Analysis of Laminar Flow and Heat Transfer Between a Stationary and a Rotating Disk

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Abstract

LAMINAR flow and heat transfer between a stationary and a rotating disk has been analyzed numerically. The fluid enters the domain through a circular opening in the center of the stationary disk and flows radially outward. Unlike most of the earlier analyses, which were of the boundary-layer type, the present study deals with the solution of the complete Navier-Stokes equations and the prediction of the entire recirculating flow. Results show that, in the presence of rotation, the flow is indeed quite complex and is characterized by recirculating zones near the inlet and the exit.

Contents

Background

Radial outward flow between two coaxial disks occurs in a number of situations of engineering interest, including turbines, pumps, diffusers, rotating heat exchangers, disk brakes, thrust bearings, etc. A recent review article by Owen[1] provides a comprehensive literature survey. Although the problem under investigation has been studied before, all of the analyses are of the boundary-layer type. Such boundary-layer analyses cannot account for the recirculation zones near the inlet or the exit and thus one must actually solve the complete set of Navier-Stokes equations to analyze this complex recirculating flow. It is this problem that provided the motivation for the present study.

Problem Definition

The problem to be solved is sketched in Fig. 1. It concerns the prediction of flow and heat transfer between a pair of coaxial disks of diameter D_o. The spacing between the disks is represented by h. The upper disk is stationary and the lower disk rotates at an angular frequency Ω. Flow enters the domain axially through a circular opening of diameter D_i in the stationary disk. A parabolic velocity profile with an average velocity u_{in} and a swirl free flow is assumed at the inlet. The stationary disk is adiabatic, while the rotating disk is isothermal at a temperature T_w different from the uniform temperature T_i of the incoming fluid. Under certain conditions, there can be inflow at the exit plane at $r=D_o/2$. Such inflow is assumed to radially entrain swirl free fluid at a temperature T_i.

Solution Procedure

The flow is assumed to be laminar. The fluid is taken to be incompressible and all properties are regarded as constant. The governing equations are the Navier-Stokes equations representing the conservation of mass, momentum, and energy. These were solved numerically using the control volume-based finite difference method described by Patankar.[2] The SIMPLER algorithm was employed. A 42×22 (axial $\times$ radial) nonuniform grid was used, with the grid lines packed close to the walls and the inlet.

Results and Discussion

For the numerical solution to be credible, some comparison with experimental data is essential. In Fig. 2a, the variation of cross-section-averaged pressure with radial distance is compared with the experimental results of Coombs and Dowson.[3] These results, both computed and experimental, correspond to $h/D_i=0.0693$, $D_o/D_i=27$, $Re_t=71.8$, and $Re=28.89$. Here, the inlet Reynolds number Re is defined as $u_{in}D_i/\nu$ and the rotational Reynolds number Re_t as $\Omega D_i^2/\nu$, where ν is the kinematic viscosity. As can be seen, the agreement is quite satisfactory. In Fig. 2b, results for the average heat flux from the lower plate are presented. The experimental data of Krieth and Viviand[4] involves $D_i=1$ in., $D_o=8$ in., lower wall insulated up to a radius of 1.5 in., gap spacing varying 0.048-0.126 in., inlet volumetric flow rate varying 8 to 20×10^{-3} ft^3/s, Prandtl number of 2.4 (Schmidt number in their mass transfer experiments), and rotational rates of 200 and 300 rpm. Exactly what combination of inlet flow rate, gap spacing, and rotation rate was used to obtain different experimental points in Fig. 2b is unknown. Hence, for the computations, the following procedure was adopted. The rotational rate was kept fixed at 200 rpm. The gap spacing was first maintained as 0.048 in. (the lowest) and the flow rate was varied from 8 to 20×10^{-3} ft^3/s. This gave the lower solid line in Fig. 2b. Then, the gap spacing was kept fixed at 0.126 in. (the highest) and again the flow rate was varied from 8 to 20×10^{-3} ft^3/s. This yields the upper solid line in Fig. 2b. As can be seen from this figure, the computed results are in good agreement with experimental data—in fact, in better agreement than the approximate analysis of Ref. 4, since the latter does not account for the explicit dependence of the results on h/D_i, Re, and Re_t.

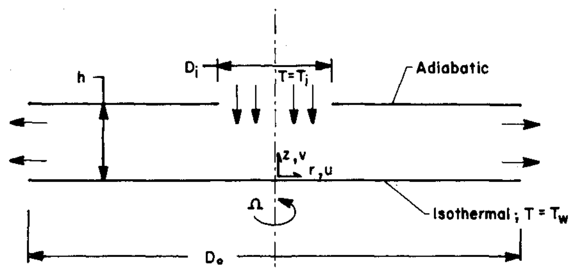

Fig. 1 Flow through a system of coaxial disks.

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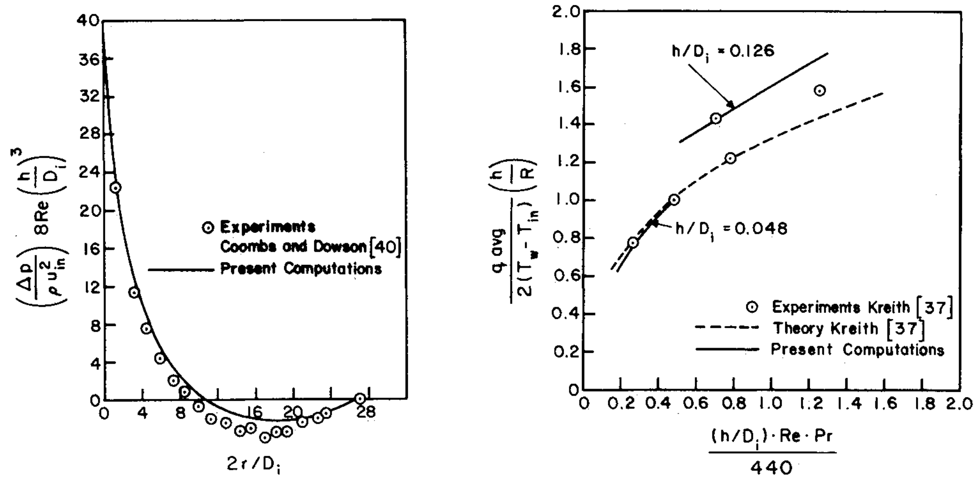

Fig. 2 Comparison with experimental data.

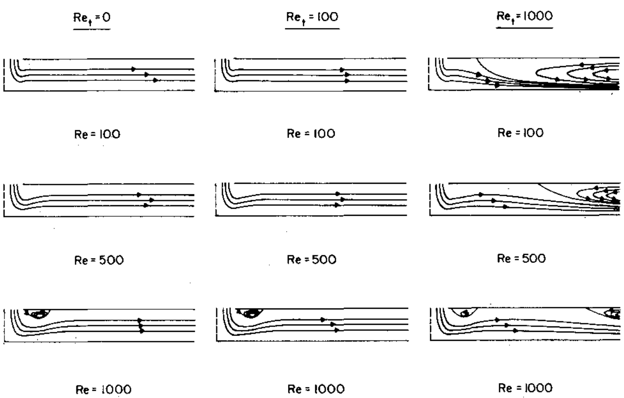

Fig. 3 Representative streamline plot; $h/D_i = 0.1$, $D_o/D_i = 10$.

With a favorable agreement between the computed and experimental results having been established, a parametric study was undertaken in which $Re = 100$, 500, 1000; $Re_t = 0$, 100, 1000; $h/D_i = 0.05, 0.1, 0.5$, and Prandtl number $= 0.7$. Detailed results regarding the radial pressure variation, torque on the disks, heat flux from the lower disk, etc., are presented in Ref. 5. Representative streamline plots describing the flowfield are shown in Fig. 3. For stationary disks ($Re_t = 0$) or for the low-rotational Reynolds number of $Re_t = 100$, there is no recirculation region for the inlet Reynolds number of 100 and 500. At the high inlet Reynolds number of 1000, a small recirculation bubble appears close to the inlet. The flowfield is more interesting for the highest rotational Reynolds number of 1000. At the inlet Reynolds number of 100, there is one recirculating zone close to the exit of the disks. The size of this eddy decreases as the inlet number increases to 500. This is due to the fact that the incoming fluid at higher velocity has a greater capability of surmounting the adverse pressure gradient. For the inlet Reynolds number of 1000, the outer eddy becomes very small and the eddy at the inlet appears, giving rise to two recirculating eddies. If the inlet Reynolds number were increased further, the inner eddy would grow in size and the outer eddy would disappear. An increase in h/D_i increases the size of both the eddies.

The complex flowfield is a consequence of the interaction between various hydrodynamic forces. The pressure gradient can be favorable or adverse, details of which, as already mentioned, are discussed at length in Ref. 5.

Conclusion

Results indicate that flow in a coxial disk system is quite complex and consists of recirculation zones near the inlet and the exit. This effects the heat transfer and other flow characteristics.

References

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