NUMERICAL SIMULATION OF THE OSCILATORY FLOW THROUGH AN AXIALLY FED RADIAL DIFFUSER

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Abstract. *The raddial difuser geometry has been used as a simplified model for flow investigations in many industrial devices, mainly for simulating the flow through refrigeration compressor valves. A lot of effort has been undertaken to understand all the flow patterns associated with this geometry, and the fast computational advance allowed that numerical simulations could capture new flow patterns. In this work a numerical simulation of the unsteady flow through a radial diffuser is carried out in order to investigate the transitional flow pattern. The governing equations are discretized using the Finite Volume Method. The pressure-velocity coupling problem is solved with the SIMPLEC – Semi-Implicit Method for Pressure Linked Equations Consistent algorithm applied to a staggered mesh for the velocity. The algebraic equations system are solved by the SOR – Successive Over-Relaxation algorithm for the velocities and energy, while the pressure correction equation is solved by the SIP – Strongly Implicit Procedure algorithm. The Power-Law scheme is used for interpolating the difusive/advective terms. Local Nusselt number and dimensionless pressure profiles on the frontal disk surface are obtained for different Reynolds numbers.The results are confronted to experimental results showing good agreement. It was observed that the vortexes released at the inlet of the diffuser region are advected in the flow direction, but they are dissipated before reaching the exit of the diffuser. It was possible to find out that for 1900<Re<2000 the flow starts oscillating, becoming an asymmetric unsteady flow, which characterizes the transition from laminar to turbulent flow.*

Keywords*: Radial diffuser, Heat Transfer, Unsteady Flow, Transitional Flow.*

1. INTRODUCTION

The radial flow is investigated by the scientific community due to its presence in several engineering projects. Among them, the circular airfoil of airplanes can be cited. This kind of flow is also present in radial or mixed flow machines, in hydrocyclones used to separate dispersed elements in liquid medium, in aerostatic radial bearings, where the pressurized air is forced through a feeding orifice to the radial flow, and in reed type valves of hermetic compressors used in refrigeration systems, which is the main motivation of this work.

In refrigeration compressors, the suction and discharge valves are responsible for the retention and passage of the fluid flow from the suction chamber to the cylinder chamber, and from the cylinder chamber to the discharge chamber; respectively. Designers of valve system seek for valves with fast response, low pressure losses, and reduced gas return in order to increase the compressor efficiency. As the opening and closing of the valves are caused by the force produced by the refrigerant flow, the understanding of the flow through the valve is of fundamental importance. The numerical simulation of the flow is an efficient method to perform this task.

Due to the complex geometry of this type of valve, simplified geometries, as the radial diffuser shown in Fig. 1, are usually used to model the valve.

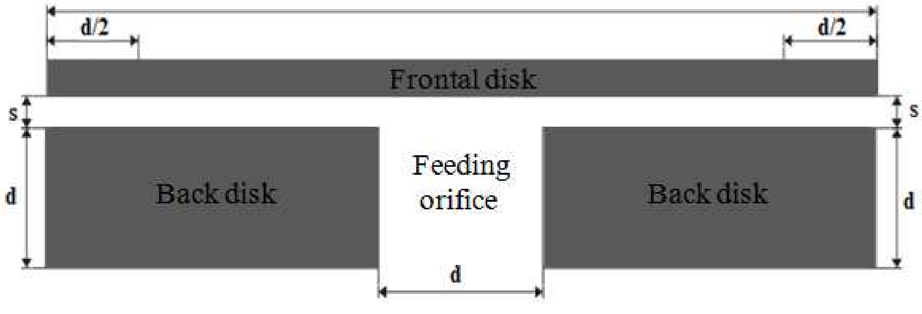


Figure 1. Radial Diffuser geometry.

The flow pattern in radial diffusers depends on the Reynolds number and the gap between disks, s. In general, for lower values of Reynolds number and gap, the viscous force of the flow between disks, usually called diffuser region, prevails and the flow is stable. For higher values of the Reynolds number and gap, the hydrodynamic boundary layer tends to separate from the inner surface of the feeding orifice, generating a recirculation fluid region on the diffuser entrance. For further Reynolds number increase, the flow becomes unstable and an unsteady oscillatory flow pattern arises, characterizing a transitional flow.

Souto (2002) has presented an extensive review on radial diffuser flows with applications in compressor valves. Works on analytical solutions for laminar, incompressible, and steady flows have been performed by Woolard (1957), Livesey (1960), Savage (1964) and Killmann (1972), while Ishizawa *et al.* (1987) have presented analytical solution for the incompressible, laminar and unsteady flows. Numerical solution for laminar and incompressible flows has been accomplished by Hayashi *et al.* (1975), Raal (1978), Piechna and Meier (1986), Ferreira *et al.* (1989), Deschamps *et al.* (1989), Ferreira *et al.* (1989), Langer *et al.* (1990), Gasche (1992) and Possamai *et al.* (1995). On the other hand, numerical solutions for turbulent and incompressible flows have been obtained by Deschamps *et al.* (1988) and Deschamps *et al.* (1996). Experimental works on the subject have been developed by Jackson and Simmons (1965), Wark and Foss (1984), Ferreira and Driessen (1986), Tabatabai and Pollard (1987), Ervin *et al.* (1989), and Gasche (1992). Some researchers also have obtained numerical solutions for laminar and incompressible flows including the frontal disk dynamics: Matos *et al.* (1999), Matos *et al.* (2000), Matos *et al.* (2001), and Salinas-Casanova (2001).

Pilichi (1990) investigated the radial diffuser flow considering the heat transfer. Numerical results for the local Nusselt number on the frontal disk surface were obtained using the Finite Volume Method described for discretizing the governing equations. The Power-Law scheme was used for interpolating the advective/diffusive terms and the SIMPLER algorithm was used for the pressure-velocity coupling. Eexperimental results were also obtained using the naphthalene sublimation technique. For lower Reynolds numbers the results agreed well, validating the numerical method for the steady laminar flow. For these cases, a single peak on the local Nusselt number profile was observed at the diffuser entrance. For higher Reynolds numbers, a secondary peak on the Nusselt number profile was observed in the experimental results, which was not observed in the numerical results. The author commented that this secondary peak could be associated with the flow instabilities.

A year later, Langer (1991) studied several characteristics of radial flow, searching for explanations for those flow instabilities. The results led to two important conclusions: the invalidation of the geometric symmetry hypothesis usually used for many researches to analyze this kind of flow, and the existence of instabilities and bifurcations due to the reverse pressure gradient present in the radial flow.

Peters (1994) studied self induced bifurcations and oscillations in radial diffuser flow. In this work, radial diffusers axially and radially feeded were simulated by using several numerical methods with high order interpolation schemes. The data confirmed the results obtained by Langer (1991) for the first bifurcation point, and a second point was identified for a transition between an asymmetric steady to an asymmetric unsteady flow (Hopf bifurcation). The local Nusselt number profile on the frontal disk surface was also obtained, seeking for the secondary peak obtained experimentally by Pilichi (1990). The results agreed well, despite some deviation on the position of the second peak.

The present work presents an analysis of the oscillatory flow in radial diffuser axially fed, in order to investigate the transition of the steady to unsteady flow pattern.

2. NUMERICAL METHOD

The governing equations of the incompressible Newtonian flow through the radial diffuser (mass conservation, momentum and energy equations) are:

$$\vec{\nabla} \bullet \vec{V} = 0 \tag{1}$$

$$\rho\left[\frac{\partial \vec{V}}{\partial t} + \vec{V} \bullet \vec{\nabla}\vec{V}\right] = -\vec{\nabla}p + \vec{\nabla} \bullet \left[\mu\left(\vec{\nabla}\vec{V} + \vec{\nabla}^T\vec{V}\right)\right] \tag{2}$$

$$\rho\left[\frac{\partial T}{\partial t} + \vec{\nabla} \bullet (\vec{V}\,T)\right] = \frac{1}{C_p}\vec{\nabla} \bullet (k\vec{\nabla}T) \tag{3}$$

where ρ is the fluid density, μ is the dynamic viscosity, C_p is the specific heat at constant pressure, k is the thermal conductivity, p is the pressure, V is the velocity vector, and T is the temperature. The necessary boundary conditions to solve the problem are indicated in Fig. 2. The geometric parameters that characterize the model are D/d=3.15, d, and variable *s*.

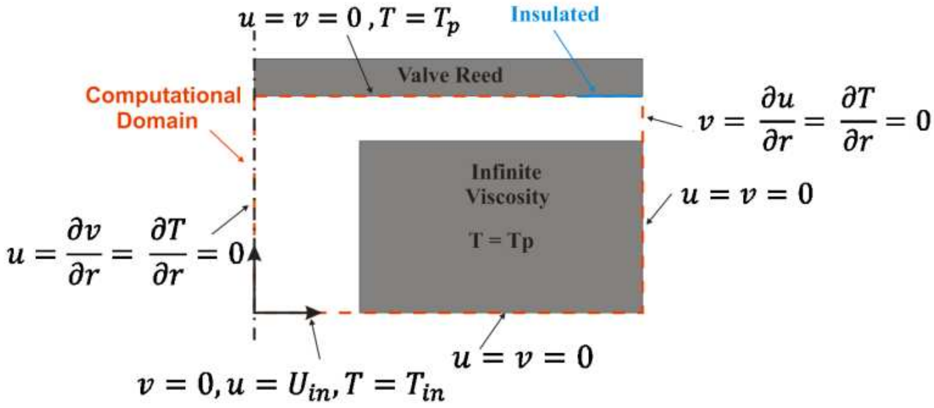


Figure 2. Computational domain and boundary conditions.

The governing equations and boundary conditions are discretized using the Finite Volume method, using the Euler (first order) discretization for the transient term. The pressure-velocity coupling is treated by the SIMPLEC – Semi-Implicit Method for Pressure Linked Equations Consistent algorithm, applied to a staggered mesh for velocity, while the algebraic equations systems are solved by the SOR – Successive Over-Relaxation algorithm for momentum and energy equations, and the SIP – Strongly Implicit Procedure algorithm for the pressure field. The Power-law scheme is used to interpolate the advective/diffusive terms of the Navier-Stokes equations.

A non-uniform mesh refined near the highest gradient regions and solid/fluid interface was used to generate all the results, totalizing 48,300 volumes in the entire domain (24,000 volumes in the diffuser region).

The dimensionless parameters characterizing the flow are the local Nusselt number, the Reynolds number and the dimensionless pressure on the frontal disk surface, defined by Eqs. 4, 5, and 6.

$$Nu_s = \frac{hs}{k} = \frac{q"s}{(T_p - T_{in})k} \tag{4}$$

$$Re_d = \frac{\rho \bar{U} d}{\mu} \tag{5}$$

$$P_{adm} = \frac{p}{\frac{1}{2}\rho \bar{U}^2} \tag{6}$$

where h is the convection heat transfer coefficient, k is the thermal conductivity of the fluid, T_p is the prescript temperature on the frontal disk surface, T_{in} is the fluid temperature at the entrance of the feeding orifice, $\bar{U}$ is the mean velocity at the feeding orifice entrance, μ is the absolute viscosity of the fluid, and p is the fluid pressure. In this work T_p=70°C and T_{in}=20°C were used.

3. RESULTS

Results for steady and unsteady flows were obtained for a gap between disks (s/d) equal to 0.07, the same gap used by Pilichi (1990) in his experiment. Numerical local Nusselt number profiles on the frontal disk surface were obtained for Reynolds number equal to 2727 and Prandtl number equal to 2.6. The results were confronted with the experimental data from Pilichi (1990) and the numerical results from Peters (1994) in order to validate the numerical results. The convergence criterion was based on the mass conservation residue, considering acceptable results for residues lower than 10^{-10}. A time-step equal to $2\ 10^{-4}$ and a total simulation time of 1000 seconds were used.

The instantaneously local Nusselt number profile was time integrated and confronted with experimental and numerical results from Pilichi (1990) and Peters (1994), respectively. As shown in Fig. 3, one can notice that the first peak at the diffuser entrance agrees well with the numerical data, but is larger than the experimental data. However, the second peak does not agree well neither with the numerical result nor with the experimental data.

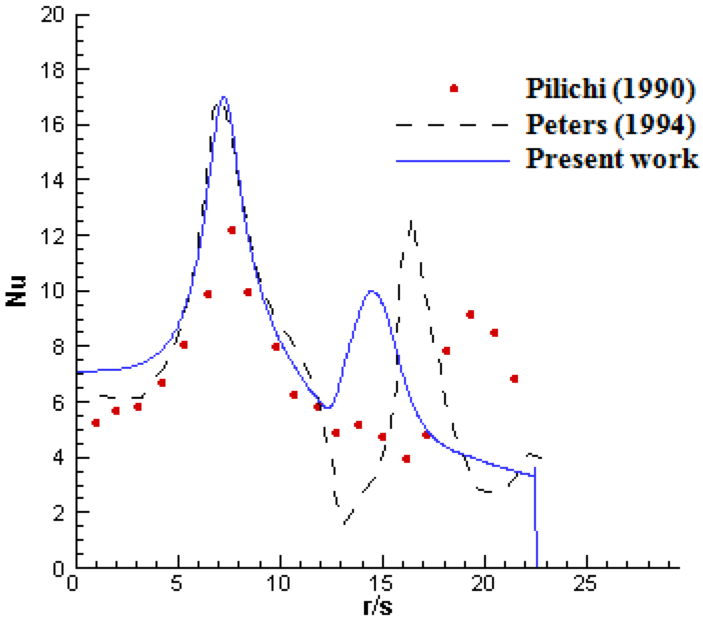
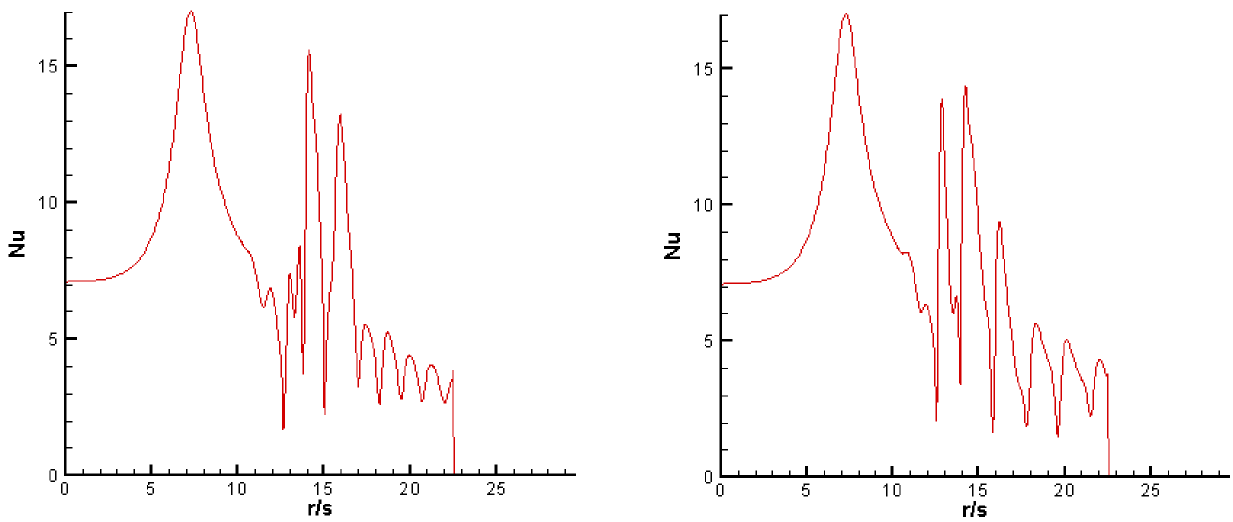


Figure 3. Local Nusselt number profiles on the frontal disk surface for Re=2727, Pr=2.6, and s/d=0.07.

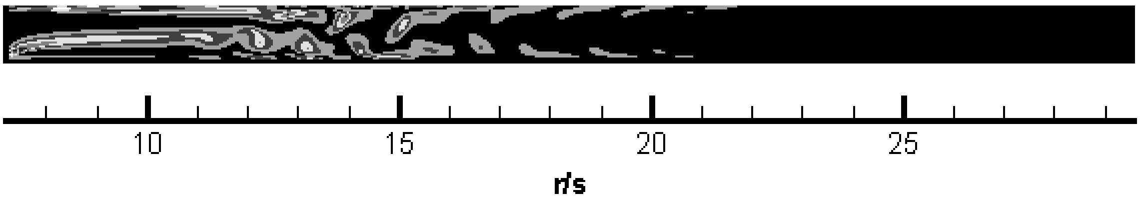
It should be emphasized that the numerical method has considerable influence on the results for this type of flow, as concluded by Peters (1994). Another possible explanation for the results deviation could be the experimental uncertainty, which is not addressed by Pilichi (1990). The naphthalene sublimation technique used by Pilichi allows the calculation only of the time average Nusselt number by measuring the thickness of a thin naphthalene layer. As the flow does not reach the steady state regime, the instantaneous naphthalene mass transfer modifies the local layer thickness, which can modify the flow over the surface. Therefore, the geometry of the real mass transfer problem is different from the geometry of the numerical heat transfer problem. This difference can be another source of discrepancy between the experimental and numerical results.

Figure 4 shows the instantaneous local Nusselt number profile for the same case in two different times. It can be noticed that the first peak, which is caused by the high acceleration and curvature of the flow at the entrance of the diffuser, is stationary and well defined. However, there are several Nusselt peaks in the diffuser region. These peaks are due to the radial movement of consecutive vortexes detached from the entrance of the diffuser, which can be seen through the isovorticity field in Fig. 5. Therefore, the unique second peak measured by Pilichi is in fact a time average value.

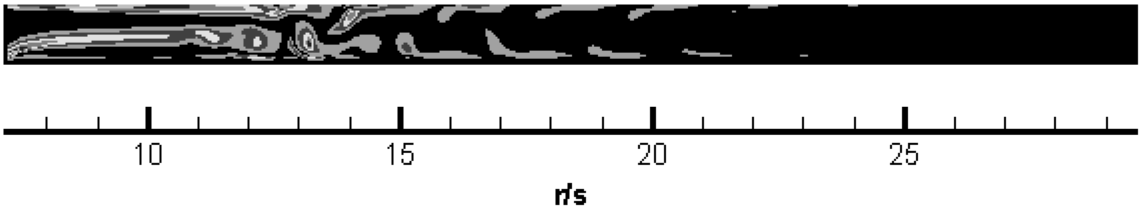


(a) t = 450 s (b) t = 900 s

Figure 4. Instantaneous local Nusselt number profiles for Re=2727.



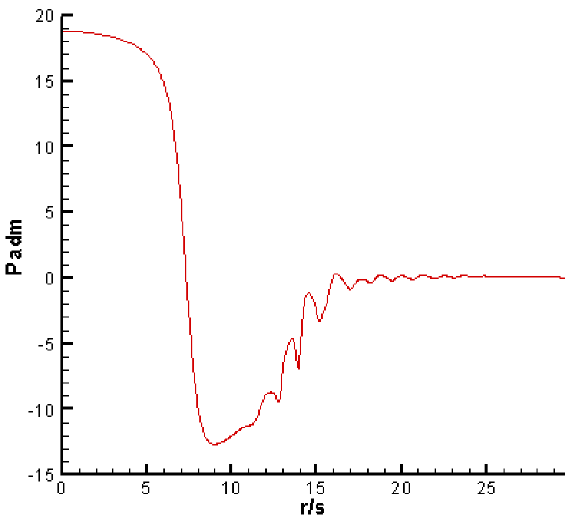
(a) t = 450 s



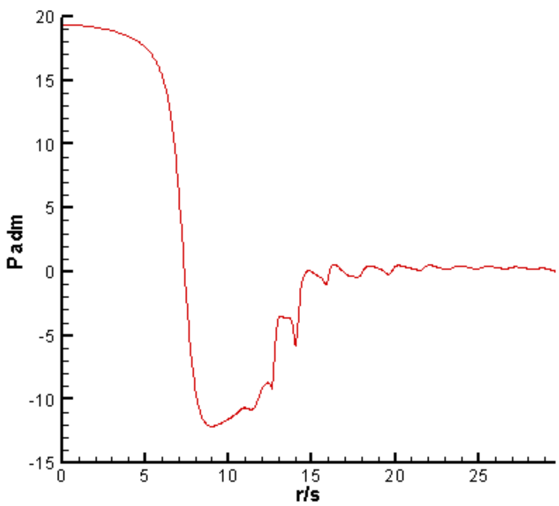
(b) t = 900 s

Figure 5. Isovorticity fields for Re=2727 and s/d=0.07.

Figure 6 shows the dimensionless pressure profile on the frontal disk surface at two different times for the same case. The region of the highest values is due to the fluid stagnation fluid on the frontal disk, before entering the diffuser region. The flow acceleration at the diffuser entrance produces a large pressure decrease, which is recovered as the flow decelerates due to de augmentation of the cross section area. The pressure oscillations in the diffuser region are caused by the vortex detachment at the entrance, which makes the entire profile oscillates with time.



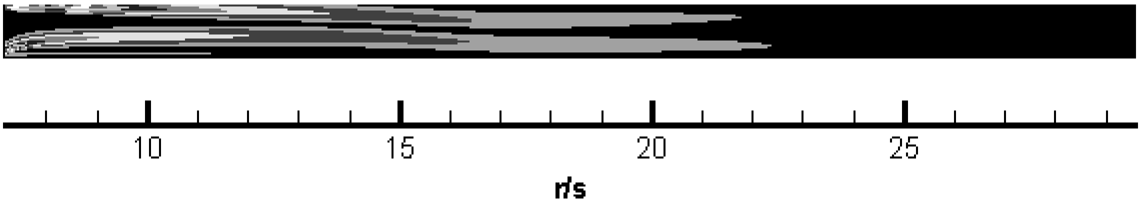
(a) t = 450 s



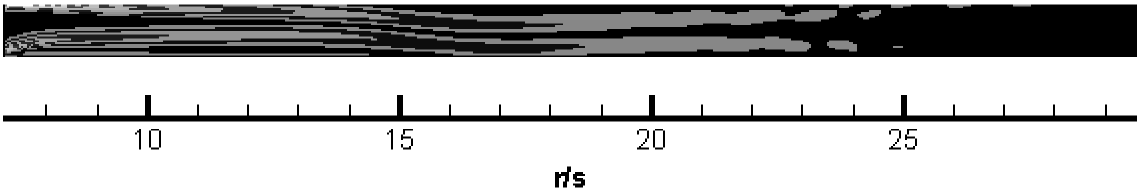
(b) t = 900 s

Figure 6. Dimensionless pressure profiles for Re=2727.

The problem was also solved for Reynolds number equal to 1900, 2000, 2100 and 2200, considering s/d=0.07. For Re=1900 the flow reached a steady state regime after about 750 seconds, as shown in Fig. 7. In the initial time, before reaching the steady state regime, there has been observed some oscillatory flows which are dumped along time, as predicted by Peters (1994). For Re=2000, the flow configuration is practically the same when compared to the Re=1900. However, there are small vortexes close to the diffuser exit. Figure 8 depicts small oscillations on the local Nusselt and dimensionless pressure profiles for both cases.



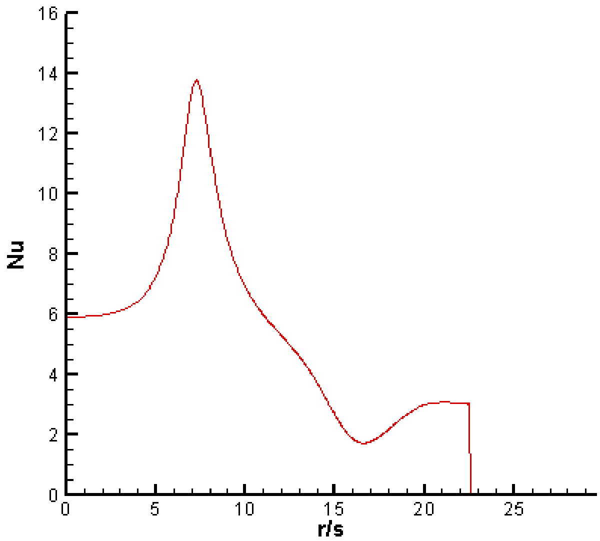
(a) Re = 1900; t = 900 s



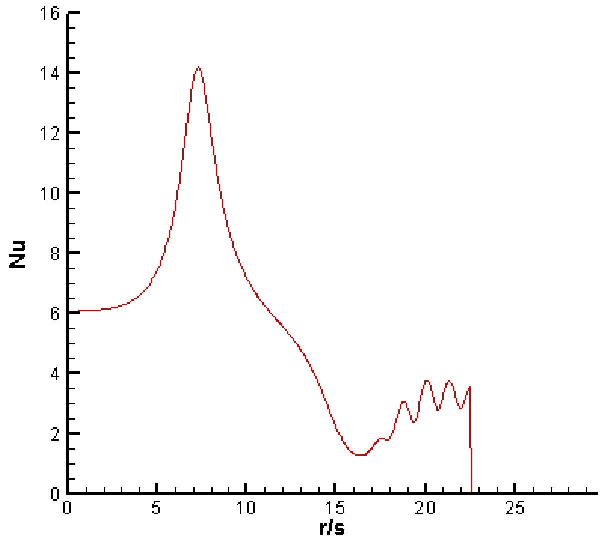
(b) Re = 2000; t = 900 s

Figure 7. Isovorticity fields: (a) Steady; (b)Unsteady

Analyzing the local Nusselt number profiles at the same time (t=900s), the flow characteristics are more evidenced. The oscillations verified in Fig. 8b are not observed in Fig. 8a. Based on these results, one can say that the transition from laminar to turbulent flow occurs for 1900 <Re<2000.



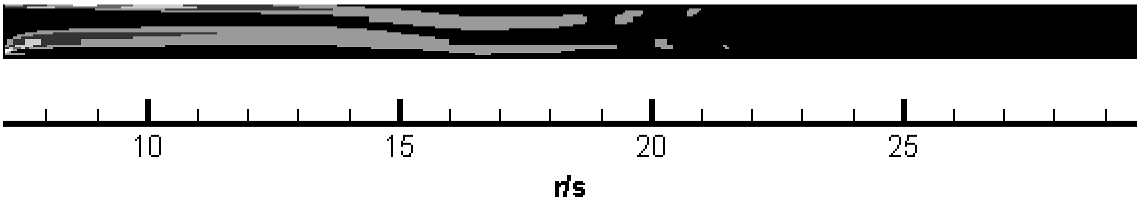
(a) Re = 1900;



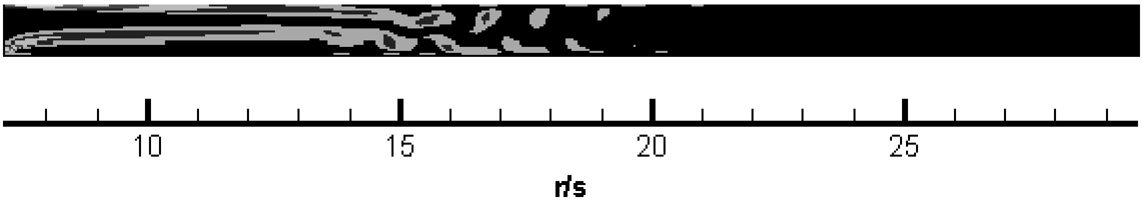
(b) Re = 2000;

Figure 8. Local Nusselt number profiles for t=900 s.

With Reynolds number augmentation, results show that the vortex detachment increases. From Figs. 9 and 10, which show the results for Re= 2100 and 2200, it is possible to verify that the vortex detachment increases for increasing Reynolds numbers.

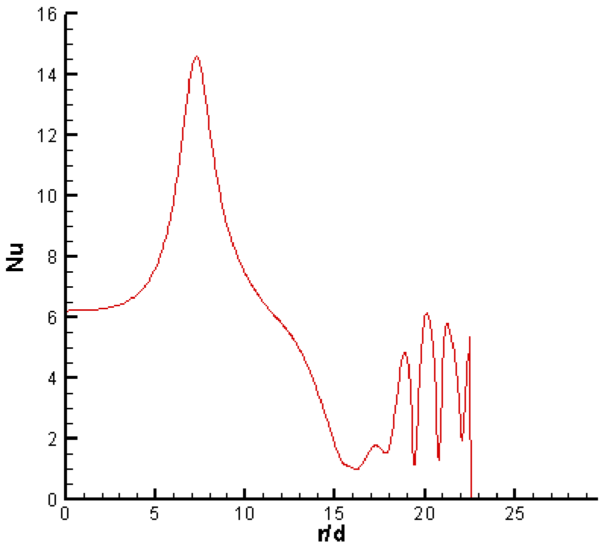


(a) Re = 2100;

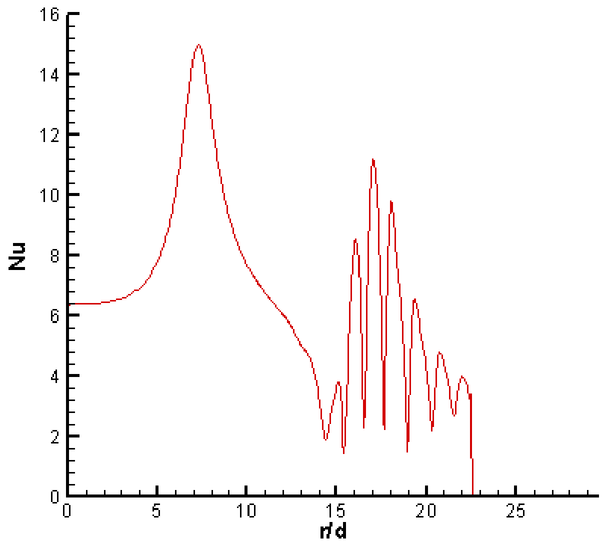


(b) Re = 2200;

Figure 9. Isovorticity fields for t=900 s.



(a) Re = 2100



(b) Re = 2200

Figure 10. Local Nusselt number profiles at t=900 s for a) Re=2100 and b) Re=2200 at=900 s.

4.CONCLUSIONS

The present work presented a numerical analysis of the incompressible flow through a radial diffuser axially fed, focusing the characterization of the transitional flow pattern. The numerical results were confronted with experimental data and numerical results for Re=2727 and s/d=0.07. The numerical method was able to detect two Nusselt peaks as obtained numerically by Peters (1994) and experimentally by Pilichi (1990).

Analyzing the pressure profile on the frontal disk surface, it was possible to find out that for 1900<Re<2000 the flow starts oscillating, becoming an asymmetric unsteady flow, which characterizes the transition from laminar to turbulent flow.

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