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Heat Transfer and Fluid Flow Considerations in Automatic Valves of Reciprocating Compressors

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HEAT TRANSFER AND FLUID FLOW CONSIDERATIONS IN AUTOMATIC VALVES OF RECIPROCATING COMPRESSORS

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ABSTRACT

Several aspects related to fluid flow and heat transfer in reed type valves of reciprocating compressors are explored in the present work. Both numerical and experimental results for pressure distributions along the valve in laminar and turbulent flows and results for heat transfer from the valve walls are presented and discussed. The presence of hydrodynamic instabilities related to self-sustained flow oscillations is confirmed experimentally, and a criterion is established to predict those instabilities.

INTRODUCTION

The flow of refrigerant through automatic valves in reciprocating compressors has been of great interest to those who are involved in compressor design or in improving compressor efficiency. During the last two decades, there has been a rapid development of mathematical models, for use with a computer, to simulate compressors and so provide an aid to the design of the valve system. In general, the analysis yields two non-linear differential equations which relates the many parameters involved, as pointed out by MacLaren (1972):

- a "flow" equation which relates the pressure difference across the valve to mass flow rate and valve opening and

- a "dynamic" equation which describes the valve movement.

The interaction of the flow and the valve system has been translated to the mathematical models through the effective flow area, the effective force area, the valve natural frequencies and its normal modes, as emphasized by Soedel (1984). This simple approach is generally adequate when the main purpose of the simulation model is to predict global compressor performance. However, this approach supplies only rudimentary information of use in the detailed design of a valve as stressed by Papastergiou et al. (1980).

As mentioned by Maclaren and Kerr (1970), manufacturers and users of compressors are deeply concerned in the event of valve failure, short valve life or poor compressor performance due to valve malfunction. Hence any study which may lead to improvement of this vital component is worthwhile. In order to determine the most important parameters for the performance analysis of valve systems, it is necessary to fully understand the gas flow through the valves. The analysis of the flow field throughout automatic valves is very complex and laborious, especially if the system geometry is fully taken into account. Therefore, simple geometries, like radial diffusers, are generally used in the laboratory and for numerical calculations, once the main features of the flow are also present there as mentioned by Deschamps et al. (1988) and Ferreira et al. (1989). In the radial diffuser flow, including the feeding orifice, one faces usually a turbulent flow in presence of stagnation, recirculation, acceleration and variable adverse pressure gradients. The presence of these effects in the flow makes it very difficult to be analyzed by using the available numerical models. Therefore several modifications have to be implemented in a standard κ-ε model in order to numerically calculate this "simple" laboratory flow.

Even for laminar, incompressible isothermal flow through the same geometry, several difficulties arise for the flow understanding due to the presence of self-sustained radial oscillations which are generated by the adverse pressure gradients in the radial diffuser and also by shear instabilities in the corner of the feeding orifice. These instabilities show up in the flow for certain combinations of the gap between discs and the flow Reynolds number, as mentioned by Prata et al. (1989). It has been detected the enhancement of the convective heat transfer from the walls due to the existence of those oscillations. From the suction valve standpoint this phenomenon is undesirable as it represents a loss in the volumetric efficiency of the compressor.

Regarding heat transfer aspects in compressor valves very few works are encountered in the literature. Hughes (1972), presented average Nusselt numbers correlated to Reynolds and Prandtl numbers for ring-type valves. Hurjun and Yezheng (1988) performed mass transfer experiments employing the naphtalene sublimation technique to obtain Nusselt number correlations for different valve geometries. The situation investigated by Hurjun and Yezheng (1988) resembles that studied here; however, several imperfections detected in their work preclude the use of the information presented therein.

Two works dealing with heat transfer in radial diffusers have been found in the literature. Kreith (1966), investigated the heat transfer from the diffuser walls to the fluid assuming known velocities profiles. Mochizuki and Yao (1983) presented local Nusselt numbers results for situation in which the axial gap between the disks is very small compared to the disks diameter. Their experiments indicated laminar flow separations occuring periodically, and alternating from both disks walls at the same radial location. The measurements of Mochizuki and Yao (1983) showed a substantial increase in the heat transfer due to separation. None of the previously cited works deals with the heat transfer in radial diffuser including the flow entrance in the feeding orifice as is the case of the present work.

In this paper, several aspects related to fluid flow and heat transfer in reed type valves of reciprocating compressors are discussed. The results for pressure distributions and the corresponding resultant force for both laminar and turbulent flow are presented for different mass flow rates and different gaps between the reed and the seat. Also, contour plots for turbulent viscosity are included in order to indicate the regions of higher turbulence. A whole section is devoted to hydrodynamic instabilities related to self-sustained flow oscillations. Experimental results confirm the existence of flow instabilities and their relation to the heat transfer from the valve walls.

PROBLEM GEOMETRY AND FORMULATION

The investigation to be described in the following sections is based on a simplified geometry of the actual valve geometry. That is a radial diffuser with axial feeding as depicted in Fig. 1.

For the radial diffuser configuration the flow is axially symmetric. As shown in Fig. 1 the fluid is fed axially through an orifice of diameter d, length e and hits a frontal disc with diameter D. Due to the presence of the valve reed the flow is deflected and becomes radial. The disk with the feeding orifice is analogous to the valve seat, and the other disk corresponds to the valve reed. In what follows, the refrigerant gas flow through the valve system is characterized by the Reynolds number, Re, based on the orifice port diameter.

The distance between the valve seat and the valve reed has a strong effect on the flow field. This is equally valid for both laminar and turbulent regimes. An important aspect to be stressed is the flow separation at the entrance of the diffuser section for certain combinations of Reynolds number and distance between the diffuser walls. After separation, the flow reattaches downstream, forming an annular separating bubble. For increasing values of the gap between the diffuser walls, the separation bubble grows to the end of the radial diffuser, forming a radial jet over the valve reed as explored by Moller (1963).

Attention next will be focused on the differential equations that describe the problem. To this extent, consideration will be given separately to the laminar and turbulent regimes as well as to the energy equation.

For the investigation of the laminar pressure distribution along the valve reed the flow is taken to be incompressible and newtonian, and is fully described by continuity equation, and the axial and radial momentum equations. In dimensionless form the Reynolds number, and the gap between the disks are the important parameters that govern the flow. In addition to the velocity boundary conditions at the symmetry axis and at the solid walls, at the diffuser exit the flow is taken to be locally parabolic.

For the turbulent regime, use is made of the time-average forms of continuity and momentum equations. The turbulent viscosity was determined from the $\kappa-\varepsilon$ turbulent model, which included flow-relaminarization functions according to the model proposed by Nagano and Hishida (1987). Turbulence anisotropy and compressiblity effects were also incorporated into the turbulent model. In addition to the flow complexity associated to the geometry of the radial diffuser, the occurrence of strong accelerations, separation, and adverse pressure gradients make this flow very difficult to simulate.

Heat transfer computations were performed assuming the properties to be temperature independent. Therefore, the thermal problem was uncoupled from the hydrodynamic problem. Once the velocity field had been determined, the temperature could be directly obtained from the energy equation. The only non-geometrical parameter governing the thermal problem is the Prandtl number. Two boundary conditions were explored, prescribed temperature at both valve seat and valve reed, and prescribed temperature at the valve seat and zero heat flux at the valve reed. It is expected that those situations represent limiting cases for actual boundary conditions at the valve reed. For valves under operation, heat is transferred from (and to) the refrigerant inside the cylinder to (and from) the reed. This heat may then be conducted through the reed to the cylinder walls or may be convected to (or from) the refrigerant passing the valve. In order to precisely determine the actual boundary conditions, the computation domain would have to include the valve seat as well as the compression chamber. Had this computation been tryed, cost and computer memory would have made the solution of the governing equations prohibitive. For the heat transfer problem, the refrigerant inlet temperature was prescribed, and at the diffuser exit convection is dominant over conduction, and, therefore, boundary condition for locally parabolic flow was adopted.

EXPERIMENTAL SETUP AND PROCEDURES

The experimental setup for both pressure and heat transfer measurements operated in open circuitry. Air was taken to the test section from a constant temperature quiescent ambient. For flow regulations, use was made of a flowrate control valve and a calibrated orifice flow meter.

Pressure measurements were performed using a frontal disk especially designed and constructed for the present investigation. Along the disk diameter there is a sliding bar provided with a small tap hole and an internal connecting perforation up to one of the ends of the bar. Through this perforation a differential pressure inductive transducer detected the pressure profile along the frontal disk. At the other end of the sliding bar, an inductive displacement transducer indicated the tap hole location. A positioning system was used to correctly establish the gap between the valve reed and the valve seat. This gap had to be precisely determined in order to minimize experimental errors. Sensivity tests indicated that the pressure distribution was highly dependent upon the gap between reed and seat. For example, at Reynolds number of 1450, and gap to feeding orifice diameter ratio of 0.009, a gap variation of 15 μm would cause the stagnation pressure to vary by 16%. In this regard, even the disk gap variations due to forces produced by the flow induced pressure distribution had to be corrected during the data reduction. Further information related to the pressure measurement system are provided in Ferreira et al. (1989).

The heat transfer measurements were made indirectly via de existing analogy between heat and mass transfer. Particularly, the naphtalene sublimation technique (see for instance Souza Mendes, 1988) was employed. In this technique the heated surface is replaced by a naphtalene-coated surface. During a data run, the coated surface is exposed to an airflow causing naphtalene sublimation from the surface. From the measurement of the surface recess depth and the time elapsed during the data run, local Sherwood numbers could be determined. Employing the heat and mass transfer analogy the Sherwood number from the experiments can be easily converted to Nusselt number corresponding to Prandtl number equal to 2.5, which is the Schmidt number of naphtalene vapor subliming into air. Naphtalene sublimation depths were on the order of 40 μm and were measured using a computerized Carl Zeiss micrometric table having resolution of ±1μm. Average Sherwood number could be obtained by direct integration of local Sherwood numbers or by weighing the naphthalene-coated piece before and after it had been exposed to the airflow. From the mass and the time elapsed during the data run the average Sherwood number could be determined.

NUMERICAL METHODOLOGY

The equations describing both the hydrodynamic and the thermal problems were solved using the finite volume method as described by Patankar (1980). In this method the solution domain is first divided in small non-overlapping control volumes, and next the differential equations are integrated over each one of the control volumes. From those integrations resulted a set o algebraic equations, which were solved iteratively via the line-by-line method (see Patankar 1980). For the solution of the velocity field, the continuity equation was used to obtain an equation for pressure according to the SIMPLER algorithm (Patankar, 1981). With this procedure a strong coupling exist between pressure and velocity which facilitates convergence. A crucial feature of a numerical solution of problems like the ones investigated here is the selection and deployment of the grid population. The reason is the existence of high gradients of the unknown variables caused by both the strong variations of the velocity field along the flow passage, and the complexity of the solution domain. For each situation to be investigated a particular grid configuration was selected. In general, the mesh chosen represented a compromise between accuracy and computer time required to achieve convergence. It is expected the accuracy of the numerical solution to be around 5%. Acceleration of the iterative scheme was achieved with use of the block correction algorithm of Settari and Aziz (1973).

LAMINAR FLOW REGIME

When the flow Reynolds number is small ($Re < 100$) and the reed is very close to the seat ($h / d = 0.01$), viscous effects predominate over inertial effects. The solution for this type of flow, when inertial effects are small, has been obtained analitically by Livesey (1960). As the flow Reynolds number increases, the inertial effects become important and can no longer be neglected. Therefore the solution can only be obtained numerically based on the differential equations governing the flow field and the associated boundary conditions. The validation of the solutions has been extensively performed as reported by Deschamps (1987).

Numerical results of pressure distribution along the diffuser for two distances between discs, $h/d = 0.01$ and 0.04, and the flow Reynolds number, Re, are presented in Figs. 2 and 3, respectively. For each value of h/d, three different values of Re have been analyzed : Re = 500,900 and 1800.

The main characteristic of the pressure distributions observed in Figs. 2 and 3 is the pressure plateau for $r/d < 0.5$. This plateau corresponds to the area in front of the feeding orifice in which the flow is altering its direction, that is, changing from axial to radial. For $r / d = 0$. the pressure on the valve plate reaches its maximum value, which is the stagnation pressure. Another characteristic of the pressure distribution is the abrupt decay at the entrance to the radial diffuser ($r/ d = 0.5$). When both the Reynolds number and the disk gap are small, the pressure drops first then decreases monotonically to the atmospheric pressure. At higher Reynolds numbers and larger values of h/d, the gauge pressure drops so sharply at the beginning of the diffuser region that it reaches negative values. These negative values are due to the flow separation close to the feeding orifice.

Integration of the pressure distribution along the valve disk gives the axial resultant force acting on the diffuser. Both numerical and experimental forces are presented in Fig. 4 where a good agreement can be observed.

Any geometrical modification performed on the valve disk, on the feeding orifice or on the valve seat, changes substantially the pressure distribution along the valve reed and consequently the resultant axial force. The numerical solution can be safely used to get the desired flow information.

TURBULENT FLOW REGIME

As the flow Reynolds number increases substantially, the viscous effects are completely overcome by the inertial effects and then the flow becomes turbulent. The only possible solution for the flow is based in turbulent models requiring a very fine grid mesh in the regions of high velocity gradients and a very slow convergence process. Any attempt to reduce the relaxation coefficients and speed up the solution makes the process to diverge. As this turbulent flow occurs in the presence of stagnation, recirculating and accelerating regions and also under variable adverse pressure gradients, standard turbulence models do not furnish accurate results for the pressure profile along the valve disk. Several modifications have to be implemented in the model in order to overcome these difficulties, as reported by Deschamps et al. (1988).

Figs. 5 and 6 show the comparison between numerical and experimental results for the pressure profile along the valve disk for h/d = 0.05 and Re = 13 325 and 23 275, respectively.

As it can be observed, the pressure distribution on the valve disk for turbulent flow shows the same aspect as for laminar flow although the dimensionless stagnation pressure plateau does not present such high levels but the negative peak is deeper.

Figs. 7 and 8 present the contour plots for the ratio of the turbulent and the fluid absolute viscosity along the feeding orifice and the radial diffuser, respectively. These Figures indicate high turbulent intensity in the radial diffuser, and very small values along the feeding orifice. Although the comparison between numerical and experimental results for the pressure distribution along the valve disk can be considered reasonably good, other turbulence models have to be used in order to capture the deeper negative pressure in the radial diffuser.

HEAT TRANSFER

Heat transfer from the suction valve walls to the refrigerant is undesirable as it represents losses in the compressor volumetric efficiency. In this section results for the refrigerant bulk temperature and local and average Nusselt numbers will be explored. From the designer point of view those results are most important as they yield insight on how the fluid is heated as it is admitted into the compressor cylinder. In what follows, attention will be focused on the laminar flow regime.

Fig. 9 presents the dimensionless bulk temperature at the end of the feeding orifice (x=e according to Fig. 1) as a function of h/d having the Reynolds number as a curve parameter. The dimensionless temperature θ is given by (T-Te)/(Tp-Te), where Te is the temperature at the orifice entrance and Tp is the wall temperature. The solid lines are for adiabatic valve reed whereas the dashed lines are for isothermal valve reed. Both orifice and valve seat walls are heated at constant temperature, equal to the valve reed temperature for the isothermal case.

It is seen that as Re increases the bulk temperature at the orifice exit decreases. Larger values of Re result in larger heat transfer coefficients; however, for larger values of Re the fluid residence time in the orifice is smaller resulting in less heating. It is also observed in Fig. 9 that the bulk temperature at the orifice exit decreases as h/d is increased. This is due to the fact that for larger gaps between seat and reed, lower temperature and velocity gradients are induced at the orifice walls resulting in smaller heat transfer coefficients. Additionally, the bulk temperature at the orifice exit for the isothermal reed is always higher than that for the adiabatic reed, as expected.

Fig. 10 shows the influence of the Reynolds number on the dimensionless bulk temperature at the diffuser exit plotted against h/d. For low Reynolds numbers, the refrigerant exits at a bulk temperature very close to the wall temperature, that is $\theta_d = 1$.

Attention is now turned to overall Nusselt numbers, Nu_T. To this extent Fig. 11 was prepared. Again, the dashed lines are for isothermal reed and the solid lines are for adiabatic reed. The Nusselt number is defined using the temperature difference (Tp - Te). In this regard, Nu_T is in fact the dimensionless heat transfer from the walls to the refrigerant.

At low Reynolds numbers, the fluid reaches the wall temperature prior it leaves the diffuser. Therefore, it gets saturated and stops receiving heat. This can be seen from Fig. 11 in which at Re = 150, both the isothermal and the adiabatic cases yield the same value of Nu_T. As Re increases, Nu_T for the isothermal reed overcomes more and more Nu_T for the adiabatic reed. At a given Reynolds number, Nu_T decreases as h/d increases. This behaviour is due to the existence of higher temperature and velocity gradients for lower values of h/d. The results presented in Figs. 9 and 10 were obtained by Puff (1988) and Todescat (1988).

Local Nusselt numbers will now be explored. Attention will be given to a situation in which the walls of the orifice and the valve seat are maintained adiabatic, while the valve reed is kept isothermal. To this extent Fig. 12 was prepared. Fig. 12a is for Re = 1542, and Fig. 12b is for Re = 2477; for both figures $h/d = 0.07$. These Figures present experimental and numerical results, and except near the reed exit a very good agreement prevailed between both. As the flow exits from the feeding orifice it impinges on the valve reed yielding the heat transfer peaks observed in Figs. 12a and 12b. The stagnant region comprised between $(-d/2 < r < d/2)$ is represented by the valley on the local Nusselt number profiles. As the flow progresses through the diffuser, the Nusselt number decreases asymptotically, as observed in the figures. An interesting aspect observed from the figures is the existence of secondary peaks on the local Nusselt number near the reed exit. Those peaks were not captured in the numerical solution, even when the grid population was substantially increased in that region. The secondary peaks increase with increasing values of Re. The existence of those peaks is believed to be related to flow instabilities as will be discussed in the next section.

HYDRODYNAMIC INSTABILITIES

For radially outward flow between parallel disks, when inertia effects are large compared to viscous effects, the increase in the cross-sectional area in the flow direction cause the fluid to go from low to high pressure regions. This adverse pressure gradient can be so strong as to cause the fluid to separate in cells that change their position with time. Substantial increase in heat transfer is due to separation as observed by Mochizuki and Yao (1983).

A scale analysis of the terms of the energy equation indicates that the local Nusselt number is function of two parameters, StPe and Pe(h/d), where St is the Strouhal number (hf/v) and Pe is the Peclet number (vh/α). For stable regimes in which there is no flow oscillations, $St \rightarrow 0$, and the Nusselt number is function of Pe(h/d).

Usually in compressor valves $St \gg (h/d)$, and for unstable regimes the Nusselt number is function of Pe only. The previous relationships can be corroborated using the experimental results of Mochizuki and Yao (1983), as shown by Prata et al. (1990).

In compressor valves it is important to identify whether the flow regime is stable or unstable. At present, no definite conclusion has been given to explain the transition from one regime to the other when the orifice port is considered in fluid flow in radial difuser. However, as a first indication, use can be made of the work by Todescat (1988). There, numerical and experimental results showed good agreement for low values of the Reynolds number $(Re = 4\dot{m}/\pi\mu d)$. As the Reynolds number increases, there is a critical value of Re, denoted here by Re_c, beyond which the experimental results start departuring from the computations. Values of Re_c are presented in Fig. 13, and can be used as a first approximation to indicate the transition from the stable to the unstable regime.

For the unstable regime the Nusselt number seems to be indepedent of the gap between the diffuser walls as indicated by the solid line in Fig. 14.

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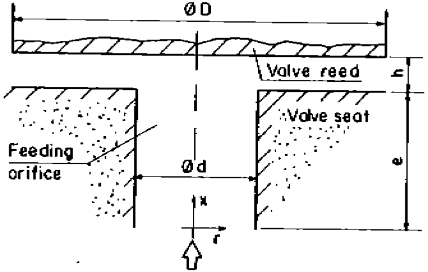


Fig. 1 Flow geometry

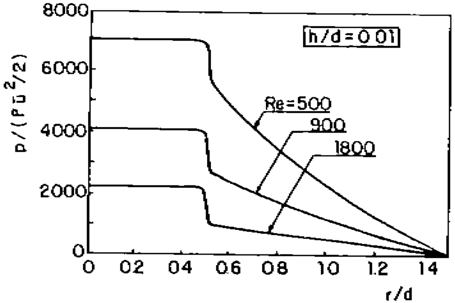


Fig. 2 Influence of Reynolds number on pressure distribution along valve reed for h/d=0.01

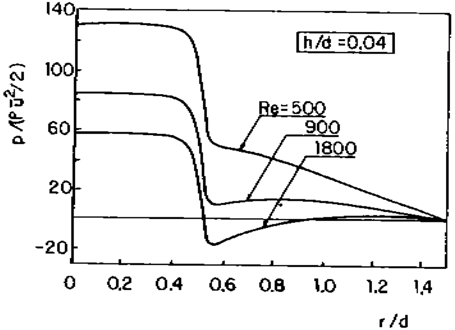


Fig. 3 Influence of Reynolds number on pressure distribution along valve reed for h/d=0.04

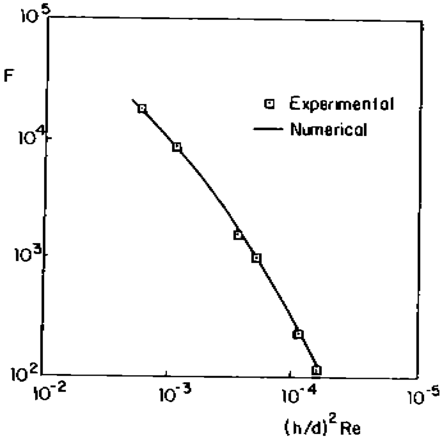


Fig. 4 Axial force on the valve reed

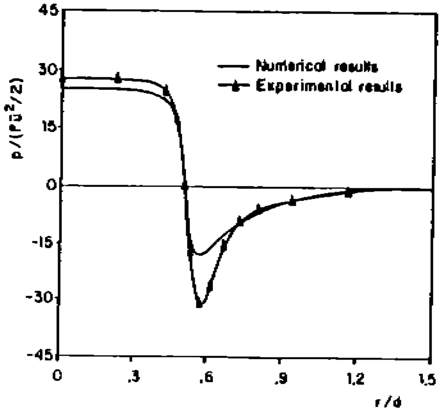


Fig. 5 Comparison between numerical and experimental results for h/d=0.05 and Re=13325

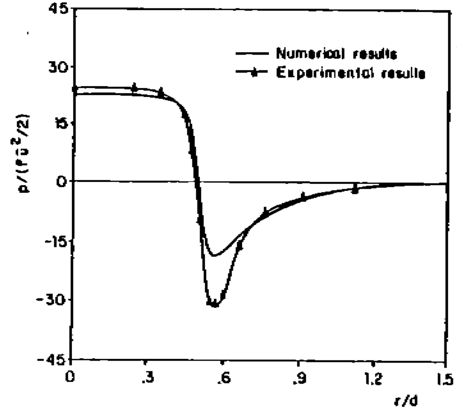


Fig. 6 Comparison between numerical and experimental results for h/d=0.05 and Re=23275

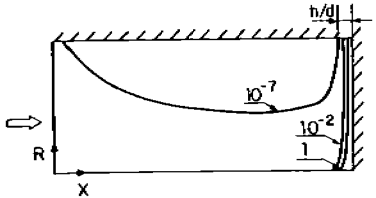


Fig. 7 Contour plots of constant μ_T/μ in the feeding orifice; h/d=0.05 and Re=23275

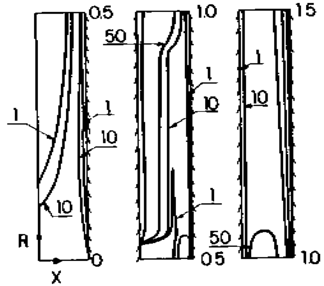


Fig. 8 Contour plots of constant μ_T/μ in the radial diffuser for h/d=0.05 and Re=23275

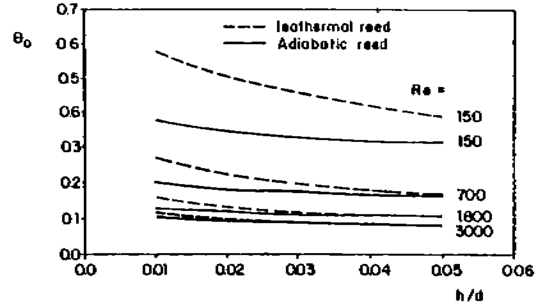


Fig. 9 Dimensionless bulk temperature at the orifice exit

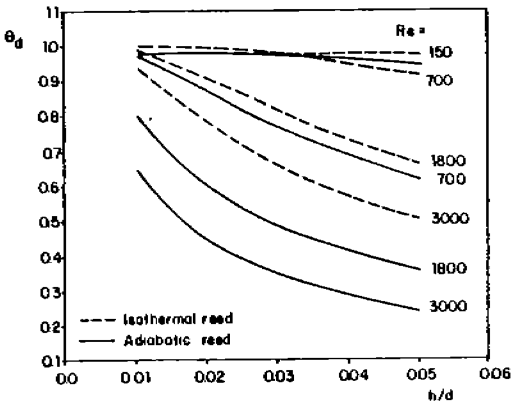


Fig. 10 Dimensionless bulk temperature at the diffuser exit

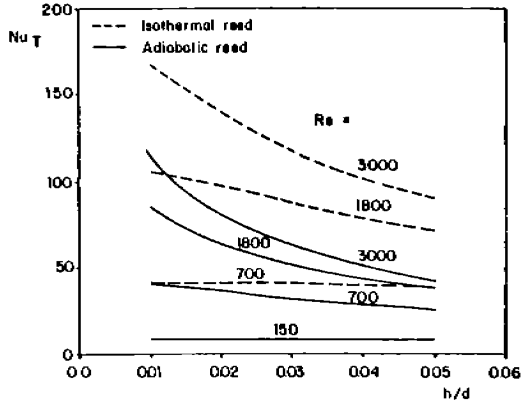
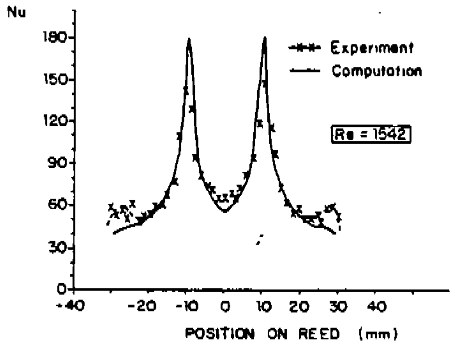
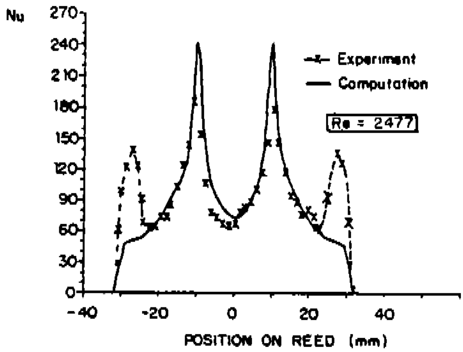


Fig. 11 Overall Nusselt number in the valve



a) Re = 1542



b) Re = 2477

Fig. 12 Local Nusselt number on the valve reed for adiabatic orifice and valve seat; h/d=0.07

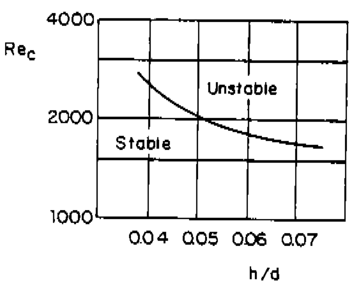


Fig. 13 Critical Reynolds number for the valve

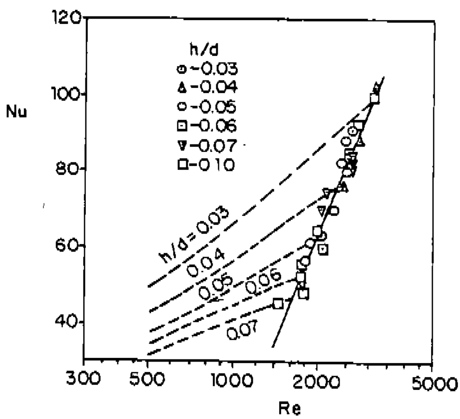


Fig. 14 Average Nusselt number for orifice and seat (adiabatic reed)