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HEAT TRANSFER FROM A DISCRETE HEATED SURFACE IN A 2D CHANNEL NEAR AN OSCILLATING INTERFACE

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ABSTRACT

Numerical simulations of the local heat transfer from a discrete heat source in a 2D channel near an oscillating gas/liquid interface were carried out by using the software package PHOENICS. The flow fields and heat transfer coefficients obtained are in reasonable agreement with the experimental results. It was found that changes in the interfacial surface tension from 0 to 73×10^{-3}N/m, have a relatively small influence on the local heat transfer rate.

NOMENCLATURE

A	oscillating amplitude
C_p	specific heat capacity of the fluid
f	oscillating frequency
g	gravity acceleration
h	average heat transfer coefficient
H_{ch}	channel length
H_{f2}	height of phase two
H_p	height of expansive section
k	thermal conductivity of the fluid
l	length of the heated section
Nu	Nusselt number, Nu=hl/k
p	pressure
Re_o	oscillating Reynolds number $(2/\pi)(W_o l/\nu)$
t	time
T	temperature
T_w	wall temperature on the heated section
v	y component velocity
w	z component velocity
W_0	maximum piston velocity
W_c	width of the channel
y	channel normal direction coordinate
z	channel axial coordinate

Greek symbols

μ	dynamic viscosity
ν	kinematic viscosity
ρ	density
σ	surface tension
ω	oscillating angular frequency

INTRODUCTION

Interfaces undergoing wave motion are encountered in many metallurgical processes, such as the slag/metal interface in a steel tundish for continuous casting and the electrolyte/metal interface in a Hall-Héroult aluminium reduction cell[1]. The local disturbances on the flow field due to the interfacial wave movement significantly affect the local heat and mass transfer characteristics. So far, the flow and transport characteristics near a solid wall due to an interface undergoing wave motion have not been extensively investigated. Sha *et al*[2] and Ting and Perlin[3] have studied the dynamic behaviours of an interface near a vertically oscillating wall. These investigations were focused on the dynamic behaviour of the three phase contact line (TPL) and the transport characteristics had not been studied. Chen *et al*[4] and Chen and Chen[5,6] experimentally investigated the heat transfer characteristics near an interface undergoing wave motion and found that the heat transfer rate near the interface was significantly higher than that for uni-directional flows over a flat plate.

The heat transfer enhancement was found to be due to the periodic renewal of the boundary layer and the high initial heat and mass transfer rate during the establishment of a new temperature or concentration field. Figure 1 gives typical flow patterns near an oscillating air/water interface obtained from an earlier visualization study[4]. Figure 2 shows a schematic illustration of the transport process.

During the half oscillating period when the interface

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moves downwards as shown in Fig. 1a and 2a, the water in the boundary layer is peeled off the wall and transferred into the bulk liquid. With the scraping-off of the boundary layer, the temperature profile near the wall is destroyed and the heat in the boundary layer is transferred into the bulk water.

When the direction of the interfacial movement reverses, a new boundary layer is established along the solid wall. Figures 1b and 2b show that fresh bulk water is drawn into the wall region. Thus, after each oscillation cycle, a new boundary layer is formed with fresh bulk water and the water in the boundary layer from the previous oscillation cycle is replenished. With the start of the next oscillation cycle, the liquid exchange between the bulk and that in the wall boundary layer is repeated as shown on Fig. 1c and 2c, and the transport process continues with further oscillations.

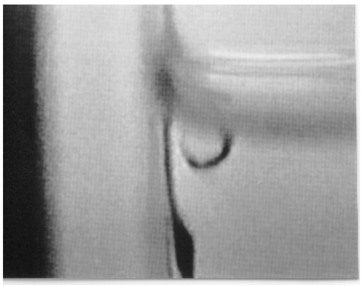

Figure 1a

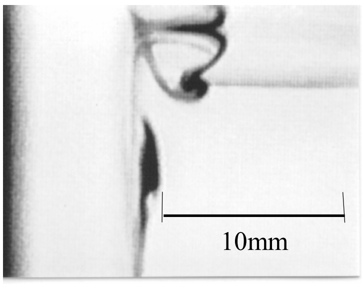

Figure 1b

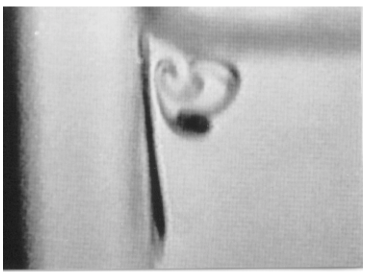

Figure 1c

Figure 1: Typical dye traces near an oscillating air/water interface for A=18.2mm and f=0.65Hz: (a) interface moving downwards, first cycle. (b) interface moving upwards, first cycle. (c) interface moving downwards, second cycle.

In this work, numerical simulations for the local heat transfer from a discrete heat source in a 2D channel near an oscillating gas/liquid interface were carried out using a commercial software package PHOENICS[7]. The flow fields and heat transfer results obtained numerically are compared with the experimental results obtained under similar conditions.

PROBLEM DESCRIPTION AND GOVERNING EQUATIONS

Figure 3 shows a schematic representation of the problem investigated. A liquid column with fluid1/fluid2 interface undergoes oscillating movement in a two dimensional channel.

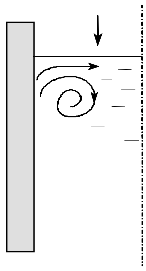

Figure 2a

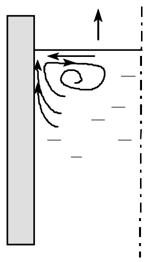

Figure 2b

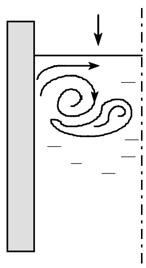

Figure 2c

Figure 2: Schematic view of the transport process, (a) interface moving downwards, first cycle; (b) interface moving upwards, first cycle; (c) Start of the second cycle.

An expansive section at the bottom of the channel acts as a piston and drives the oscillating movement of the liquid column. A discrete heated section with a constant wall temperature T_w is located on the wall with its centre at the equilibrium interfacial position.

Newtonian, incompressible two-dimensional flows with constant physical properties were assumed in this work. For such an unsteady flow system, neglecting viscous dissipation, equations governing the mass, momentum and energy transport can be simplified as:
Continuity conservation

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho w}{\partial z}=0 \quad \textbf{(1)}$$

Momentum conservation

$$\frac{\partial \rho \mathrm{v}}{\partial \mathrm{t}}+\frac{\partial \rho \mathrm{wv}}{\partial \mathrm{z}}+\frac{\partial \rho \mathrm{v}^2}{\partial \mathrm{y}} =\frac{\partial}{\partial z}\left(\mu\frac{\partial \mathrm{v}}{\partial \mathrm{z}}\right)+\frac{\partial}{\partial y}\left(\mu\frac{\partial \mathrm{v}}{\partial \mathrm{y}}\right)-\frac{\partial \mathrm{p}}{\partial \mathrm{y}} \quad \textbf{(2)}$$

$$\frac{\partial \rho \mathrm{w}}{\partial \mathrm{t}}+\frac{\partial \rho \mathrm{w}^2}{\partial \mathrm{z}}+\frac{\partial \rho \mathrm{vw}}{\partial \mathrm{y}} =\frac{\partial}{\partial y}\left(\mu\frac{\partial \mathrm{w}}{\partial \mathrm{y}}\right)+\frac{\partial}{\partial z}\left(\mu\frac{\partial \mathrm{w}}{\partial z}\right)-\frac{\partial \mathrm{p}}{\partial \mathrm{z}}+\rho g \quad \textbf{(3)}$$

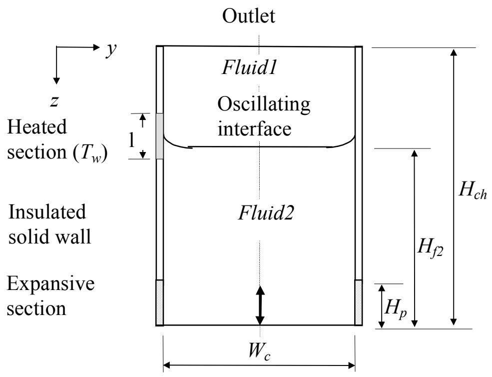

Figure 3 A schematic view of the heat transfer from a discrete heated wall near an oscillating interface.

Energy conservation

$$\rho\frac{\mathrm{dC_p}T}{d\mathrm{t}}=\frac{dp}{dt}+\frac{\partial}{\partial z}\left(\mathrm{k}\frac{\partial \mathrm{T}}{\partial \mathrm{z}}\right)+\frac{\partial}{\partial y}\left(\mathrm{k}\frac{\partial \mathrm{T}}{\partial \mathrm{y}}\right) \quad \textbf{(4)}$$

The boundary conditions are:
At the solid wall:

$$v = w = 0$$

The heated section:

$$\mathrm{T}\big|_{y=0} = T_W$$

Adiabatic wall outside the heated section

$$\mathrm{k}\frac{\partial \mathrm{T}}{\partial \mathrm{y}}\bigg|_{y=0} = 0$$

The bottom wall:

$$\mathrm{w} = \mathrm{W_0}\sin\omega\mathrm{t}, \mathrm{v} = 0, \frac{\partial T}{\partial z} = 0$$

Outlet:

$$\frac{\partial \mathrm{w}}{\partial \mathrm{z}}=0, \frac{\partial \mathrm{v}}{\partial \mathrm{z}}=0, \frac{\partial \mathrm{T}}{\partial \mathrm{z}}=0, \frac{\partial \mathrm{p}}{\partial \mathrm{z}}=0$$

NUMERICAL SOLUTION PROCEDURE

The numerical simulations were carried out using PHOENICS 2.11[7]. Liquid interface position was determined by the Height-of-Liquid (HOL) method[7]. With the HOL method, the entire liquid volume is divided into small vertical columns and the location of the interface for each liquid column is determined from the solution of the liquid-balance equations for the vertical liquid column. When the interface is not convoluted, the HOL needs no anti-dispersion device, and is simple, effective and economical. The upwind discretization scheme was used for the convection and advection terms. The transient terms were discretized by the first-order implicit method. As shown in Fig. 3, the oscillating interfacial movement was generated by the expansion and contraction of the expansive section at the bottom of the channel which drives the liquid column sinusoidally. The average heat transfer coefficient *h* was obtained by averaging the local heat transfer coefficient spatially and temporally over an oscillating cycle.

Simulation of heat transfer from a discrete heated wall near an oscillating interface is time-consuming using the available computer, IBM RS/6000. Hundreds of oscillating cycles and thousands of hours of CPU time are needed to achieve a thermal steady state for a system with heating on the wall near the interface and heat loss on the wall away from the heated section (which occurred in reality during the previous experiments[4].). In this work, the following approximate procedure was used:

First, starting from an initial position of the interface and a uniform temperature field in both the heavy and light fluids, simulation of about tens of oscillating cycles was carried out to achieve a hydrodynamic steady state without solving the energy equation. At this stage, the temperature in the channel remained at the uniform initial temperature. Then, the heat source was imposed on the wall near the interface and the energy equation was solved in one subsequent oscillating cycle. The walls were adiabatic except for the section with the heat source. An average heat transfer coefficient from the heated section was then obtained by averaging the local heat transfer over this oscillation cycle.

Such a procedure is based on the assumption that heat transfer of the previous oscillating cycle has little or no effect on that of its successive oscillating cycle. Since the mechanism of the heat transfer from the heated section near an oscillating interface is due to the periodic renewal of the fluid in the thermal boundary layer, the procedure adopted is a reasonable approximation.

The grid and time step used in this work were non-uniform 40×192 (y direction × z direction) grid system and 2400 uniform time steps. Study of grid refinement and time step refinement showed that further increase in the grid number and time step has essentially no influence on the simulation results.

RESULTS AND DISCUSSIONS

The geometry of the channel simulated is listed in Table 1.

Flow patterns and vortex formation near an oscillating air/water interface

For the investigation of the flow pattern near an oscillating interface, air/water interface was simulated under similar conditions for the visualisation experiments as shown in Fig. 1. The physical properties of the air and water at 20^{0}C are listed in Table 2.

Figure 4 show a series of flow fields near an oscillating air/water interface for A=18.2mm, f=1.53Hz.

Channel height H_{ch} (mm)	300
Channel width W_c (mm)	25
Height of liquid column H_{2f} (mm)	152.5

Table 1 Geometry of the 2D channel simulated.

	Air (20^{0}C)	Water (20^{0}C)
ρ (kg/m^3)	1.189	998.23
μ (kg/ms)	1.836×10^{-5}	1.004×10^{-3}
C_p (J/kg^{0}C)	1005	4182
k (W/m^{0}C)	0.0258	0.597

Table 2 Physical properties of air and water at 20^{0}C.

Compared to Figure 1, it is seen that the flow fields are in reasonable agreement with those obtained by previous visualization studies. In Fig. 4, a vortex is formed near the air/water interface when the interface moves downwards. The vortex is pushed away from the wall and transported into the bulk liquid when the interface moves upwards.

Effect of surface tension on heat transfer

The surface tension is implemented into the system according to the local curvature of the air/liquid interface. This was done by adding an additional momentum source to the node where the interface located. The correct implementation of the surface tension was verified by comparing the static meniscus height with analytical solution for an air/liquid interface with a surface tension of 73 mN/m and a static contact angle of 30^o.

It should be noted that there are several difficulties to implement the transient contact angle and the three phase contact line (TPL) movement in numerical simulations. In fact, there is no theory available for the TPL movement and transient contact angle during interface oscillation, especially when large Weber number (>1.) and Bond number (>1.) is encountered as it is in this work[3,8-10] and the Weber number and the Bond number represent the ratios of the inertial forces and the gravity forces to interfacial force respectively. Since both theoretical analysis and experimental correlation of TPL velocity for the current problem are unavailable, the simple non-slip boundary condition was used as the TPL boundary condition in this work. The contact angle was assumed to be determined by the inertial force, gravity force and the surface tension.

A comparison of the heat transfer results near an oscillating air/water interface obtained with and without surface tension are listed in Table 3. It can be seen that surface tension has a relatively small influence on the average heat transfer coefficient.

The mechanism of heat transfer near an oscillating interface is due primarily to the renewal of the fluid in the boundary layer. In other words, the overall heat transfer rate is mainly determined by the transient conduction from the heat source to the fluid nearby. Surface tension may have some influence on the flow fields near the oscillating interface, but does not change the transport mechanism. As long as surface tension does not significantly change the flow fields near the heated section, its influence on the local heat transfer is expected to be small.

A (mm)	f (Hz)	l (mm)	Surface tension (mN/m)	h (W/m^2 ^{o}C)
18.2	1.53	10	0.	2344.8
18.2	1.53	10	73.	2547.9

Table 3 The influence of surface tension on the local heat transfer coefficient.

Comparison between simulated and experimental heat transfer coefficients

In this work, numerical simulations of heat transfer from a discrete heated wall near an oscillating air/liquid interface were carried out for the air/liquid system with an amplitude range of 9.6 to 18.2mm, a frequency range of 0.5 to 2.5Hz and a liquid Prandtl number range of 1 to 7. In all these simulations, the centre of the heated section is located at the equilibrium interfacial position.

Fig. 5 shows the comparison of heat transfer coefficients obtained by numerical simulation for a constant Prandtl number of 4.2 and the predicted results by the experimental correlation, Eq. (5)[4]:

$$Nu = 0.399\,\mathrm{Re}_o^{\;0.5}\,\mathrm{Pr}^{0.5} \qquad \textbf{(5)}$$

Where

$$Nu = \frac{hl}{k}$$

$$\mathrm{Re}_0 = \frac{2}{\pi}\frac{W_0 l}{\nu}$$

It is seen that the heat transfer results obtained are in reasonable agreement with the experimental correlation.

Figure 6 shows the influence of Prandtl number on the average heat transfer coefficient for an amplitude of 18.2mm and a frequency of 1.53Hz. It is found that in the Prandtl number range investigated, the heat transfer coefficients are proportional to $Pr^{0.47}$ which is very close to the experimental correlation with $Nu \propto Pr^{0.5}$.

CONCLUSIONS

- The flow fields obtained by the present numerical model are in reasonable agreement with the dye patterns obtained by previous visualisation studies.
- Surface tension has a relatively small influence on the local heat transfer rate near an oscillating interface.
- Heat transfer results obtained numerically in this work are in reasonable agreement with previous experimental data.

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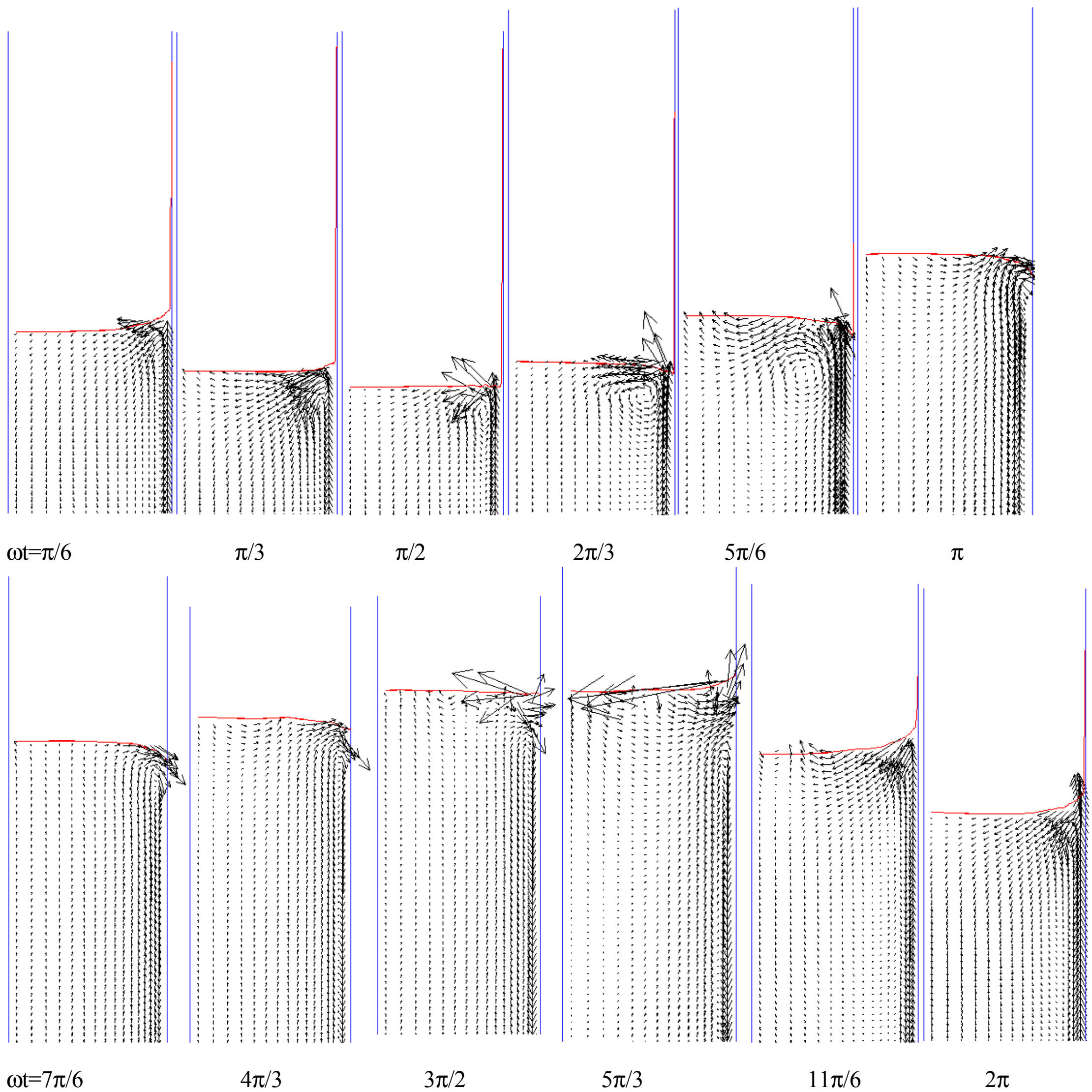

Figure 4 Simulated flow fields near an oscillating air/water interface under the conditions of A=18.2mm, f=1.53Hz.

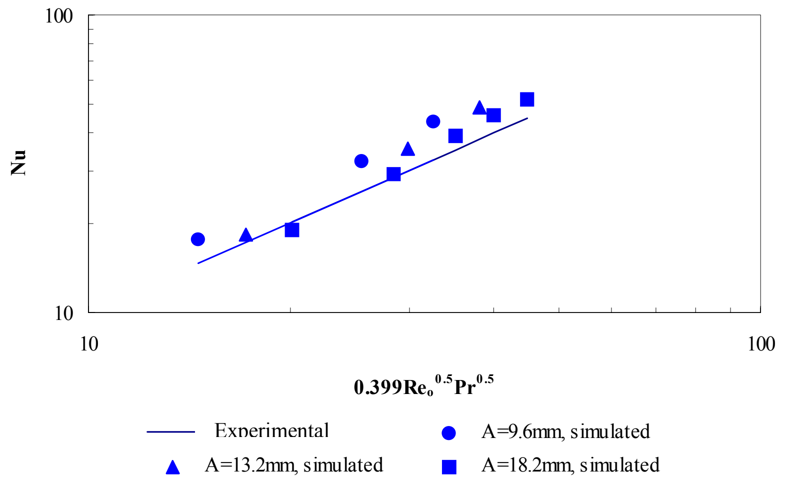

Figure 5 Comparison of simulated heat transfer results and those obtained by experimental correlation for air/water system, A=9.6-18.2mm and f=0.5Hz-2.5Hz.

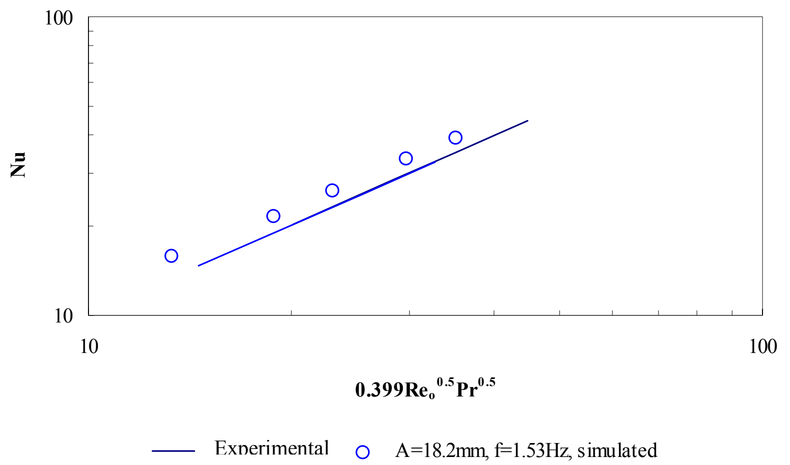

Figure 6 Comparison of simulated heat transfer results and those obtained by experimental correlation for liquid Prandtl number from 1 to 7, A=18.2mm and f=1.53Hz.