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Heat transfer rate and uniformity of mist flow jet impingement for glass tempering

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ABSTRACT

The thinning and miniaturization of components, such as liquid crystal display and solar cells, has increased the market demand for tempered ultra-thin glass. The physical tempering in glass, which is less than 2 mm thick, is difficult to achieve in air jet impingement. In this study, we propose the use of mist flow jet impingement cooling technology on ultra-thin glass. The effects of the mist flow droplets diameters, nozzle inlet temperature, and mass fraction on the heat transfer rate and uniformity are numerically studied. Our performance in terms of the heat transfer rate and the uniformity is accounted for through an evaluation of the surface-averaged temperature, averaged Nusselt number, surface-averaged Nusselt number, and surface standard deviation percentage of the Nusselt number. The empirical correlation for the surface averaged Nusselt number as a function of mist flow droplet diameters is provided. The droplet diameter is the major factor affecting the heat transfer rate, while the jet inlet temperature is a minor factor. We can obtain a better heat transfer uniformity by controlling the droplet diameter of the mist flow jet impingement. Using mist flow jet impingement, we would not only greatly reduce the amount of air and save energy, but also meet the needs of sudden cooling for tempered glass with a thickness of less than 2 mm.

1. Introduction

The thinning and miniaturization of components, such as liquid crystal display and solar cells, has increased the market demand for ultra-thin glass. In these applications, the ultra-thin glass should be tempered to ensure the strength and safety of final products. During the glass tempering process, glass is heated close to its melting temperature, and then exposed to a sudden cooling process with fluid jets. This sudden cooling results in a sharp contraction of the glass surface, thereby generating compressive stress in the outer regions of the glass, while simultaneously slowing tensile stress formation in the inner regions [1].

The jet impingement heat transfer characteristics of 4–6-mm-thick glass samples are presently being investigated [2–7]. The physical tempering in glass, which is less than 2 mm thick, is difficult to achieve in air jet impingement [8]. The jet impingement heat transfer during glass tempering is usually performed under conditions of high temperature difference and small jet-to-plate distance.

Glass tempering experiments on the water spray cooling method with different thicknesses (2.1–4.9 mm) [9–11] have also been studied. However, spray cooling can cause the uneven heat transfer [12]. The Non-uniformity of the glass surface temperature causes an uneven stress distribution in the glass, which can result in fragility.

Experimental studies [13–15] and simulations [16–20] on air-mist jet impingement heat transfer have mainly focused on other applications, with conditions being large jet-to-plate distances and a small temperature difference between the nozzle inlet and the target plate, instead of glass tempering. However, in obtaining ultra-thin tempered glass, we need not only a shorter cooling time and faster surface cooing during the glass tempering, but also a uniformity of the surface temperature, which has a large effect on glass tempering. Hence, the heat transfer rate and uniformity of the mist flow jet impingement in the glass tempering process are studied.

Most of the abovementioned studies focused on the enhancement of the heat transfer rate and the experiments on the glass

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Nomenclature

a_1, a_2, a_3	constant experimental parameters
B_i	Biot number
C_D	drag coefficient
C_p	specific heat capacity J/(kg K)
D	nozzle diameter (mm)
d_p	droplet diameter (µm)
F_x	additional acceleration force (N/kg)
h	heat transfer coefficient (W/m^2 k)
H	jet to impingement plate distance (mm)
h_{fg}	latent heat (J/kg)
$\dot{m}$	mass flow rate of the droplets (kg/s)
M_{L1}	mass fraction of droplets
N_u	Nusselt number
$N_{u_{ave}}$	averaged Nusselt number
$\overline{N_u}$	surface averaged Nusselt number
P_r	Prandtl number
q''	wall heat flux (W/m^2)
R_e	Reynolds number
S_h	source term of energy (J/m^3 s)
S_m	source term of mass (kg/m^3 s)
T	temperature (K)
ΔT	temperature difference (K)
u	fluid velocity (m/s)
x	axial distance along impingement plate (mm)
y	distance along impingement plate width (mm)

Greek symbols

ρ	density (kg/m^3)
λ	thermal conductivity (W/K m)
μ	dynamic viscosity (kg/m s)
ξ	heat transfer enhancement ratio
σ_{N_u}	surface standard deviation percentage of the Nusselt number

Subscripts

aw	adiabatic wall temperature
p	particle (water drolet)
Pr	pressure gradient effects
s	solid
∞	bulk flow

tempering using the method of air jet impingement heat transfer on large jet-to-plate distances. In this regard, we remark that very few studies have focused on glass tempering with the air/mist jet impingement and the heat transfer uniformity for the jet impingement heat transfer under conditions of small jet-to-plate distance and high temperature difference between the inlet and plate. Several empirical correlations were suggested for the air jet, while whose for the air/mist jet are still limited. This study aims to ensure ultra-thin glass tempering, which is helpful in investigating the effect of the mist parameters on the heat transfer rate and the uniformity of air/mist impinging jets under the condition of a small jet-to-plate distance. This study also propose the use of the surface standard deviation percentage of the Nusselt number as a quantitative estimator for the heat transfer uniformity. The surface-averaged temperature, averaged Nusselt number, surface-averaged Nusselt number, and surface standard deviation percentage of the Nusselt number obtained from the numerical studies are presented in an attempt to quantify the heat transfer rate and the uniformity. The current simulation is conducted at a fixed Reynolds number *Re* = 30,000, which is based on the hydraulic diameter of the circular jet [21], a fixed jet-to-plate spacing ($H/D = 0.2$) [38], the mist flow droplets diameters that varied from 5 to 20 µm, a nozzle inlet temperature that varied from 283 K to 303 K, and 5% and 10% mass fraction of the mist flow. These data will be useful in understanding the heat transfer rate and the uniformity of the mist flow jet impingement at a small jet-to-plate distance and in understanding that designing an air/mist jet impingement system has a certain theoretical importance.

2. Method

We use the Eulerian model (EM) and the discrete phase model (DPM) to simulate the continuous and discrete phase, respectively, in our two-phase flow scenario.

2.1. Governing equations of the continuous phase

The transient transport equations of mass, momentum, energy, and species are presented as follows [22]:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial X_i}(\rho u_i) = S_m \tag{1}$$

$$\frac{\partial}{\partial t}(\rho u_j) + \frac{\partial}{\partial X_i}(\rho u_i u_j) = \rho g_j - \frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i}\left(\tau_{ij} - \rho\overline{u_i' u_j'}\right) + F_j \tag{2}$$

$$\frac{\partial}{\partial t}(\rho C_p T) + \frac{\partial}{\partial X_i}(\rho C_p u_i T) = \frac{\partial}{\partial X_i}\left(\lambda\frac{\partial T}{\partial X_i} - \rho C_p\overline{u_i' T'}\right) + \mu\varnothing + S_h \tag{3}$$

$$\frac{\partial}{\partial t}(\rho C_j) + \frac{\partial}{\partial X_i}(\rho u_i C_j) = \frac{\partial}{\partial X_i}\left(\rho D_j\frac{\partial C_j}{\partial X_i} - \rho\overline{u_i' C_j'}\right) + S_j \tag{4}$$

where $\mu\varnothing$ is viscous dissipation; λ is thermal conductivity; C_j is the mass fraction of species j; S_j is the source term for this species; D_j is the diffusion coefficient of species j; $\rho\overline{u_i' u_j'}$, $\rho C_p\overline{u_i' T'}$ and $\rho\overline{u_i' C_j'}$ represent the Reynolds stresses, turbulent heat fluxes, and turbulent concentration (or mass) fluxes, respectively; τ_{ij} is the symmetric stress tensor [22]; and S_m, F_j, and S_h represent the source terms of mass, momentum, and energy, respectively. These terms are calculated via the following equations [37]:

$$S_m = \frac{\Delta m_p}{m_{p,0}}\dot{m}_{p,0} \tag{5}$$

$$F_j = \sum\left(\frac{18\mu C_d R_e}{\rho_p d_p^2 24}(u_p - u) + F_{other}\right)\dot{m}_p \Delta T \tag{6}$$

$$S_h = \left[\frac{\bar{m}_p}{m_{p,0}} c_p \Delta T + \frac{\Delta m_p}{m_{p,0}}\left(-h_{fg} + \int_{T_{ref}}^{T_p} C_{p,i}\, dT\right)\right]\dot{m}_{p,0} \tag{7}$$

where $\dot{m}_p$ represents the mass flow rate of the droplets; Δm_p is the total evaporated water mass in control volume; $\bar{m}_p$ is the average droplet mass in the control volume; and $m_{p,0}$ is the initial droplet mass at the inlet.