Heat and Mass Transfer in Rotating Machinery

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HEAT AND MASS TRANSFER IN ROTATING MACHINERY

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Preface

This volume contains papers from the XIV Symposium of the International Centre for Heat and Mass Transfer, held September 1982 in Dubrovnik, Yugoslavia.

The symposium was organized to focus attention on heat and mass transfer processes especially associated with rotating machinery components. The understanding of such processes plays an increasingly significant role in the continued development of many different types of rotating machinery.

The objective of the Dubrovnik meeting was to bring together researchers and practitioners in a forum for exchange of information both on research topics and on design problems and strategies. Contributed papers were organized into sessions on generic research areas and on specific types of machines. The same general format has been followed in arranging this volume, although in some cases papers span more than a single category and placement is therefore somewhat arbitrary.

The editors would like to acknowledge the contribution of the following organizing committee members and session chairmen: G. Bois, Ecole Centrale de Lyon, France: M. E. Elovic, General Electric Company, USA; B. Gal'Or, The Technion, Israel; M. Hirata, University of Tokyo, Japan; M. Majcen, University of Zagreb, Yugoslavia; P. J. Marto, U.S. Naval Postgraduate School, USA; R. E. Mayle, Rensselaer Polytechnic Institute, USA; W. D. Morris, University of Hull, UK; J. M. Owen, University of Sussex, UK; D. B. Spalding, Imperial College of Science and Technology, UK; S. L. K. Wittig, University of Karlsruhe, FRG; C. H. Wu, Engineering Thermophysics Research Institute, People's Republic of China.

Darryl E. Metzger
Chairman, Symposium Committee

Naim H. Afgan
Scientific Secretary, ICHMT

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GAS TURBINES

STEAM TURBINES

ROTATING HEAT PIPES AND THERMOSYPHONS

ROTATING TUBES AND CHANNELS

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ABSTRACT

The objective of this paper is to give a general review of secondary flows and enhanced heat transfer in rotating pipes and ducts. The secondary flows are caused by body forces such as Coriolis force, that are caused by density variation in a centrifugal field or the resultant of that by density in a centrifugal field and centrifugal force due to curvature of a duct. Heat transfer in rotating ducts is enhanced by the secondary flow and its performance varies with the shape of the duct cross-section and the intensity and orientation of the body force.

Based on these understandings, theoretical and experimental works on featuring secondary flows and heat transfer performances in rotating and revolving pipes and ducts are summarized. As theoretical ones, the two analytical methods and numerical works are explained and their results are compared with experiments. Data for helium and two phase flows are shown as getting attractive, but more work in these fields is required in the future.

NOMENCLATURE

a	:	radius of pipe or characteristic length of pipe (m)
Cp	:	isobaric specific heat (J/kg K)
$\vec{F}$	:	body force (Pa/m)
f	:	nondimensional body force, or friction factor
G	:	temperature distribution in the cross-section
g	:	nondimensional temperature $= G/\tau a$
K	:	dynamical parameter
k	:	thermal conductivity (W/(m·K))
Nu	:	Nusselt number
P	:	pressure (Pa)
p	:	nondimensional pressure $= (a^2/\nu^2)(P/\rho)$
Pr	:	Prandtl number
R	:	radius of curvature (m)
Ra	:	Rayleigh number
Re	:	Reynolds number
r	:	radial coordinate (m)
T	:	temperature (K)
Tw	:	wall temperature (K)
$\vec{V}$	:	velocity vector (m/s)
U,V,W	:	components of $\vec{V}$ in X, Y and Z directions (m/s)

u,v,w : nondimensional velocity $= (U,V,W)\times(a/\nu)$
X : axial coordinate (m)
Y : coordinate in the body force direction (m)
Z : coordinate in the cross-section (m)
x,y,z : nondimensional coordinate $= (X,Y,Z)\times(1/a)$
β : expansion coefficient (1/K)
δ : boundary layer thickness (m)
η : nondimensional radial coordinate $= r/a$
μ : viscosity (Pa s)
ν : kinematic viscosity $= \mu/\rho$ (m^2/s)
ρ : density (kg/m^3)
τ : temperature gradient in the axial direction (K/m)
ψ : angular coordinate in the cross-section (rad.)
Ω : angular velocity (1/s)
ω : nondimensional angular velocity $= 2a^2\Omega/\nu$
κ : thermal diffusivity (m^2/s)

Superscript

$\overline{\ }$: mean value in Z direction
$'$: perturbed component by secondary flow

Suffix

δ : outer edge of boundary layer
1 : main flow

Other Symbols

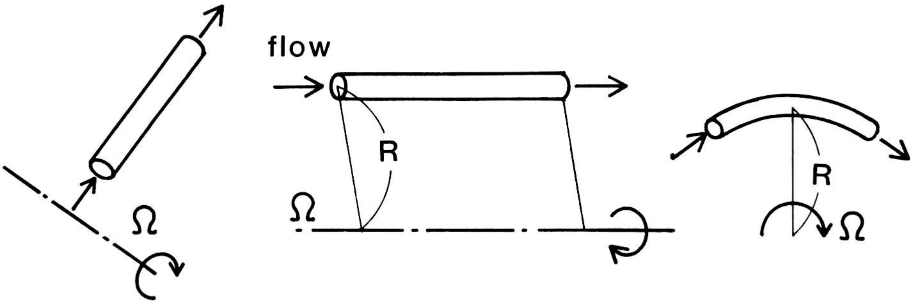
< > : mean value in Y·Z section

1. INTRODUCTION

Heat transfer in rotating ducts and pipes has become a subject of great importance for engineers in various industries. Research works on coolant flow and heat transfers in turbine blades have been intensified with the aim of raising the operating temperature of gas turbines. As the electric generator in power stations has increased in capacity, cooling of the rotating field windings has called for important design consideration. Recently, the research and development of a superconducting generator is being carried out in several countries, for which the flow of liquid helium in the cryogenic rotor is an important research subject. Basides those advanced technologies, conventional rotating machines also pose cooling problems as they are housed in noiseproof or dustproof containers.

Those diverse industrial needs have motivated the present authors to conduct theoretical and experimental studies in the past years. This paper presents a summary of our works, together with an updated review of the works made by other investigators.

In the coolant passages of rotating systems, the fluid is subject to a centrifugal force or a Coriolis force. Those forces, where they act in the direction of fluid passage, either accelerate or retard the flow. Where the forces act in the transverse direction to the flow, they cause secondary flows, which increase the heat transfer coefficient on the passage wall and the resistance to a coolant flow. The secomdary flows are the most distinguishing phenomena in heat transfers in rotating pipes and ducts, and are our primary concern



(a) Radial rotation (b) parallel revolution (c) Circumferential revolution

Fig. 1 Principal orientation of rotating and revolving duct flows

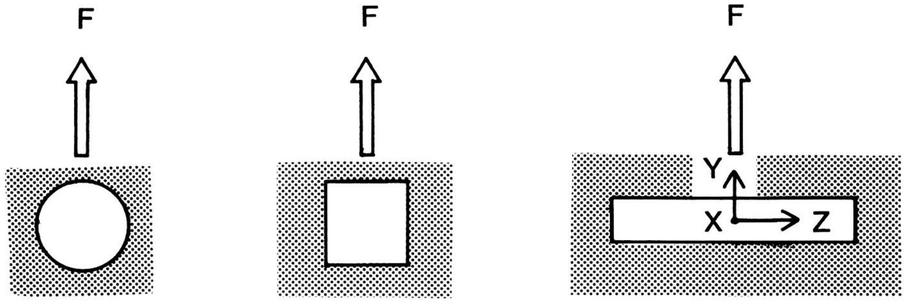
	radial rotating duct	parallel revolving duct	circumferential revolving duct
source of body force	Coriolis force acting on a primary velocity	density variation in a centrifugal field	density variation in a centrifugal field + centrifugal force due to curvature of duct
dynamical parameter	$Re \cdot \omega = (W_m d/\nu) \cdot (2a^2\Omega/\nu)$	$Ra = R\Omega^2 \beta \Delta T a^3/\kappa\nu$	$Ra = R\Omega^2 \beta \Delta T a^3/\kappa\nu$ $Re\sqrt{a/R} = (\frac{W_m d}{\nu})\sqrt{\frac{a}{R}}$

Table 1 Body forces and dynamical parameters for single-phase flow

in the present paper, because they pose challenging problems to researchers and the accurate estimate of their effects on heat transfer performances is important for cooling system designs.

Fig. 1 shows three principal orientations of flow passage against the axis of rotation taking a circular pipe as a representative passage. We call those passages (a) a radial rotating duct, (b) a parallel revolving duct, and (c) a circumferential revolving duct. A coolant passage in real rotating machines is equivalent to any of those ducts, or it is formed by joining them in a series of bends. The body forces of primary importance which cause secondary flows are listed in Table 1. For single phase flows, the dynamical parameters listed in Table 1 signify the intensity of secondary flows. They and conventional parameters of Reynolds number, Prandtl number and Graetz number make up a set of parameters needed to define flow and temperature fields.

As for the cross-sectional geometry of the coolant passage, the configurations of practical importance are those illustrated in Fig. 2, (a) circular, (b) square, and (c) narrow rectangular. In general, the body force F acts in any direction depending on the posture of a cross-section with respect to the axis of rotation. However, the practically important directions of forces are those of Fig. 2.



(a) circular (b) square (c) narrow rectangular

Fig. 2 Cross-section of flow passage and action of body force

In the following section 2, the fundamental cases of fully developed flows will be discussed. That is followed by a summary of recent numerical studies. In the section 3, the experimental works will be reviewed. The section 4 is devoted to a review of recent works on two-phase cooling schemes.

2. SINGLE-PHASE FLOW AND HEAT TRANSFER — THEORY

Equation of momentum and energy for the flow under the influence of a body force $\vec{F}$ are shown by equations (1) and (2);

$$\rho(\vec{V}\cdot\nabla)\vec{V} = -\nabla P + \mu\nabla^2\vec{V} + \vec{F} \tag{1}$$

$$\rho\, c_p\, (\vec{V}\cdot\nabla)T = k\nabla^2 T \tag{2}$$

respectively, where $\vec{V}$ is the velocity vector, T the temperature, P the pressure, and other notations are listed in NOMENCLATURE. The assumption of constant physical properties is adopted for the sake of simplicity to explain the features of secondary flow in rotating ducts.

Equation (1) is resolved into the equation for a primary flow (velocity component u) and those for secondary flow (v, w). In the cases of a radial rotating duct and a circumferential revolving duct, the momentum equation is solved taking the body force into account except the effects of physical property variation due to temperature distributions. On the other hand for a parallel revolving duct, the energy and momentum equations are coupled through the body force term. In order to solve a set of those coupled equations, one must resort to either one of the following analytical methods.

(1) Perturbation method.
(2) Boundary layer method.
(3) Energy balance and Entropy balance methods.
(4) Direct numerical integration by either finite difference methods or finite element methods.

The perturbation method suits the analysis of flows in circular pipes accompanied by weak secondary flows. Fig. 3(a) shows the secondary flow pattern, the axial velocity and temperature distributions displaced only slightly from the axially symmetric distributions. When the secondary flow is caused with a considerable intensity, the velocity and temperature distributions are distorted remarkably from symmetric ones, as shown in Fig. 3(b). In such cases, the boundary layer modeling proves to be a powerful analytical tool, and this has been used extensively by the present authors for laminar as well as turbulent flows [1, 2, 3, 4]. For the cross-sectional geometries of square and rectangle, the perturbation method is difficult to apply because the expansion in terms of the perturbation parameters becomes complex. The boundary layer method requires a rather complex modeling. The method of more integral nature based on the balance of energy and entropy was proposed to handle the problems of square and rectangular ducts [5]. The above analytical methods have merits in yielding concise and general correlations among friction factors, heat transfer coefficients and the dynamical parameters. The recent advances in numerical analysis described later on have given a well founded proof to the boundary layer modeling, and also provide one with detailed features of flow and temperature fields. Numerical analyses have successfully correlated the results obtained by perturbation methods for small dynamical parameter and boundary layer method for large parameter.

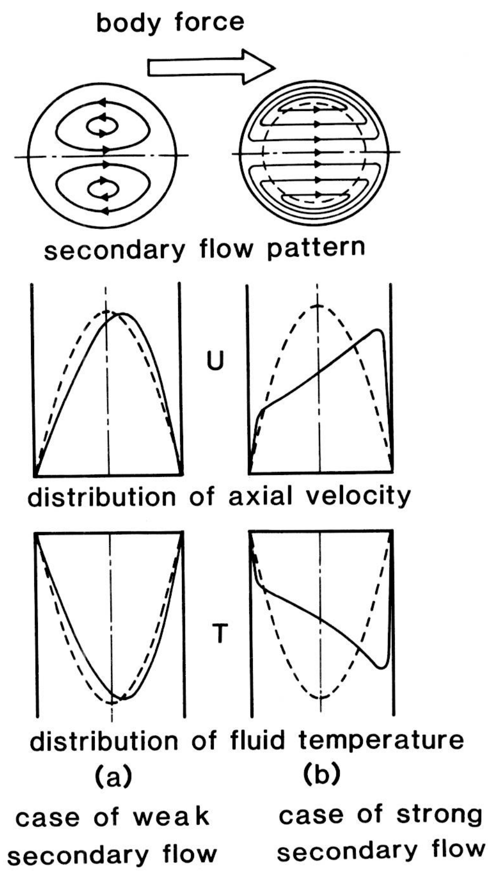


Fig. 3 Secondary flow pattern and distributions of axial velocity and temperature

In what follows, the dynamical parameter will be denoted by ε or K for the sake of conciseness. It should be noted, however, although equations can be formulated in such general terms, quantative results differ for different cases. This is also pointed out experimentally by Trefethen, as quoted in [6]. Recently, progresses for numerical analysis have been extending the applicable region of numerical results to large dynamical parameter for which so far the boundary layer method has been only the way of analysis of secondary flows. However, it should be noted that the boundary layer method can produce general relations for secondary flows when numerical analysis is not easily applicable.

2.1. Fully-Developed Flows with Weak Secondary Flows

(1) Perturbation method. For laminar and fully developed flows, the non-dimensionalization of equations (1) and (2) leads to expansion of the velocity components and the temperature in terms of a dynamical parameter listed in Table 1, denoted here by ε to signify the smallness of the parameter. In terms of the characteristic length a, the kinematic viscosity ν, the non-dimensional