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Turbulent flow and transient convection in a semi-annular duct

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Turbulent flow and heat transfer in an annular geometry have been previously studied experimentally or numerically. Velocity and temperature profiles have been measured and correlations for the wall shear stress and heat transfer have been derived. However there exists no study in turbulent flow for a semi-annular geometry. This work aims to study steady and transient convection in a semi-annular test section for a wide range of Reynolds numbers from 10,000 to 60,000, the inner cylinder being heated by Joule effect. The velocity profile in the symmetry plane is measured by Particle Image Velocimetry and the temperature of the inner heated cylinder is measured by infrared thermography. The experimental results are complemented by numerical simulations which give also access to the velocity and temperature profiles in the whole test section.

These results are compared to those obtained in an annular geometry for the same inner and outer cylinders radii and an equivalent flow rate. The velocity and temperature profiles and the wall shear stress are the same as in an annular section in an angular sector of $\pi/2$ around the symmetry plane. Both velocity and temperature profiles follow a logarithmic law. In steady convection, the local heat transfer has been characterized in several azimuthal positions. The local Nusselt number can be expressed versus a Reynolds number based on the local friction velocity. Characteristic thermal boundary layer thicknesses are also defined. Finally, transient convection tests are performed with a square power generation. The wall heat transfer and the evolution of the liquid temperature near the wall have the same self-similar evolution, with a characteristic time scale, which only depends on the flow Reynolds number.

1. Introduction

The core of a Pressurised Water Reactor (PWR) contains thousands of cylindrical fuel rods. Between them, some control rods allow controlling the nuclear reaction. All these rods are immersed in high pressure water that acts as coolant and moderator fluid. If a control rod is accidentally ejected, a local and sudden reactivity increase happens leading to a reactivity initiated accident (RIA). It leads to an increase of the rod temperature with first convective heat transfer and then boiling heat transfer. There is still a lack of knowledge for the rod's clad to coolant heat transfer in those transient conditions. The present study takes part of the research program defined by IRSN (Institut de Radioprotection et de Sûreté Nucléaire) on this topic.

For a RIA, the wall to fluid heat transfer till the onset of nucleate boiling occurs over a very short time scale. Therefore, only the very near wall flow and heat transfer has a relevant impact on the fuel rod behavior. The transient is thus studied only around a single rod surrounded by a vertical flow. The latter is idealized as a flow in an annular cross section with equivalent hydraulic diameter of the typical sub-channel defined between neighboring rods (in yellow in Fig. 1). To conduct experiments in a simple geometry but similar to this sub-channel, an experimental set-up was designed by Visentini et al. [1]. To measure the wall temperature with a high temporal and spatial precision, the infrared thermography is used and so an optical access to the wall is needed. Therefore a half-annular cross section with the heated inner half cylinder has been preferred to the annular one. Before the boiling incipience, heat is transferred from the wall to the liquid by single-phase convection. It is thus necessary to characterize the velocity and temperature fields and heat transfer in a semi annular geometry. Depending on the power generation in the heated wall, the convection is either steady or transient.

In the literature, the studies concerning heat transfer in annular-

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Nomenclature

Greek symbols

α	Thermal diffusivity, $\alpha = \frac{\lambda}{\rho C_p}$, ($m^2\ s^{-1}$)
β	Isobar expansion coefficient (K^{-1})
δ	Boundary layer thickness, (m)
γ	Apex angle of the cross section
κ	Von Karman constant
λ	Thermal conductivity, ($W\ m^{-1}\ K^{-1}$)
μ	Dynamic viscosity, (Pa s)
V	Cinematic viscosity, ($m^2\ s^{-1}$)
ϕ	Heat flux, ($W\ m^{-2}$)
Φ_h	Enthalpy flux, ($W\ m^{-1}$)
ρ	Density, ($kg\ m^{-3}$)
τ	Total shear stress, ($m^2\ s^{-2}$)
θ	Azimuthal angle

Latin symbols

$\mathscr{I}$	Current (I)
$\mathscr{P}$	Generated power, $\mathscr{P} = \mathscr{U}\mathscr{I}$, (W)
$\mathscr{U}$	Voltage (V)
$\frac{dp}{dz}$	Pressure gradient, ($Pa\ m^{-1}$)
A,A'	Constants
Bi	Biot number, $Bi = \frac{h\, e_w}{\lambda_w}$
C_p	Specific heat capacity, ($J\ kg^{-1}\ K^{-1}$)
D_h	Hydraulic diameter (m)
e_w	metal foil thickness, (m)
f_D	Darcy friction coefficient, $f_D = -\frac{dp}{dz}\frac{D_h}{0.5\rho u_m^2}$
f_F	Fanning friction factor, $f_F = \frac{\tau_{w,m}}{0.5\rho u_m^2}$
g	Gravity acceleration, ($m\ s^{-2}$)
Gr	Grashof number, $Gr = \frac{g\beta\Delta T D_h^3}{\nu^2}$
h	Heat transfer coefficient, $h = \frac{\phi_w}{T_w - T_l}$, ($W\ m^{-2}\ K^{-1}$)
k_e	Turbulent kinetic energy, $k_e = 0.5\langle u'^2 + v'^2 + w'^2\rangle$, ($m^2\ s^{-2}$)
Nu	Nusselt number, $Nu = \frac{\phi_w D_h}{\lambda(T_w - T_b)}$
Pr	Prandtl number, $Pr = \frac{\nu}{\alpha}$
Q	Flowrate ($m^3\ s^{-1}$)
r	Radial position (m)
r^*	Dimensionless radial position, $r^* = \frac{r - r_i}{r_o - r_i}$
r_0	Radial position of the zero of total shear stress (m)
Re	Reynolds number, $Re = \frac{\rho u_m D_h}{\mu}$
S	Surface section (m^2)
T	Temperature, (K)
t	Time, (s)
T^*	Dimensionless temperature, $T^* = \frac{T_w - T_b}{T_{w,stat} - T_b}$
T^+	Dimensionless temperature, $T^+ = \frac{T_w - T_l}{T_f}$
t_c	Characteristic time to establish stationary convection, (s)
T_f	Friction temperature, $T_f = \frac{\phi_w}{\rho C_p u_i^*}$, (K)
u	Axial velocity, ($m\ s^{-1}$)
u',v',w'	Turbulent velocities, ($m\ s^{-1}$)
u^*	Friction velocity $u^* = \sqrt{\frac{\tau_w}{\rho}}$, ($m\ s^{-1}$)
u^+	Dimensionless axial velocity, $u^+ = \frac{u}{u^*}$
u_m	Mean velocity $u_m = Q/S$, ($m\ s^{-1}$)
y	Radial position from the inner wall, $y = r - r_i$(m)
y^+	Dimensionless position, $y^+ = \frac{\rho y u^*}{\mu}$

Subscripts

θ	At a given azimuthal position
b	Bulk
eq	Equivalence between annular and semi-annular geometries
exp	Experimental
gen	Generated by Joule effect
i	Inner wall
$inlet$	Inlet
l	Liquid
m	Mean
max	Maximum
o	Outer wall
$simul$	Simulated
$stat$	Stationary
th	Theoretical
w	Wall

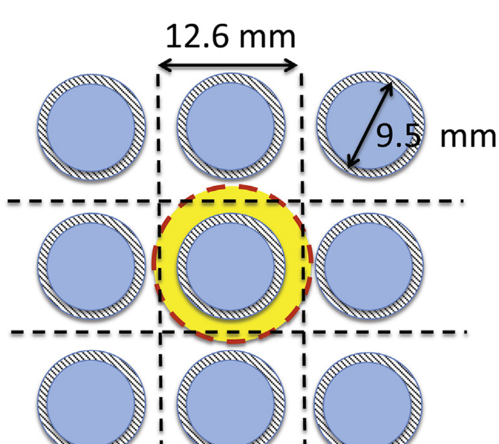

Fig. 1. Scheme of fuel rods and sub-channel.

sector ducts mainly concern heat exchangers. The cross section (Fig. 2) is characterized by the inner and outer wall radii r_i and r_o and the apex angle γ. Sparrow et al. [2], Soliman et al. [3] and Ben Ali et al. [4] studied heat transfers in sector-annulus with apex angles from 5° to 350°. However, these studies are all in laminar flow. Several studies were also performed in turbulent flows with different inner and outer radii, apex angle, Reynolds number Re, fluids, heated walls (Table 1). Tao et al. [5] and Li et al. [6] studied turbulent flow and heat transfer in sector-annulus with apex angles from 18° to 40°. The boundary conditions for the heat transfer are different from ours because they heat the outer wall. Li et al. found that the friction factor increases with the apex angle and varies from 0.03 to 0.02 when the Reynolds number Re ranges from 10,000 to 50,000 for a 40° apex angle.

Another geometry of interest is the annular geometry, for which there are a lot of studies. Kang et al. [7] conducted experiments for isothermal and heated turbulent upward flow in a vertical annular channel. The inner part is heated with a steady heat flux and the fluid is a refrigerant R-113. They measured liquid velocity profiles with laser Doppler velocimetry and the temperature field with a cold wire. Using the same test section, Hasan et al. [8] deduced a correlation for the Nusselt number Nu versus the Reynolds number Re and the Prandtl number Pr:

$$Nu = 0.0106 Re^{0.88} Pr^{0.4} \quad (1)$$

It is known that the correlations in convection depend on the