

## TURBULENT HEAT TRANSFER FOR IMPINGING JET FLOWING INSIDE A CYLINDRICAL HOT CAVITY

by

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*Convective heat transfer from an isothermal hot cylindrical cavity due to a turbulent round jet impingement is investigated numerically. Three-dimensional turbulent flow is considered in this work. The Reynolds stress second order turbulence model with wall standard treatment is used for the turbulence predictions the problem parameters are the jet exit Reynolds number, ranging from  $2 \cdot 10^4$  to  $10^5$  and the normalized impinging distance to the cavity bottom and the jet exit  $L_f$ , ranging from 4 to 35. The computed flow patterns and isotherms for various combinations of these parameters are analyzed in order to understand the effect of the cavity confinement on the heat transfer phenomena. The flow in the cavity is divided into three parts, the area of free jet, and the area of the jet interaction with the reverse flow and the semi-quiet flow in the region of the cavity bottom. The distribution of the local and mean Nusselt numbers along the cavity walls for above combinations of the flow parameters are detailed. Results are compared against to corresponding cases for impinging jet on a plate for the case of the bottom wall. The analysis reveals that the average Nusselt number increases considerably with the jet exit Reynolds number. Finally, it was found that the average Nusselt number at the stagnation point could be correlated by a relationship in the form  $\bar{Nu} = f(L_f, Re)$ .*

Key words: round jet, cylindrical cavity, turbulence, thermal effect,  
finite volume method, Nusselt number

### Introduction

Impingement flows are extensively used for heating, cooling, and drying processes in different industrial applications. Impinging jets are used in paper and textile industries, the thermal treatment of materials, cooling of electronic components and turbine blades, and drying of continuous sheets of materials, etc. Some applications of dehydration by impinging jets are included in manufacturing of printed wiring boards, production of foodstuffs, de-icing of aircraft wings, annealing of metal sheets, tempering of glass, etc.

Although, numerous studies were conducted for jets impingement, it remains an active domain of research. Polat *et al.* [1] published a comprehensive review on the numerical analysis of impinging jet flow and heat transfer before 1989.

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Impinging jets onto a cavity are extremely complex flow configurations, where several interactions can occur on many surfaces simultaneously. Analysis of turbulent jet impingement on cavity is characterized by strong streamline curvature, entrainment, flow recirculation, pressure gradient, and boundary layer development on several impingement surfaces. Relatively few studies that have reported on the flow field and heat transfer characteristics due to an impinging jet with a cavity even though there are many engineering applications of such flow and heat transfer process.

This configuration was usually used in many industrial devices. It can simulate the ventilation of mines, the injection of fuel into the combustion chambers. There are many works published on this subject where the effect of the cavity shape and the jet location were examined. Gilard and Brizzi [2, 3] studied experimentally the behavior of a plane jet of thickness  $d$ , flowing into a cylindrical cavity provided with a hemispherical bottom. They used for the visualization and the measurements of the flow fields, the PIV technique. Choi *et al.* [4] present an experimental study of fluid flow and heat transfer of cooling by an impinging jet onto a semi-circular concave surface. The distributions of the mean velocity and fluctuations were measured within the free jet region, the impinging region and the wall jet flow region by using laser Doppler anemometer (LDV) technique. Local Nusselt numbers have also been computed in this work.

Shuja *et al.* [5-7] studied a flow exiting from a combination of conical and annular nozzles impinging a hot cylindrical cavity. They have examined the effects of the external angle of annular nozzle and the jet exit velocity on the flow interaction. The heat transfer and skin friction coefficients within a conical cavity are also illustrated in another study of these authors, where they presented the effect of nozzle shape, cavity diameter, and cavity depth, on the flow structure and heat transfer rates. Terekhov *et al.* [8] studied experimentally the heat transfer and the flow field generated by a turbulent jet impinging a spherical cavity. Colucci and Viskanta [9] considered the effects produced by a hyperbolic nozzle of a confined impinging air jet on the local heat transfer coefficient by thermo chromatic liquid-crystal technique. Jeng and Tzeng [10] investigated numerically an impinging confined slot jet for the cooling of porous metallic foam heat sink. Mataoui *et al.* [11, 12] have investigated experimentally and numerically using two scales turbulence model the case of a turbulent plane jet flowing into a rectangular cavity. The effect of the jet unsteadiness was detailed depending on the jet location inside the cavity. Jaramillo *et al.* [13] have tested numerically the case of a confined plane jet impinging a flat plate. They used the results obtained by a direct numerical simulation DNS of previous study to improve the performance of several Reynolds-averaged Navier-Stokes (RANS) models. Lawson *et al.* [14] have presented a technique for the control of the jet oscillation in a thin rectangular cavity, by subjecting perpendicularly another jet to the main jet through the lateral cavity wall. This study was conducted experimentally and numerically by 2-D and 3-D simulation. Schwarzer *et al.* [15] have made a combined experimental and numerical study of a turbulent round jet submerged in a rectangular cavity. Kang and Tao [16] have studied experimentally the heat and mass transfer of an axisymmetric jet impinging a cylindrical cavity with one end open to the ambient flow for the jet Reynolds number range of  $10^4$  to  $8 \cdot 10^4$ , and the dimensionless impinging distance between the jet exit and the cavity bottom  $S/d$  range from 1 to 4.5. Risso and Fabre [17] carried out an experimental study of an hydrodynamic jet confined in a vertical cylindrical cavity, open in its upper part. Baydar and Ozmen [18] conducted an experimental and numerical study for deepen the flow of a confined jet impinging perpendicularly the upper surface of the cavity. The mean velocity, turbulence intensity and pressure distributions in the stagnation zone were determined for Reynolds numbers ranging from 30,000 to 50,000 and for the dimensionless impinging distance between 0.2 and 6. Prakash *et al.* [19, 20] have presented a

comparative study between the experiment of visualization, the measurements by LDV and the CFD simulation of a round jet flowing in a cylindrical cavity with a porous layer. Graminho and De Lemos [21] have examined the influence of the presence of a porous layer at the bottom of a hot cavity cooled by a turbulent round jet. Benaissa [22] and Kendil *et al.* [23] have achieved an experimental and a numerical study of a turbulent axisymmetric round jet flowing inside a cylindrical cavity open upstream, for Reynolds numbers ranging between 22,000 and 50,000. The distance between the jet exit and the cavity bottom,  $L_f$  is varied from 2 to 30 jet diameters. The flow velocity field was measured using hot wire anemometry. The computations were achieved using the standard  $k-\varepsilon$  model combined to the wall functions. Chandratilleke *et al.* [24] have proposed a new technique for improving the cooling of a hot surface by a coaxial jet directed to the surface of a cylindrical cavity. Voropayev *et al.* [25] have realized an experimental study of a turbulent round jet located in a long cylinder for the applications to crude-oil storage tanks for strategic petroleum U.S.

In the present paper, the case of a round jet impinging perpendicularly a hot cylindrical cavity is investigated numerically using the Reynolds stress model (RSM) coupled to the logarithmic wall function for the wall treatment. The jet exits from a long horizontal pipe in a cylindrical hot cavity. The radius of the pipe and the cavity are  $R_{jet}$  and  $R_{cav}$ , respectively. The configuration is sketched in fig. 1. For the validation, the flow field are compared to the measurements of Benaissa [22] for the case  $L_f = 8d$ . The purpose of this paper is to investigate the thermal study and to examine the phenomenon for different values of the impinging distance ( $4 \leq L_f \leq 35$ ).

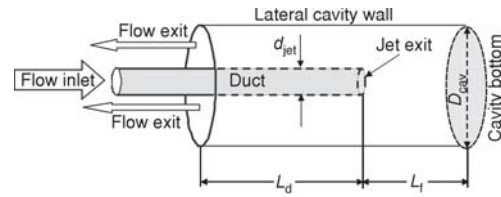


Figure 1. Geometry and computational domain

## Numerical modeling and computational methodology

### Modeling of flow field and heat transfer

The flow is assumed steady in average, turbulent, and axisymmetric. The fluid is incompressible with constant thermo physical properties. The steady Reynolds average Navier Stokes (SRANS) equations coupled to the averaged energy equation of the turbulent compressible flows are written as:

- the mass equation

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (1)$$

- the momentum equation

The Reynolds averaged momentum are:

$$\frac{\partial(\rho U_i U_j)}{\partial x_i} - \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) \right] = - \frac{\partial}{\partial x_i} \left( P + \frac{2}{3} \mu \frac{\partial U_k}{\partial x_k} \right) - \frac{\partial}{\partial x_i} (\overline{\rho u_i u_j}) + S_{Mi} \quad (2)$$

where  $P$  is the static pressure and  $S_{Mi}$  – the sum of body forces. This equation is coupled to second order RSM turbulence model. Second order models are based on transport equations for every component of the Reynolds stress tensor and the dissipation rate. These models do not use the eddy viscosity hypothesis, but solve an equation for the transport of Reynolds stresses. Algebraic Reynolds stress models solve algebraic equations for the Reynolds stresses [26], whereas

differential Reynolds stress models solve differential transport equations individually for each Reynolds stress component. The exact production term and the inherent modeling of stress anisotropies theoretically make Reynolds stress models more suited to complex flows. In the differential stress model, a separate transport equation must be solved for each of the six Reynolds stress components of  $\rho u_i u_j$ . The standard Reynolds stress model is based on dissipation  $\varepsilon$  equation:

$$\frac{\partial}{\partial x_i} (\rho U_k \overline{u_i u_j}) - \frac{\partial}{\partial x_k} \left[ \left( \mu \delta_{kl} + C_s \rho \frac{k}{\varepsilon} \right) \frac{\partial (\overline{u_i u_j})}{\partial x_l} \right] = P_{ij} - \frac{2}{3} \delta_{ij} \rho \varepsilon + \Phi_{ij} \quad (3)$$

where  $P_{ij}$ , the production term, is given by:

$$P_{ij} = \overline{\rho u_i u_k} \frac{\partial U_j}{\partial x_k} - \overline{\rho u_i u_k} \frac{\partial U_i}{\partial x_k} \quad (4)$$

and  $\Phi_{ij}$  is the pressure strain correlation. This term can be separated into two parts:

$$\Phi_{ij} = \Phi_{ij,1} + \Phi_{ij,2} \quad (5)$$

where  $\Phi_{ij,1}$  is the "slow" term, also known as the return-to-isotropy term, and  $\Phi_{ij,2}$  is called the "rapid" term.

As the turbulence, dissipation appears in the individual stress equations, an equation for  $\varepsilon$  is then required. This later is given by:

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho U_k \varepsilon)}{\partial x_k} = \frac{\varepsilon}{k} (C_{\varepsilon 1} P_k - C_{\varepsilon 2} \rho \varepsilon + C_{\varepsilon 1} P_{\varepsilon b}) + \frac{\partial}{\partial x_k} \left[ \left( \mu \delta_{kl} + C_\varepsilon \rho \frac{\varepsilon}{k} \overline{u_k u_l} \right) \frac{\partial \varepsilon}{\partial x_l} \right] \quad (6)$$

There are three version of the standard Reynolds stress models based on the  $\varepsilon$ -equation developed by Launder *et al.* [26]. These are called as LRR-IP, LRR-QI, and SSG. For the LRR-IP and LRR-QI models, the pressure-strain correlation is linear; "IP" stands for isotropization of production and is the simplest of the three models. The SSG model was developed by Speziale, *et al.* [27] uses a quadratic relation for the pressure-strain correlation. The pressure-strain correlations have a following form depending on the anisotropy tensor  $a_{ij}$ , the mean strain rate tensor  $S_{ij}$  and vorticity tensor  $\Omega_{ij}$ .

$$\Phi_{ij,1} = -\rho \varepsilon \left[ C_{S1} a_{ij} + C_{S2} \left( a_{ik} a_{kj} - \frac{1}{3} a_{mn} a_{mn} \delta_{ij} \right) \right] \quad (7)$$

$$\begin{aligned} \Phi_{ij,2} = & -C_{r1} P a_{ij} + C_{r2} \rho k S_{ij} + C_{r3} \rho k S_{ij} \sqrt{a_{mn} a_{mn}} + \\ & + C_{r4} \rho k \left( a_{ik} S_{jk} + a_{jk} S_{ik} - \frac{2}{3} a_{kl} S_{kl} \delta_{ij} \right) + C_{r5} \rho k (a_{ik} \Omega_{jk} + a_{jk} \Omega_{ik}) \end{aligned} \quad (8)$$

where

$$a_{ij} = \frac{\overline{u_i u_j}}{k} - \frac{2}{3} \delta_{ij}, \quad S_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right), \quad \Omega_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} - \frac{\partial u_j}{\partial x_i} \right)$$

The production due to the buoyancy is neglected in this study.

– the energy equation

$$\frac{\partial T}{\partial t} + U_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i} \left( a \frac{\partial T}{\partial x_i} - \overline{u_i \theta} \right) \quad \text{where} \quad -\overline{\rho u_i \theta} = \alpha_t \frac{\partial T}{\partial x_i} \quad (9)$$

By analogy with molecular transport, the turbulent Prandtl number for energy transport equation can be deduced by:

$$\alpha_t = \frac{v_t}{\text{Pr}_t} \quad (10)$$

The model constants are listed in tab. 1.

**Table 1. Reynolds stress constant model**

$C_{\epsilon 1}$	$C_{\epsilon 2}$	$C_\mu$	$C_{s1}$	$C_{s2}$	$C_{r1}$	$C_{r2}$	$C_{r3}$	$C_{r4}$	$C_{r5}$	$\text{Pr}_t$
1.44	1.92	0.09	1.7	-1.05	0.9	0.8	0.65	0.625	0.2	0.85

### Wall function approach

The wall treatment boundary conditions were achieved using the wall function technique of Launder and Spalding [28] that combine the viscosity affected region (viscous sub-layer) between the wall and the fully turbulent region, and were most widely used for industrial flows. Reynolds analogy between momentum and energy transport equation gives a similar logarithmic law for mean temperature.

The dimensionless variables of distance and velocity are defined as:

$$U_p^+ = \frac{U_p}{U_\tau}; \quad y_p^+ = \frac{\partial U_\tau y_p}{\mu} \quad \text{where} \quad U_\tau = \sqrt{\frac{\tau_p}{\rho}} \quad \text{and} \quad \tau_p = \rho \frac{\sqrt[4]{C_p} \sqrt{k_p} U_p}{U_p^+}$$

The index  $p$  means the value at the first point of grid close to the wall.

In the case of impinging jets, it is recommended to use near the wall the non-equilibrium wall function proposed by Kim and Choudhury [29].

### Geometry and boundary conditions

The jet exit velocity is imposed to a given Reynolds numbers. The length of the pipe is of 24 nozzle diameter ( $L_d = 24d$ ) for all configurations in order to obtain the fully developed flow at the exit. The nozzle diameter  $d$  is of 3.7 cm. The cavity radius  $R_{\text{cav}}$  is of  $3.92d$ . Tridimensional computations were achieved over the half of the configuration domain in order to spare calculation time. Four types boundaries conditions are imposed.

*Air jet exit:* The conditions at the duct inlet boundary are important in predicting the centerline velocity and shear stress. Constant values were considered for this boundary, as:

$$U = U_0, \quad V = 0, \quad T = T_0, \quad k_{in} = IU_0^2 \quad \text{and} \quad \epsilon_{in} = \frac{k_{in}^{1.5}}{0.15d} \quad \text{where} \quad I = 10^{-4}$$

*The wall boundary:* The velocity components are set to zero for each wall. Constant temperature  $T_w > T_0$  was imposed for each cavity walls and the duct wall is considered adiabatic.

*Air exit boundary:* The OUTFLOW (fully developed) boundary conditions were used. Constant relative static pressure kept at atmospheric level at this boundary was imposed;

*The symmetry:* This boundary condition is imposed in ( $y$ - $z$ ) plane ( $x = 0$ ) (fig. 2). The perpendicular velocity and Reynolds stresses components and the gradient of the all other dependent variables are set to zero.

### Modeling parameters

The numerical thermal and flow control parameters used in this study are summarized in tab. 2. The validation was achieved according to experiments data for the case  $L_f = 8d$ . The numerical predictions were performed in order to investigate the effect of the impinging distance  $L_f$  on the flow structure and heat transfer.



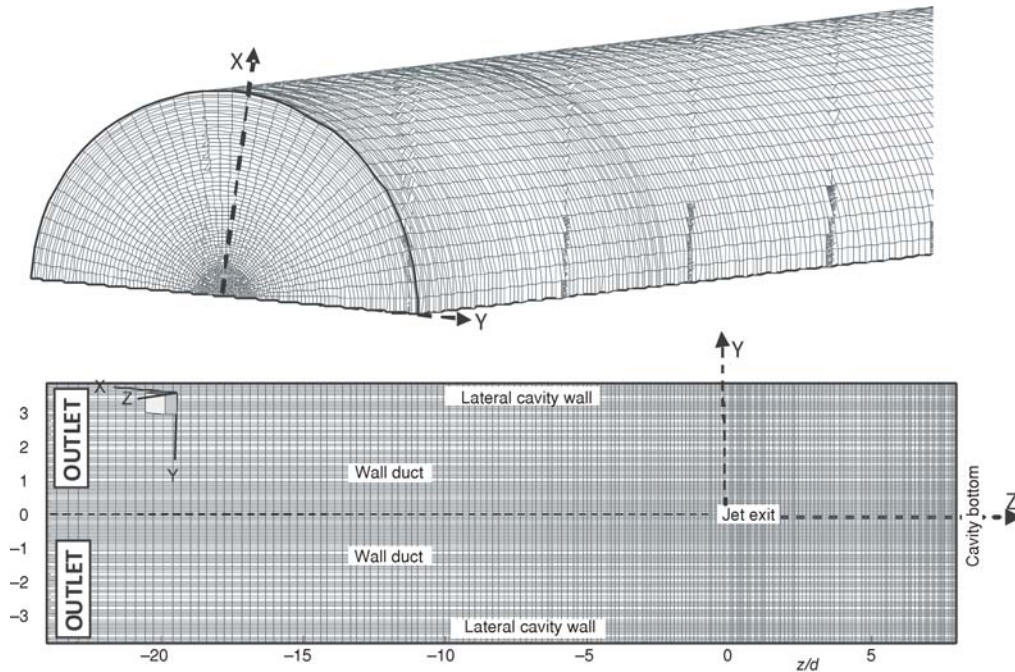
Figure 2. Geometry and a typical mesh ( $L_f = 8d$ ) 3-D view and symmetry plane

Table 2. The flow control parameters

$R_{cav}$	$d$	$L_d$	$L_f$	Re	$T_0$	$T_w$
14.5 cm	3.7 cm	$24d$	$2d \leq L_f \leq 30d$	$2 \cdot 10^4 \leq Re \leq 1 \cdot 10^5$	300 K	360 K

The equations for the mean and turbulent quantities are transformed by transport equations composed of convection, diffusion and source terms. They are discretized using finite volume technique on collocated non-uniform meshes. The convection and diffusion terms were discretized by means of POWER LAW schemes for all variables ( $U$ ,  $V$ ,  $T$ ,  $k$ ,  $u_i u_j$ , and  $\varepsilon$ ); while for the pressure the second-order scheme was used. The pressure velocity coupling is achieved using SIMPLEC algorithm. Usually, the source terms of all equations are linearized to insure the stability of the solution. The non-uniform structured grids were used. Refinements before and after the nozzle exit were managed in order to describe the entrainment accurately. Sufficiently fine grids were used near the cavity and duct walls to predict very high gradient of variables prevailed in the viscous sub-layer (fig. 2). The uncertainties in numerical results were evaluated by checking the convergence through mesh refinement calculations. Calculations show that for example for the case  $L_f = 8d$  (tab. 3); the number of nodes used in case (c) were adequate, since no difference was noted when compared with case (d). Further mesh refinement and numerical tests were carried out to check the stability of the solution

Table 3. Different grids ( $L_f = 8d$ )

Test	(a)	(b)	(c)	(d)
Nodes	231557	350003	454217	558886

## Results and discussion

### Dynamic field

The present work has described the behavior of a confined jet impinging a heated cylindrical cavity. Both, the velocity and the thermal fields are investigated to evidence the effect of the jet interaction on the cavity. The numerical simulations were carried out for Reynolds number ranges from  $2 \cdot 10^4$  to  $10^5$  and different impinging distances (jet exit nozzle-cavity bottom  $L_f$ ) ranging from  $2d$  to  $30d$ . For the validation, the reduced axial velocity prediction was compared to experimental values of Tani and Komatsu [30] determined on behalf of the same conditions (fig. 3). Good agreement with the experiment is obtained for three impinging distances  $L_f$  ( $L_f = 4d$ ,  $L_f = 8d$ , and  $L_f = 12d$ ).

The evolution of the axial component of the velocity vector (fig. 4) was compared to its corresponding experimental values of Benaissa [22] for the case  $L_f = 8d$ . Four cross-sections of the cavity were considered:  $z/d = 0.08$ ,  $1.89$ ,  $4.06$ , and  $6.48$ . It is noticed that the numerical predictions are in good agreement with the experimental data. The evolution of the Reynolds shear stress  $\overline{uv}$  and  $u^2$  according to the reduced radial distance  $\eta$  are also compared to corre-

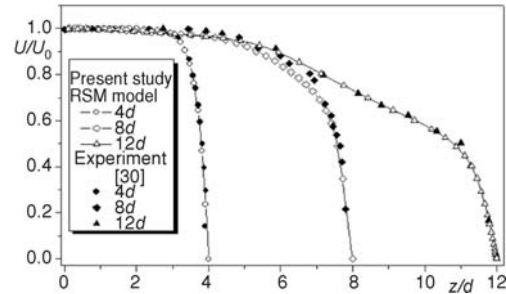


Figure 3. Reduced axial velocity profile along the jet axis

Figure 4. Radial evolution of the axial velocity at different downstream sections  $L_f = 8d$

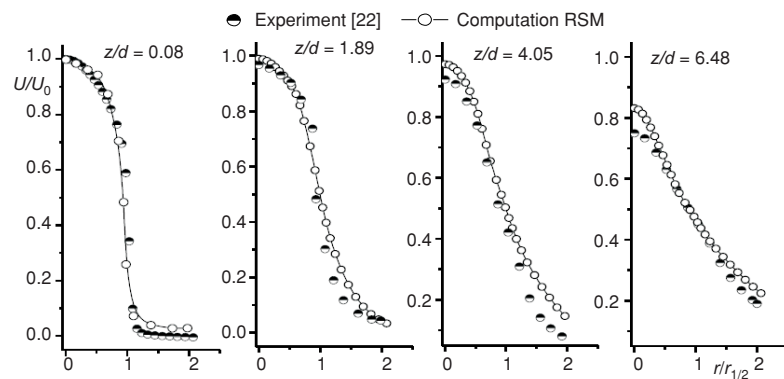
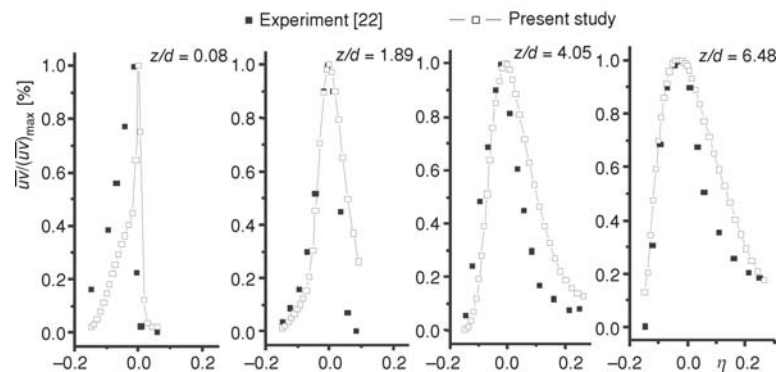
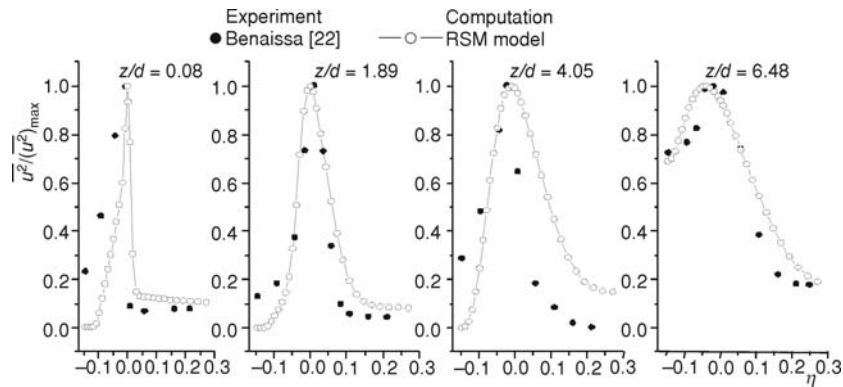


Figure 5. Radial evolution of component of Reynolds stress at different cross-sections  $L_f = 8d$  and  $Re = 37400$



sponding experimental data of Benaissa, [22] (figs. 5 and 6). Maximum value are reached around  $\eta = 0$ . These first comparisons indicate that the simulation for the case  $L_f = 8d$  will predicts well this flow. Minor discrepancies are observed between the two predictions in the areas of reverse flow, which are caused by hot-wire measurements that are not re-commended for re-circulated flow.



**Figure 6. Radial evolution of component of Reynolds stress for several cross-sections  $L_f = 8d$  and  $Re = 37400$**

The influence of the impinging distance  $L_f$  on the flow structure interaction is described by the magnitude of the vorticity contours computed using the Reynolds stress model (fig. 7). The contours plots show that the interaction of the jet into the cavity produces two deterministic eddies on each side of the jet with opposite swirl. For short impinging distances  $2d \leq L_f \leq 8d$ , the jet impinges the cavity bottom and returns towards the exit of the cavity. The reverse flow is then split into two parts:

- the outflow which leaves the cavity at low velocity in the opposite direction of the jet, and
- an entrained flow is sucked by the jet, leads the formation of a stable toroidal eddy near the cavity bottom. This zone is characterized by a weak mean velocity, high turbulence energy and a negative pressure due to the Coanda effect [11, 12].

The principal jet is also narrowed. For  $L_f \leq 4d$ , the center of this eddy remained identical for several impinging distance. The case of  $L_f = 8d$  is in good agreement with experimental visualisation of Benaissa [22]. For  $L_f = 20d$ , appears close to the cavity bottom wall a secondary toroidal vortex which increases in volume when  $L_f$  increases. The size of the first vortex remains constant for  $8d \leq L_f \leq 35d$ . The size of the first vortex remains constant. The space between this later and the cavity bottom is a seat for the generation of secondary eddies. For very large impinging distance  $L_f \leq 24d$ , third contra rotating toroidal eddy (swirl) appears in the corner of the cavity, this swirl becomes more obvious and larger when the impinging distance increases  $L_f \leq 30d$ . The maximum values of vorticity are attained around the jet exit area. The Reynolds number does not have any influence on the global structure of the vorticity, it has only an effect on the magnitude as expected.

Figure 8 shows the influence of the impinging distance  $L_f$  on the axial velocity profiles across several perpendicular sections ( $\xi$ : 0.01, 0.25, 0.5 and 0.8) at  $Re = 37400$ . For several impinging distance, the reverse flow was evidenced by negative values of axial velocity near the lateral cavity wall.



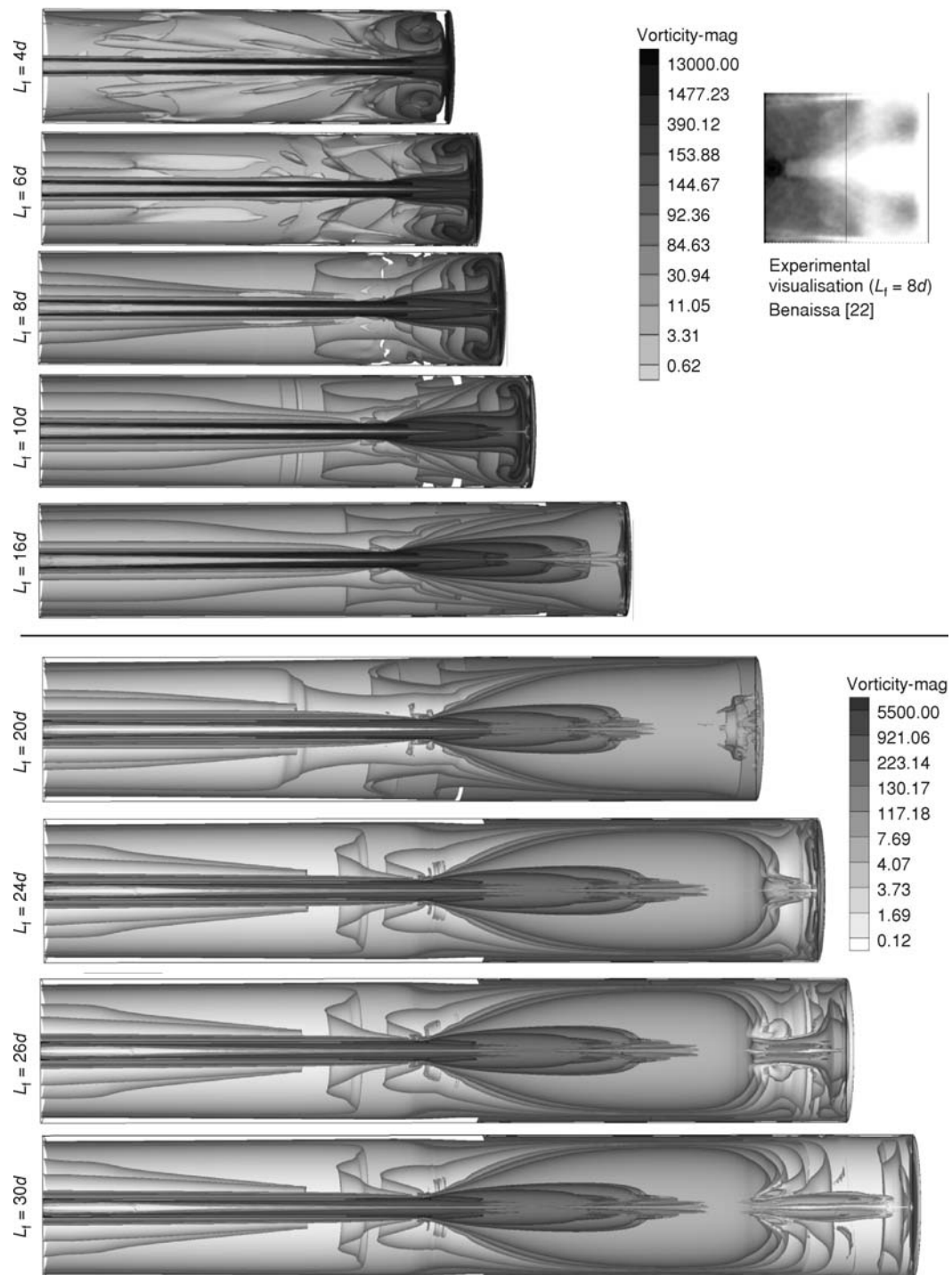


Figure 7. Effect of the impinging distance  $L_f$  on vorticity magnitude in  $s^{-1}$  ( $Re = 37400$ )

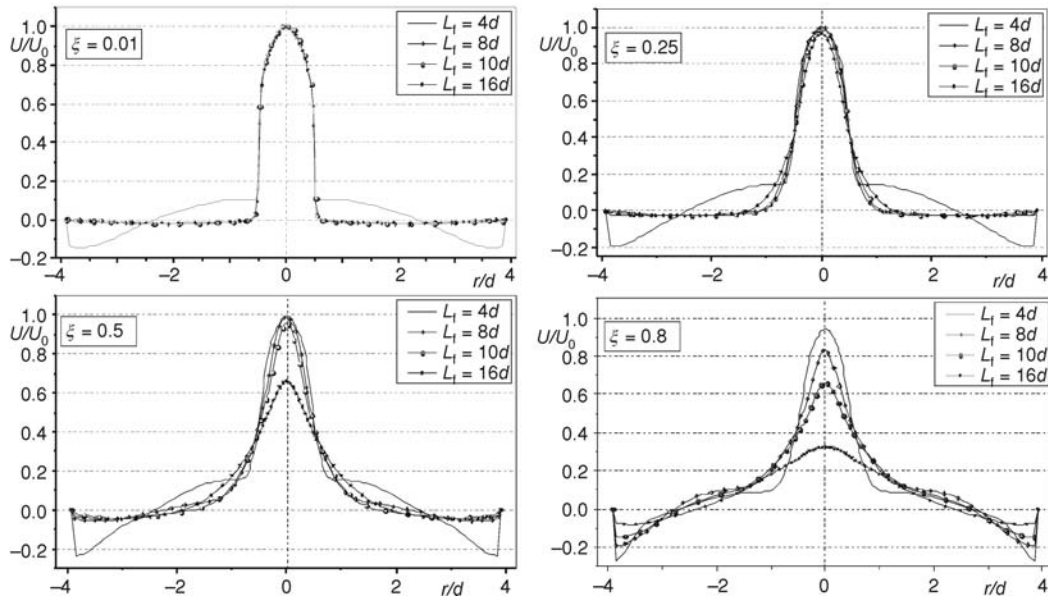


Figure 8. Effect of the impinging distance  $L_f$  on axial velocity ( $Re = 37400$ )

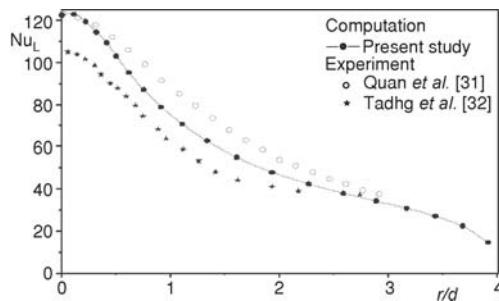


Figure 9. Local Nusselt number on the cavity bottom

### Thermal study

For the thermal study, the effects of the impinging distance  $L_f$  and Reynolds number on the local Nusselt number  $Nu_L$ , were investigated. The local Nusselt number is examined for the two cavity walls (bottom face and lateral face).

In order to validate the numerical method for heat transfer predictions, comparisons were performed with experiment data of Quan, [31] and O'Donovan and Murray [32] of the case of a round jet impinging perpendicularly a hot wall. A good agreement is obtained qualitatively with the experiment (fig. 9). Minor differences are

noticeable between the curve shapes, which can be explained by the effect of confinement of lateral cavity wall.

Figures 10(a) and (b) illustrate, respectively, the local Nusselt number evolution along the cavity bottom walls and cavity lateral, according to the Reynolds number, for the case  $L = 8d$ . The visual analysis of these two figures shows that the increase in the Reynolds number led to the increase of the heat transfer. This evolution is completely predictable since the increase of Reynolds number, hence enhances the heat transfer. Particularly on the bottom cavity wall, optimal and minimum Nusselt number values corresponding to stagnation and re-attachment points, respectively. The influence of the impinging distance was evidenced through temperature profiles (fig. 11). For big impinging distance ( $L_f = 16d$ ), a homogeneous heating of flow is obtained. Minimum temperature values are observed at jet exit and maximal values are reached at lateral cavity wall.

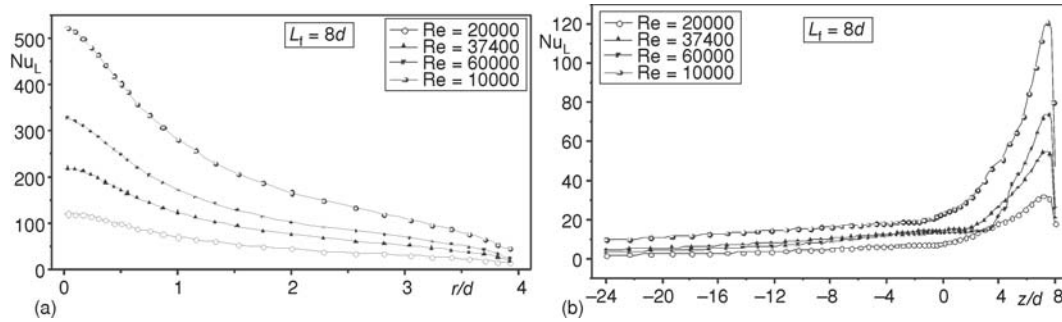


Figure 10. Effect of Reynolds number on Nusselt number ( $L_f = 8d$ )

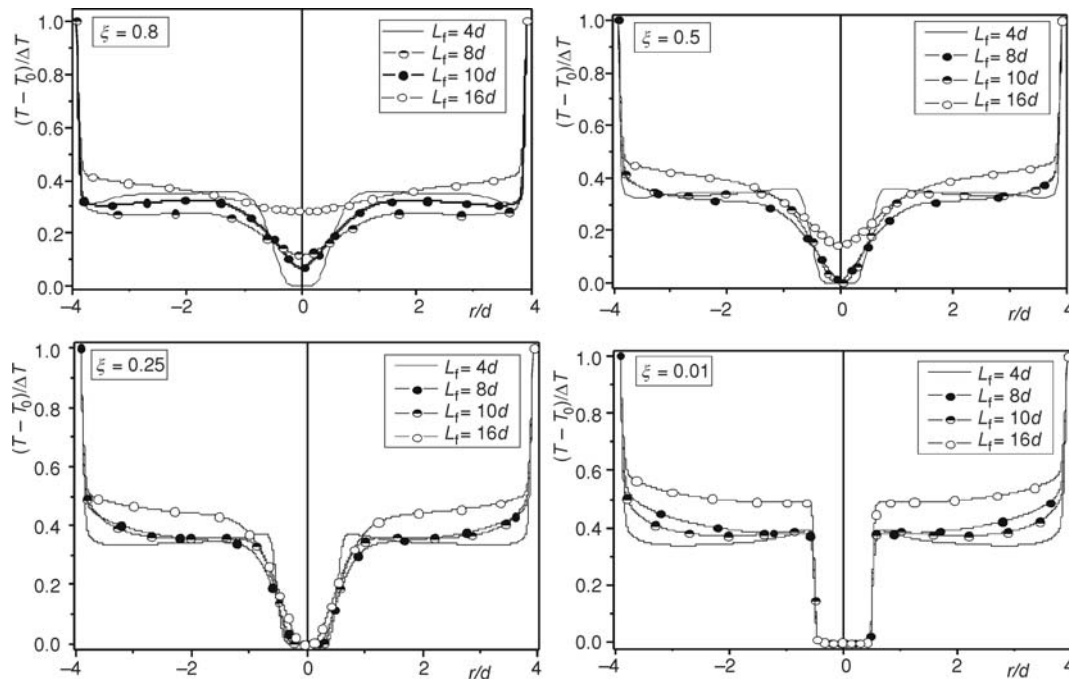


Figure 11. Effect of the impinging distance  $L_f$  on temperature ( $Re = 37400$ )

Figure 12 shows the effect of the impinging distance on local Nusselt number along each cavity wall. The first peak (maximum Nusselt number) corresponds to the stagnation point and decrease to  $r = 0.5d$  matching to the section jet this maximum is due to the existence of a quasi-uniform velocity profile around the stagnation point. For a largest impinging distance, this maximum disappears caused by secondary eddies located in the cavity bottom region, creating an air cushion preventing the cooling of the cavity hot walls by the jet (see fig. 7).

Finally, the numerical results of the average Nusselt numbers evolution according to the impinging distance, fig. 13, was expressed for a several Reynolds numbers by the following correlation:

$$\overline{Nu} = 4.6017 + 10^{-3} (798 - 0.3123L_f) Re^{0.883}$$

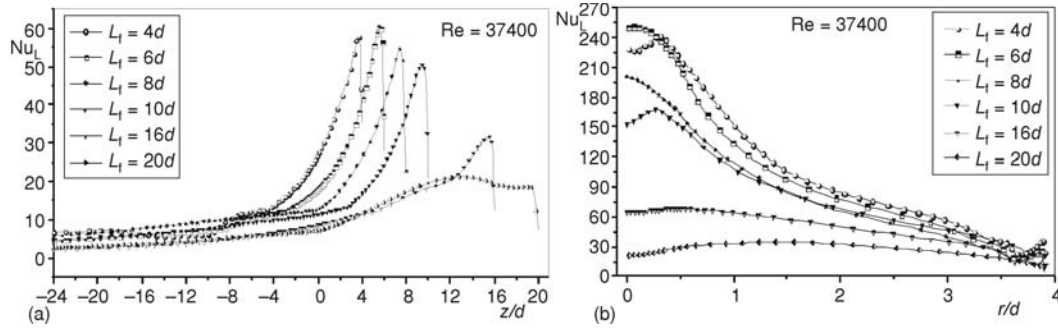


Figure 12. Effect of impinging distance on local Nusselt number (Re= 37400); (a) side wall, (b) bottom wall

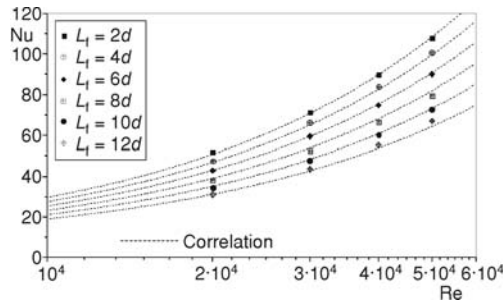


Figure 13. Effect of impinging distance on mean Nusselt number (Re= 37400)

This correlation is valid for  $L_f \leq 12d$ . The case of large impinging distance ( $L_f > 12d$ ), induce secondary eddies in the cavity bottom area, which do not characterise heat transfer of jet to wall impingement

## Conclusions

The interaction of turbulent axisymmetric jet issuing into a cylindrical hot cavity was investigated numerically using Reynolds stress turbulence model. A parametric study was conducted by varying the impinging distance for several Reynolds numbers. The fluid structure was determined for different impinging distances.

- The flow is turbulent and steady in average as confirmed by previous study of the same configuration.
- The flow structure was independent of the Reynolds number (turbulent).
- The velocity profiles agree with experimental data for the case  $L_f = 4d, 8d$ , and  $12d$ .
- A reverse flow occurs within the cavity corner and near the cavity bottom. It interacted with the main flow producing a toroidal swirl close to the jet exit.

The increase of Reynolds number enhances the heat transfer. The results are compared with an experimental available case of a jet, impinging normally onto a flat plate. It is found, that for large impinging distance, two secondary vortices weaken the heat transfer. This weakening is due to the existence of the secondary eddies in the cavity bottom area. The computed heat transfer data are believed to be in good agreement with previous works. However, the experimental verification of the results is strongly recommended and will be performed in the future. The evolution of average Nusselt number is correlated according to the impinging distance and Reynolds number  $\overline{Nu} = f(L_f, Re)$ .

## Nomenclature

$a$  – thermal diffusivity, [ $m^2s^{-1}$ ]  
 $d$  – jet exit diameter, [m]  
 $k$  – kinetic energy of turbulence, [ $m^2s^{-2}$ ]  
 $L_d$  – length of the pipe, [m]  
 $L_f$  – distances jet exit nozzle-cavity bottom, [m]

$\overline{Nu}$  – average Nusselt number  
 $[= \int_0^{R_{cav}} (2\pi r Nu) dr / \pi (R_{cav})^2]$ , [-]  
 $Nu_L$  – local Nusselt number ( $= q_w d / \lambda \Delta T$ ), [-]  
 $P$  – mean pressure, [Pa]  
 $q_w$  – heat flux per unit area, [ $Wm^{-2}$ ]

$Re$	– Reynolds number ( $= U_i d_{jet}/\nu$ ), [–]
$R_{cav}$	– cavity radius, [m]
$R_{jet}$	– jet exit radius, [m]
$r_{1/2}$	– radial position corresponding to $U_{max}/2$ , [m]
$T$	– temperature, [K]
$U_0$	– jet inlet velocity, [ $ms^{-1}$ ]
$U^+$	– non-dimensional velocity, [–]
$U, V$	– velocity components, [ $ms^{-1}$ ]
$U_{max}$	– maximum mean velocity, [ $ms^{-1}$ ]
$\overline{u_i u_j}$	– Reynolds stress component, [ $m^2 s^{-2}$ ]
$y^+$	– non-dimensional distance, [–]

#### Greek symbols

$\varepsilon$	– dissipation rate of turbulent energy, [ $m^2 s^{-3}$ ]
$\eta$	– dimensionless axial distance ( $= (r - R_{jet})/(R_{cav} - R_{jet})$ ), [–]
$\theta$	– fluctuating temperature, [K]
$I$	– turbulence intensity, [–]
$\lambda$	– thermal conductivity of air, [ $W m^{-2} K^{-1}$ ]
$\nu$	– kinematic molecular viscosity, [ $m^2 s^{-1}$ ]
$\nu_t$	– kinematic turbulent viscosity, [ $m^2 s^{-1}$ ]
$\xi$	– dimensionless axial distance ( $= (z - L_d)/L_f$ )
$\rho$	– air density, [ $kg m^{-3}$ ]

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