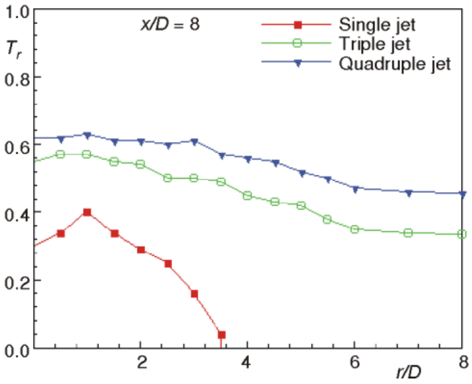

Figure 11. Dimensionless radial profiles of temperature at station x/*D=8* for a single jet, 3, and 4 swirling jets in a triangular arrangement

The development of the radial temperature at position $x/D = 8$ is shown in fig. 11 for the same configurations as above. Again, we note that the temperature decreases in the radial direction, more pronounced in the single swirling jet case. As can be seen, the evolution of the radial temperature for treated configurations remains almost constant due to the presence of central jet. This effect proves to be interesting for control the swirling flow with the aim of optimizing the power consumption by blowing system.

Plane configuration with seven swirling jets

The configuration (fig. 12) consists of a swirling central jet surrounded by six peripheral swirling jets. Such a configuration is commonly referred to seven swirling jet arrangement. This later allows increasing the flow rate. We seek here to test this arrangement in order to control several peripheral jets thanks to the driving central jet, which is in reversed rotation with respect to adjacent jets. The effect of counter rotation permits to increase the turbulence intensity in the resulting jet.

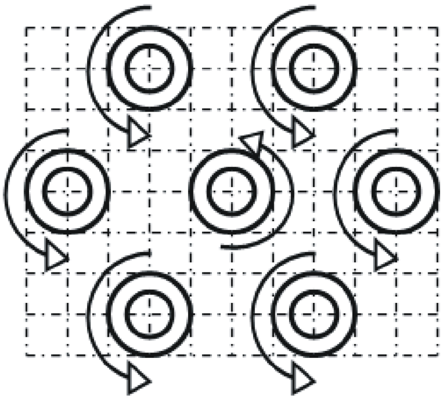

Figure 12. Arrangement of sevenfold swirling jets

After confirming the importance of the presence of the central jet in a system with three jets, we found it useful to test a configuration with a large number of peripheral jets. Our choice was a configuration with six jets including a central jet.

Figure 13 depicts the dimensionless axial evolution of the temperature in the case of seven swirling jets. It can be observed that the temperature decreases almost linearly from one station to another to reach ambient temperature downstream of the jet (*x/D* = 15). This decrease is due simultaneously to the distance from the blowing origin and mixing of jets induced by the central jet. Hence, we can say that the increasing of the number of peripheral jets leads to a significant interaction between jets, thus reducing the axial temperature more sharply. This increases the lateral diffusion, which promotes uniformity of velocities and temperature, and leading to a better homogenization.

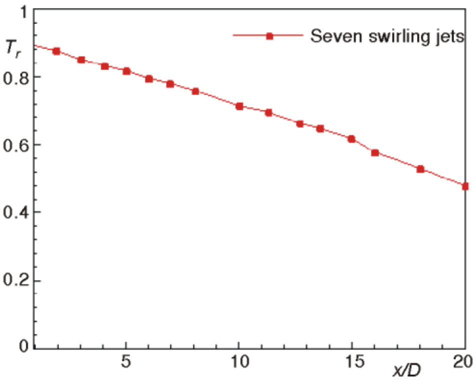

Figure 13. Dimensionless profiles of the axial temperature for the configuration of 7 swirling jets

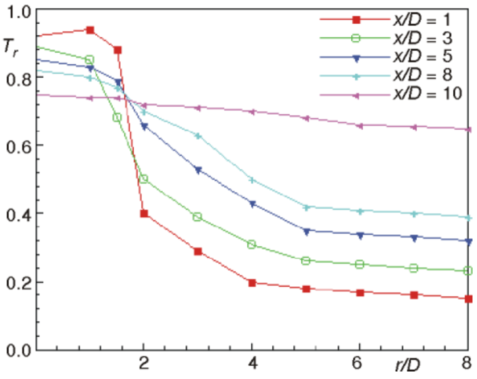

Figure 14. Dimensionless profiles of the radial temperature for the 7 swirling jets configuration

Figure 14 shows the radial evolution of the dimensionless temperature for different positions *x/D*. We note that all curves decreases from the station *x/D* = 1. Away from the blowing orifice, the radial profiles decrease slowly becoming almost constant at *x/D* = 10. At this station, the thermal homogenization is almost complete. Compared with the four-jet configuration, it can be seen that at station *x/D* = 10 and beyond, the configuration with a large number of jets leads to an almost perfect homogenization of the fluid which is initially blowed. This result may be associated with a good distant of the jet, and particularly when we aim to treat air conditioning, which requires a high rate of the resultant jet in order to cover the treated space.

Conclusions

In this work, we explored different configurations of blowing multiple swirling jets for use in ventilation applications. We have highlighted more improvement of the thermal homogenization of the treated area using multiple swirling jets with an appropriate choice of the position for blowing air. The analysis of the flow features clearly demonstrate that the interaction between swirling jets induces the redistribution of temperature in the mixing zone, while allowing the spreading of the resulting jet. It appears that the central jet plays an important role in the enhancement of the thermal homogenization. In light of the obtained results, we can conclude that:

- For the single swirling jet
  - The shape of the dimensionless radial temperature and velocities profiles are quite similar.
  - Only with α = 60°, the temperature profile decreases less rapidly giving an important spreading compared to cases corresponding to α = 0°, and α = 30°.
  - The initial angle velocity α = 60°, improves significantly a thermal homogenization of the flow.
- For the double swirling jet
  - The spacing between blowing orifices gives rise to a spreading of radial profiles with a relative decrease of temperature amplitudes in the zone between the blowing orifices confirming available results [11].
  - The spacing between blowing orifices allows the reduction of the radial amplitude of the temperature, while providing both a substantial spreading and homogenization. In the dynamic study, the space between blowing orifices causes a diminution of blowing velocities, and delays the fusion of jets while increasing the fluctuation intensities.
- For the plane configuration with three and four swirling jets in a triangle arrangement
  - When all four jets are active, the temperature decreases less rapidly compared to the single swirling jet case. Indeed, the central jet modifies the adjacent behaviour of peripheral jets and improves the homogenization of flow and the temperature decreases monotonically.
- For the plane configuration with seven swirling jets
  - At station x/D = 10 and beyond, the configuration with a large number of jets leads to an almost perfect homogenization.

The optimization of parameters such as the diffuser's geometry, the swirler vane angle, the spacing between jets, the number of blowing jets, and the relative rate between the central and peripheral jets, are power means to enhance the quality of the air contained in the confined space.

It should be noted that among all the configurations considered here, the configuration with a swirling central jet surrounded by six peripheral jets has proven the most effective in terms of thermal non-stratification. This finding highlights the simultaneous part of the central jet and of its neighbors**.**

Nomenclature

D – inner diameter of one diffuser, [m]
e – spacing between diffusers, [m]
K_1, K_2 – coefficients depending on the initial angle of the blowing velocity
Q_m – mass flow rate, [kgs^{-1}]
Re$_0$ – Reynolds number at air blowing origin, [–]
R_h – radius of the inlet duct, [m]
R_n – radius of the centre body of swirler, [m]
r – radial coordinate of air flow, [m]
S – swirl number
T – temperature of jet, [K]
T_0 – maximum temperature of the air blowing at origin, [K]
T_r – dimensionless temperature $(T - T_a)/(T_0 - T_a)$
U_0 – maximum value of the velocity of the air blowing at origin, [ms^{-1}]
U – mean axial velocity, [ms^{-1}]
W – mean tangential velocity, [ms^{-1}]
x – axial coordinate of the air flow, [m]

Greek symbols

α – inclination angle of the vanes
β_1, β_2 – coefficients depending on the initial angle of the blowing rate

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