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Enhancement of heat transportation by oscillatory flow in a curved tube

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ABSTRACT

The present paper deals with the longitudinal heat transportation by an oscillatory flow in a curved tube heat transportation pipe. The velocity and temperature fields during oscillatory water flow in a curved tube with inner diameter of 10 mm were numerically simulated using the commercial software FLUENT. To clarify the effect of the curvature on heat transportation, the ratio of the tube radius to the curvature radius was varied from 0.0125 to 0.075, and oscillatory flow frequency was varied from 0.1 to 2.0 Hz. The effective thermal diffusivity exhibited a peak at a curvature ratio around 0.05–0.07 and reached a value 12 times higher than that of a straight tube. It was also found that the dispersion of fluid particles due to secondary flow caused the enhancement of heat transportation.

1. Introduction

Heat transportation devices that are small and high-performance are required for the effective cooling of electronic devices and equipment. Heat transportation devices that utilize reciprocal flow technology are currently considered prime candidates for this function.

Since the heat transportation pipe utilizing reciprocal flow was invented by Kurzweg and Zhao [1], many researchers have attempted to improve its performance. Kaviany [2] pointed out that its heat transportation is enhanced by the utilization of unsteady heat transfer between the pipe wall and the reciprocal flow. Nishio et al. [3,4] investigated the conditions for achieving the optimum performance of the reciprocal-flow heat transportation pipe: that is, they investigated extensively the effects of the thermal properties of the working fluid, the pipe diameter, the amplitude, and frequency of the reciprocal flow, and the flow pattern (laminar or turbulent) on the heat transportation performance of the reciprocal flow. Based on these investigations, they proposed a new kind of heat transportation pipe, an inverse oscillation-phase heat transport tube, which can achieve higher heat transportation performance than a normal reciprocal-flow heat transportation pipe [5]. In addition, the authors have previously reported that it is possible to improve heat-transportation performance by using annular channels [6], grooved ducts [7], and double tubes [8].

In other work, Pedley and Kamm [9] studied mass transport when oscillatory flows are induced within curved tubes. They investigated the amount of mass transport as the oscillation flow frequency and the tube curvature were varied and reported that sometimes effective mass diffusivities were obtained that were significantly higher than those observed for straight tubes. These authors attributed the high values they observed for the effective mass diffusivity in the tube to the *convective resonance* occurring in association with a unidirectional flow component, which exhibits no reciprocating motion even in the presence of oscillatory flow, due to agreement between the period of the secondary flow circulation arising within the tube and the period of oscillation of the axial flow.

The objective of this study is to perform numerical analyses of heat transportation through oscillatory flow in curved tubes. In addition to investigating how the heat transportation in the direction of the tube axis varies with the curvature ratio of the tube and the frequency of oscillatory flow, we also visualize the flow patterns to study the mechanisms of the heat transport.

2. Numerical methods

Fig. 1 shows the analytical models of the present curved tubes, which had an inner diameter of 10 mm, outer diameter of 12 mm, and a tube length of 314 mm. Denoting the tube inner radius by a and the curvature radius by b, we considered tubes with six distinct values of the ratio a/b: $a/b = 0$ (a straight tube), 0.0125, 0.025, 0.05, 0.0667, and 0.075. The working fluid was water (Prandtl number = 7.0, thermal diffusivity $\alpha = 0.14 \times 10^{-6}$ m^2/s, kinematic viscosity $\nu = 1.00 \times 10^{-6}$ m^2/s, density $\rho = 1000$ kg/m^3). The tube was made of acrylic resin material ($\alpha = 0.12 \times 10^{-6}$ m^2/s). The flow was assumed to be 3D, laminar, unsteady, and

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Nomenclature

a	tube inner radius	Q	heat transportation rate
a/b	curvature ratio	S	cross-sectional average secondary velocity
b	radius of curvature	t	time
D_{eff}	effective thermal diffusivity in the longitudinal direction	T	oscillatory period
f	frequency of oscillatory flow	u	flow velocity
f_s	frequency of secondary flow circulation	θ	temperature
L	tube length	σ^2	variance of fluid particles in the longitudinal direction

incompressible. In addition, we assumed that the flow is plane-symmetric with respect to the plane containing the tube axis and center of curvature. The governing equations were the equations for the conservation of mass, momentum, and energy. The energy equation was solved in the liquid column and at the tube walls respectively.

$$\frac{\partial u_x}{\partial x}+\frac{\partial u_y}{\partial y}+\frac{\partial u_z}{\partial z}=0, \tag{1}$$

$$\frac{\partial u_x}{\partial t}+u_x\frac{\partial u_x}{\partial x}+u_y\frac{\partial u_x}{\partial y}+u_z\frac{\partial u_x}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left(\frac{\partial^2 u_x}{\partial x^2}+\frac{\partial^2 u_x}{\partial y^2}+\frac{\partial^2 u_x}{\partial z^2}\right), \tag{2}$$

$$\frac{\partial u_y}{\partial t}+u_x\frac{\partial u_y}{\partial x}+u_y\frac{\partial u_y}{\partial y}+u_z\frac{\partial u_y}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left(\frac{\partial^2 u_y}{\partial x^2}+\frac{\partial^2 u_y}{\partial y^2}+\frac{\partial^2 u_y}{\partial z^2}\right), \tag{3}$$

$$\frac{\partial u_z}{\partial t}+u_x\frac{\partial u_z}{\partial x}+u_y\frac{\partial u_z}{\partial y}+u_z\frac{\partial u_z}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+v\left(\frac{\partial^2 u_z}{\partial x^2}+\frac{\partial^2 u_z}{\partial y^2}+\frac{\partial^2 u_z}{\partial z^2}\right), \tag{4}$$

$$\frac{\partial \theta}{\partial t}+u_x\frac{\partial \theta}{\partial x}+u_y\frac{\partial \theta}{\partial y}+u_z\frac{\partial \theta}{\partial z}=\alpha\left(\frac{\partial^2 \theta}{\partial x^2}+\frac{\partial^2 \theta}{\partial y^2}+\frac{\partial^2 \theta}{\partial z^2}\right). \tag{5}$$

As a boundary condition, a sinusoidal reciprocal flow with a frequency of f (=0.1, 0.5, 1.0, 1.5, 2.0 Hz) and a displacement amplitude of s = 0.045 m was imposed from one end of the tube. Water with a uniform temperature of θ = 323 K flowed into the tube when the longitudinal flow velocity was positive and water with zero temperature gradient flowed out of the tube when the velocity was negative. At the other end of the tube, the pressure was constant ($p = 1.01 \times 10^5$ Pa) and the longitudinal velocity gradient was zero. Water with zero temperature gradient flowed out of the tube when the velocity was positive and water of θ = 293 K flowed into the tube when the longitudinal flow velocity was negative. The velocity at the inner wall surface of the tube was assumed to be zero, with insulating conditions applied at the outer wall. Within the symmetry plane, the gradients of the normal components of all the physical variables were zero.

The computational mesh consisted of a half cross-section of the curved tube with 30 subdivisions in the radial direction, 30 subdivisions in the circumferential direction, and approximately 450 subdivisions in the longitudinal direction. Fig. 2 depicts the computational mesh for the fluid part within the half cross-section. The radial subdivisions were more finely spaced as the tube wall was approached, taking into account the steep velocity gradient near the wall generated by the oscillatory flow. The subdivisions in the circumferential and longitudinal directions were evenly spaced. The portion of the mesh corresponding to the tube wall consisted respectively of 10 radial subdivisions and 30 circumferential subdivisions, both evenly spaced. The number of mesh elements for the curved tube overall was between 376,800 and 405,600. The computational meshes were generated using GAMBIT 6.3.26 meshing software.

We used the commercial code FLUENT 13.0 to conduct the numerical analysis, adopting the finite difference method for the discretization, the first-order upwind difference scheme for the convective term, the first-order accuracy scheme for the pressure correction, the SIMPLE method for the coupling of pressure and velocity, and the implicit method of the first-order accuracy scheme for the discretization of time.

To assess the adequacy of our computational mesh, we conducted simulations with parameter values a/b = 0.025 and f = 0.5 Hz using three meshes in which the tube cross-section was subdivided into 20×20, 30×30, and 40×40 mesh elements, and compared the results. Taking the results of the 30×30 mesh as a reference point, we found the heat transportation to differ by −26.5% in the 20×20 case and by −1.2% in the 40×40 case. Thus the difference between the 30×30 and 40×40 meshes is small, so we concluded that the results for the 30×30 mesh were adequate for our studies.

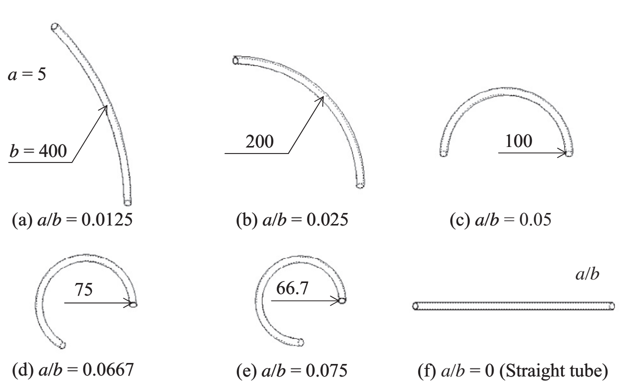

Fig. 1. Computational model.

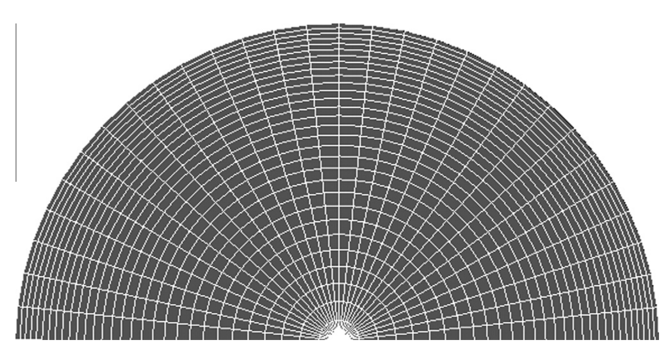

Fig. 2. Computational mesh cross-section.