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Influence of the Heat Transfer on the Pressure Field in Radial Diffusers Flows

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ABSTRACT

The radial diffuser geometry has been largely used to analyze the flow through refrigeration compressor valves. The majority of the researchers have modeled the flow as isothermal. In fact, the valve system is heated during the compression process and the refrigerant gas flows through heated channels. In this work, the flow through the radial diffuser is model as non-isothermal in order to verify the influence of the heat transfer process on the pressure field on the frontal disk, which is an important result for simulating the behavior of the valve. The Finite Volume Method is used for discretizing the governing equations. The SIMPLEC-Semi-Implicit Method for Pressure-Linked Equations Consistent algorithm applied to a staggered mesh is used for solving the pressure-velocity coupling problem. The power-law scheme is employed as interpolation function for the convective-diffusive terms, and the algebraic equations systems are solved by the TDMA-Three Diagonal Matrix Algorithm. The flow behavior is analyzed for Reynolds number varying from 500 to 2000, dimensionless gap between the frontal (valve reed) and back (valve seat) disks ranging from 0.04 to 0.1, and three heat transfer configurations. The numerical results have shown that the overall pressure drop through the flow can be 21% higher when the heat transfer process is included in the flow modeling.

1. INTRODUCTION

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 usually found in this type of valve, simplified geometries have been used to represent the valve, particularly the radial diffuser, shown in Figure 1.

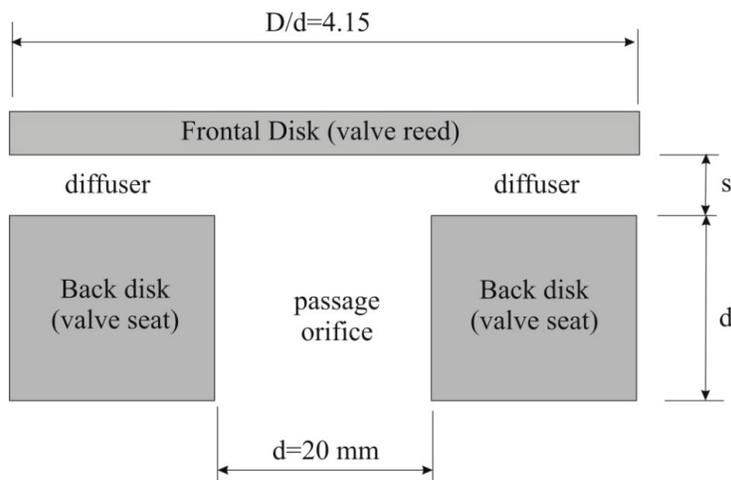


Figure 1. Radial Diffuser.

Most researchers have performed numerical simulations of the isothermal flow through the radial diffuser. The study of the heat transfer in this geometry is rare in the literature.

Souto (2002) has presented an extensive review on radial diffuser flows with applications in compressor valves. In this there are descriptions on analytical solutions for laminar, incompressible, steady and unsteady flows; numerical solution for incompressible flows, including laminar and turbulent regimes; and experimental works. All the works cited by the author have considered the flow as isothermal. Some other researchers also have obtained numerical solutions for isothermal, laminar and incompressible flows, but included the frontal disk dynamics (Lopes, 1994; Matos *et al.*, 1999; Matos *et al.*, 2000; Matos *et al.*, 2001; and Salinas-Casanova, 2001).

Pilichi (1990) is one of the few researchers who have investigated the flow in a radial diffuser including the heat transfer process. Results were obtained numerically and experimentally, using the naphthalene sublimation technique. Confronting the experimental and numerical results, an unexpected behavior was observed in the local Nusselt number profile on the frontal disk. For lower Reynolds number and gap between disks, the results agreed very well, and a single peak on the local Nusselt was obtained in the diffuser entrance region for this case. As the Reynolds number and gap were increased, a secondary peak on the Nusseld profile was observed in experimental results, which were not identified by the numerical solution.

The results obtained by Pilichi (1990) were subject of further investigations by Peters (1994), who have studied self-induced bifurcations and oscillations in the same geometry. In his work, radial diffusers with radial and axial feeding were simulated numerically with several numerical models, and bifurcation points were obtained. Local Nusselt number profiles on the frontal disk were obtained and confronted better with the Pilichi's experimental results. However, there were still some differences between the results.

In the present work, an analysis of the non-isothermal flow through the radial diffuser is performed numerically using the Finite Volume Methodology. The main goal of the work is to analyze the heat transfer influence on pressure field on the frontal disk (valve reed).

2. METHODOLOGY

The governing equations of the problem for incompressible flow (the compressibility effects are lower than 0.2%) of a Newtonian fluid (mass conservation, momentum and energy equations) are:

$$\vec{\nabla} \cdot \vec{V} = 0 \quad (1)$$

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

$$\rho C_p \left[\frac{\partial T}{\partial t} + \vec{V} \cdot (\vec{\nabla} T) \right] = \vec{\nabla} \cdot (k \vec{\nabla} T) \quad (3)$$

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, \vec{V} is the velocity vector, and T is the temperature. The necessary boundary conditions to solve the problem are indicated in Figure 2, which also shows the computational domain.

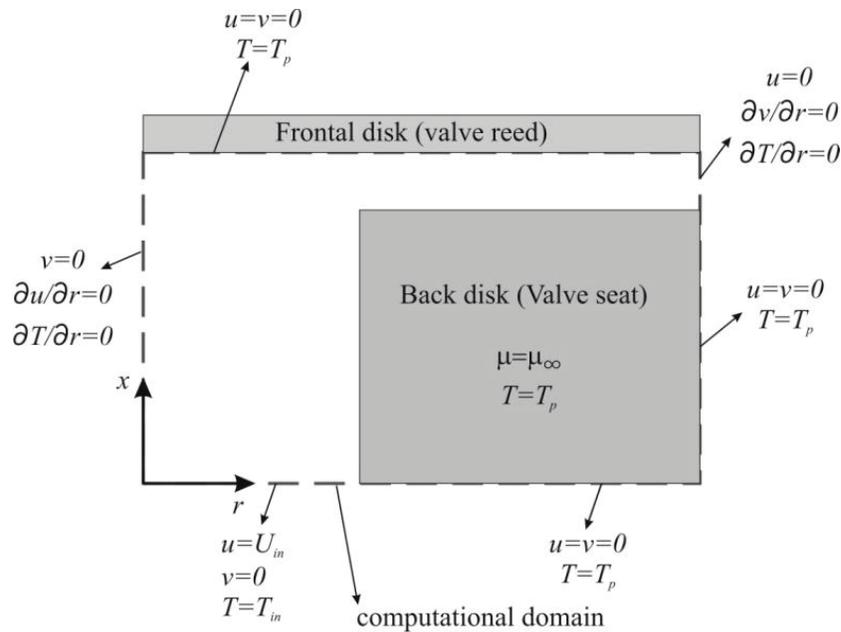


Figure 2. Computational domain and boundary conditions.

The influence of the heat transfer process in the fluid flow is taken into account in the momentum equation, Equation (2), through the variation of the dynamic viscosity with temperature, which is given by Equation (4) (Fox et al., 2004).

$$\mu = \frac{bT^{1/2}}{1+S/T} \quad (4)$$

where $b=1.458 \cdot 10^{-6} \text{ kg}/(\text{m s K}^{1/2})$, $S=110.4 \text{ K}$, and T is the temperature in Kelvin.

The governing equations and the boundary conditions are discretized using the Finite Volume method. 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. The power-law scheme is employed as interpolation function for the convective-diffusive terms, and the algebraic equations systems are solved by the TDMA-Three Diagonal Matrix Algorithm.

A non-uniform mesh refined in the highest gradient regions and near solid-fluid interfaces was used to generate the results. A number of 12,600 volumes in the diffuser region and a total of 47,600 volumes in the whole domain were used. A mass conservation residue of the order of 10^{-9} was used as convergence criterion. A more refined mesh with 190,400 volumes was tested without modifying significantly the results.

The dimensionless parameters used to present the results are the dimensionless pressure on the disk surface, P_{adm} , the Reynolds number at the entrance of the feeding orifice, and the local Nusselt number, defined by the following equations:

$$P_{adm} = \frac{P}{\frac{1}{2}\rho U_{in}^2} \quad (5)$$

$$Re = \frac{\rho U_{in} d}{\mu} \quad (6)$$

$$Nu = \frac{q''s}{(T_p - T_{in})k} \quad (7)$$

where k is the thermal conductivity of the fluid, q'' is the heat flux on the frontal disk surface, T_{in} and U_{in} are the inlet temperature and velocity of the fluid at the feeding orifice, respectively.

3. RESULTS

Dimensionless pressure profiles on the frontal disk surface were obtained for $D/d=4.15$, and for dimensionless gaps between the frontal disk and the back disk, s/d , equal to 0.04, 0.07, and 0.1. For each case four Reynolds numbers were investigated, $Re=500, 1000, 1500,$ and 2000 . Three surface temperatures were prescribed for analyzing the influence of the heat transfer on the pressure profile: $70, 100$ and 130°C . All these results were compared to the isothermal case. The results are depicted in Figures 3 to 14.

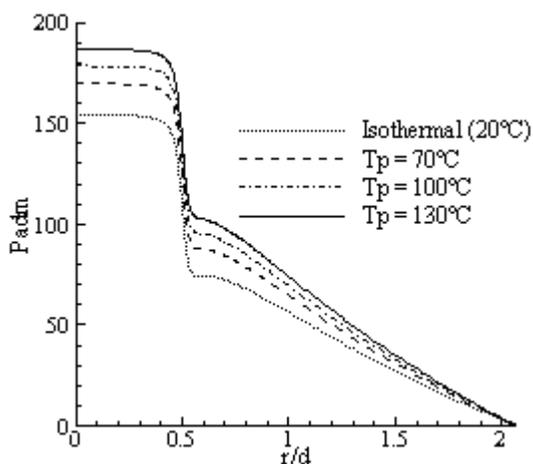


Figure 3. Pressure profile for $Re=500$ and $s/d=0.04$

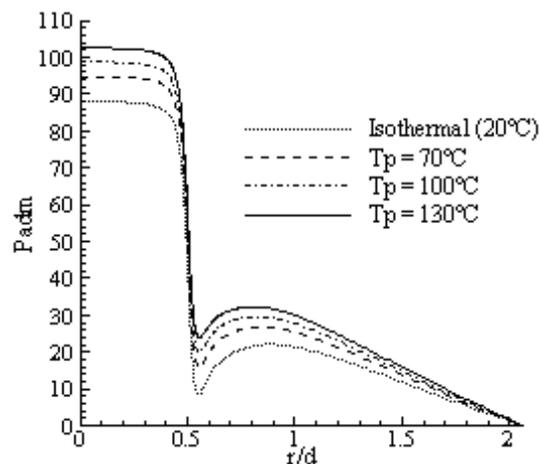


Figure 4. Pressure profile for $Re=1000$ and $s/d=0.04$

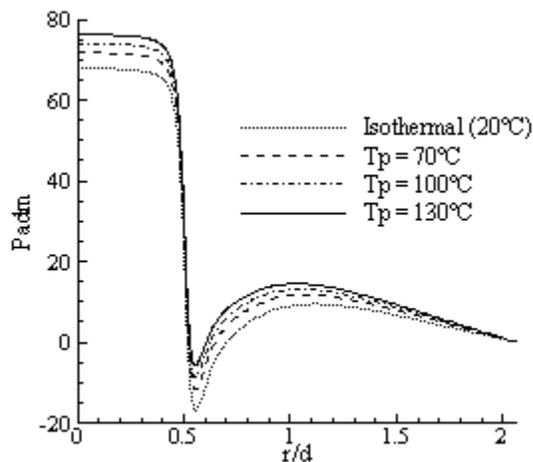


Figure 5. Pressure profile for $Re=1500$ and $s/d=0.04$

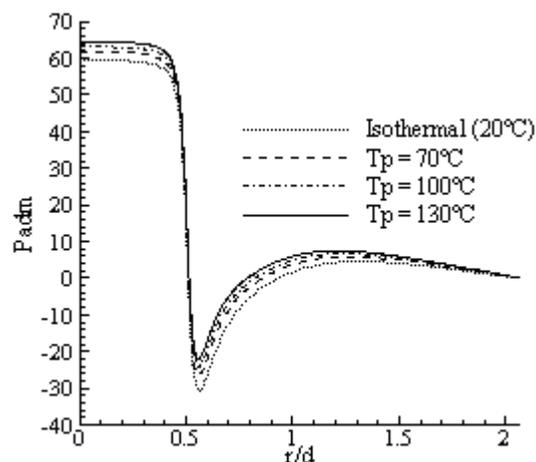


Figure 6. Pressure profile for $Re=2000$ and $s/d=0.04$

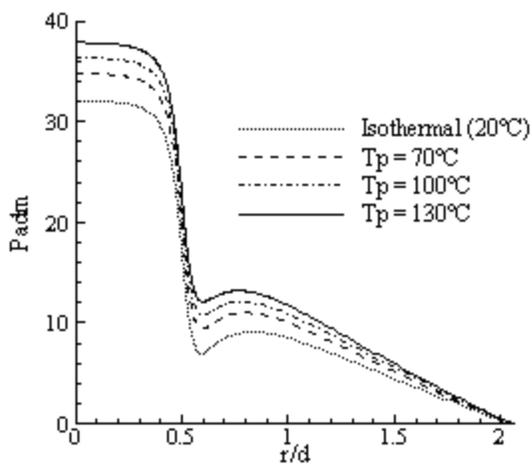


Figure 7. Pressure profile for Re=500 and s/d=0.07

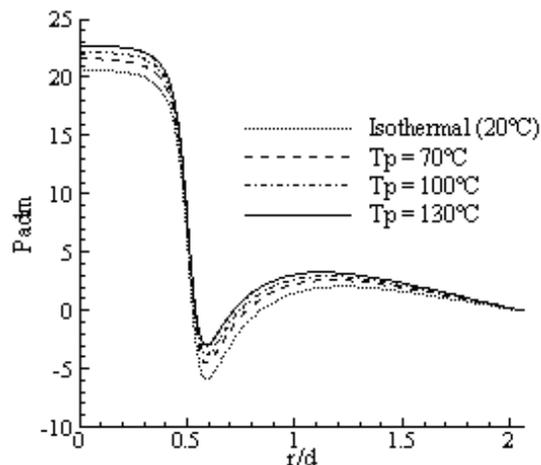


Figure 8. Pressure profile for Re=1000 and s/d=0.07

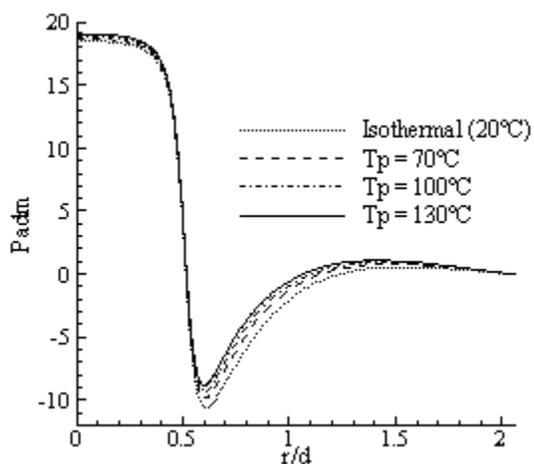


Figure 9. Pressure profile for Re=1500 and s/d=0.07

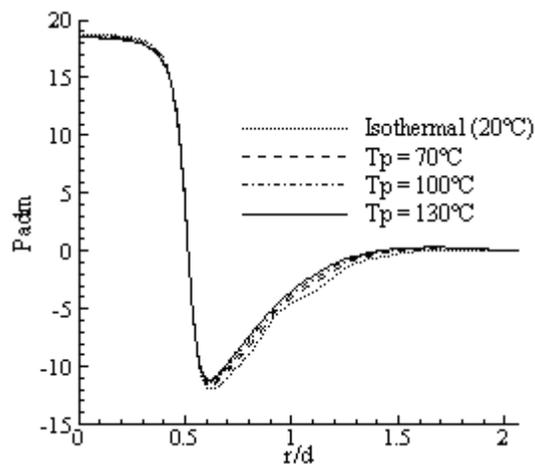


Figure 10. Pressure profile for Re=2000 and s/d=0.07

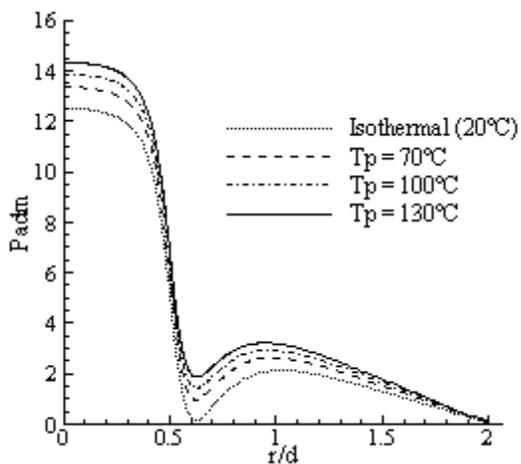


Figure 11. Pressure profile for Re=500 and s/d=0.1

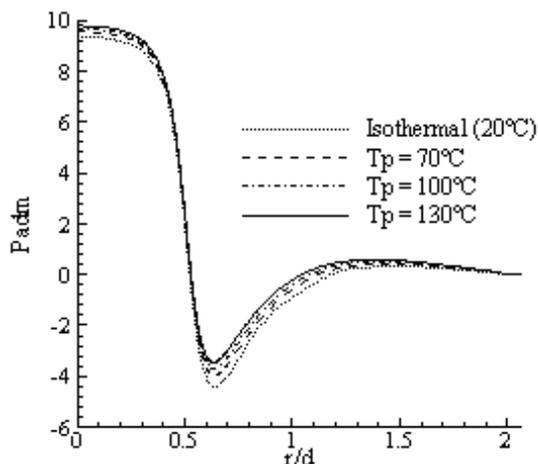
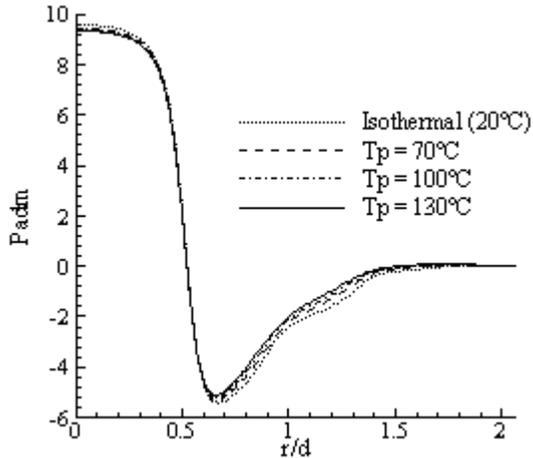
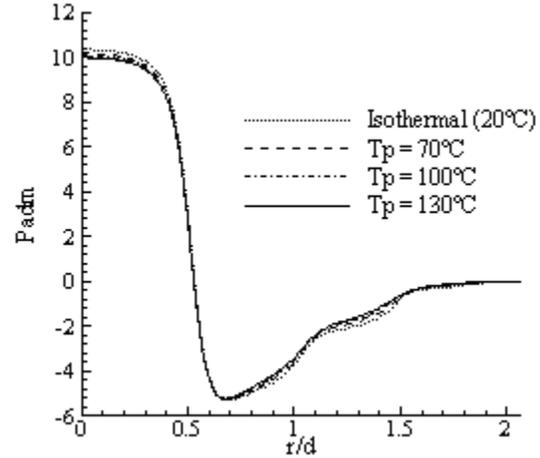


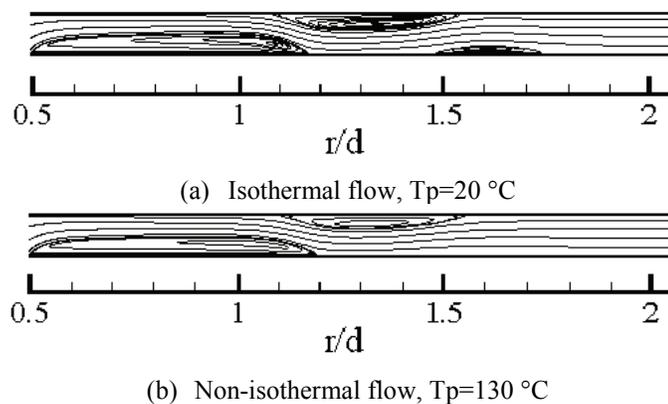
Figure 12. Pressure profile for Re=1000 and s/d=0.1

Figure 13. Pressure profile for $Re=1500$ and $s/d=0.1$ Figure 14. Pressure profile for $Re=2000$ and $s/d=0.1$

In general, the pressure profiles show a flat pressure distribution in the region of the orifice radius ($0 < r/d < 0.5$), where the fluid practically stagnates before entering the diffuser. Due to the acceleration of the fluid when it enters the diffuser region, there is a large pressure reduction. In some cases, mainly for low Reynolds, the pressure continues to decrease due to the predominance of the friction forces. As the Reynolds number increases, the flow acceleration can be so large that it produces negative pressures, followed by pressure recovery due to the increase of the cross section area with the radius. In addition, for a constant Reynolds number, the pressure level decreases for increasing gaps between disks due to the reduction of the friction force in the diffuser region.

An important parameter for valve designers is the total pressure drop through the valve. Figures 3 to 14 also can be used to analyze the influence of the prescribed wall temperature on the total pressure drop. In Figures 3 to 9 and Figures 11 and 12 it can be noted that the total pressure drop increases for increasing wall temperatures. In these cases, the heat transferred from the wall to the fluid increases the fluid viscosity, which produces higher friction forces in the diffuser region.

For Figure 10 ($s/d=0.07$ and $Re=2000$) there is practically no difference among the results. For the other cases, Figures 13 and 14, the total pressure drop decreases for increasing wall temperatures. This behavior can be explained by the modification of the flow pattern as the wall temperature increases. As can be seen in Figure 15 for $Re=2000$ and $s/d=0.1$, the size of the recirculation zones slightly decreases for increasing wall temperatures owing to the augmentation of the fluid viscosity, and this increases the effective section area of the flow in the diffuser region, which causes the reduction of the friction forces and, therefore, the reduction of the pressure drop. For $Re=500$ and $s/d=0.07$ this change of the flow pattern does not occur as shown in Figure 16.

Figure 15. Streamlines at the diffuser region for $Re=2000$ and $s/d=0.1$

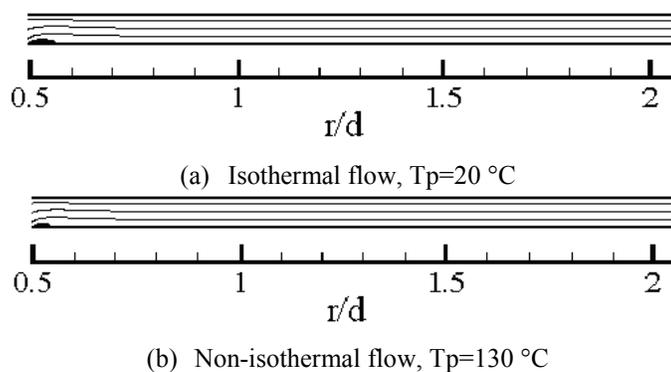


Figure 16. Streamlines at the diffuser region for $Re=500$ and $s/d=0.07$

Table (1) shows the total pressure drop difference between the isothermal and the non-isothermal flow for all analyzed wall temperatures. In this table, the influence of the heat transfer on the total pressure drop can be quantified. It is clear that this influence is higher for both lower Reynolds numbers and gaps between disks. The above discussion is clearly pointed out in the table. The major difference between the flow models achieves 21% for the lowest gap and Reynolds number.

Table 1. Total pressure drop difference between the isothermal and non-isothermal flow

	$Re=500$	$Re=1000$	$Re=1500$	$Re=2000$
$s/d=0.04$	21%	16%	12%	8%
$s/d=0.07$	18%	10%	3%	-1%
$s/d=0.1$	14%	5%	-3%	-4%

Figure 17 ($Re=500$ and $s/d=0.07$) and Figure 18 ($Re=2000$ and $s/d=0.1$) depict the distribution of the local Nusselt number on the frontal disk for the same cases whose streamlines were presented in Figs. 15 and 16. As can be noticed, for lower Reynolds number and gap, there is only one Nusselt peak at the inlet of the diffuser owing to the deflection of the flow coming from the orifice region. Instead, for higher Reynolds number and gap, a second Nusselt peak appears because of the formation of a second recirculation region (see Figure 15). The same type of behavior has been obtained by Peters (1994).

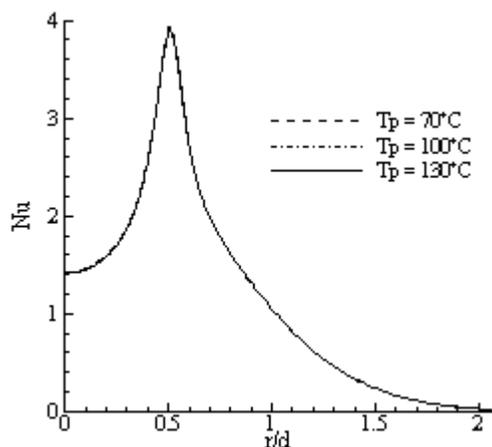


Figure 17. Local Nusselt number distribution for $Re=500$ and $s/d=0.07$

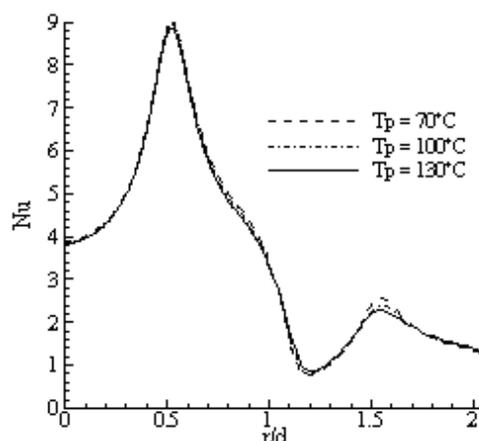


Figure 18. Local Nusselt number distribution for $Re=2000$ and $s/d=0.1$

4. CONCLUSIONS

In this work, an analysis of the heat transfer influence on the pressure distribution on the frontal disk of a radial diffuser representing the valve system of refrigeration compressors is performed numerically using the Finite Volume Methodology. The problem is analyzed for Reynolds number varying from 500 to 2000, three dimensionless gaps between the frontal (valve reed) and back (valve seat) disks: 0.04, 0.07, and 0.1, and for the following heat transfer configurations: isothermal flow, and three non-isothermal flows with wall temperatures equal to $T_p=70, 100, \text{ and } 130$ °C. In general, the numerical results have shown that the influence of the heat transfer diminishes for increasing Reynolds and gaps between disks. For low Reynolds numbers and low gaps between disks, the total pressure drop through the flow increases for increasing wall temperatures. As the Reynolds number and the gap increase, the influence of the wall temperature diminishes, and for the highest Reynolds numbers and gaps there is an inversion of behavior, that is, the total pressure drop decreases for increasing wall temperatures due to the modification of the flow pattern. The major augmentation of the total pressure drop for increasing wall temperatures achieves 21%.

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