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Condensing-convective boundary conditions in moist air impingement ovens

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Abstract

Previously developed equations for impingement heat transfer coefficients experienced by a flat plate under an array of slot nozzles were compared to values measured for discrete model food products in a commercial moist air impingement oven. The best correlation predicted the convective heat transfer coefficients to an average absolute error of 1.9%, with a standard error of prediction (SEP) of 1.9 W/m^{2}K, when accounting for edge effects. Condensation heat transfer was successfully modeled as a convective mass transfer phenomenon. The analogous nature between heat and mass transfer was used to predict the mass transfer coefficients from correlations for convective heat transfer. The same correlation found to most accurately predict the convective heat transfer coefficient also most accurately predicted the mass transfer coefficient for impingement flow. When accounting for edge effects, the model predicted the mass transfer coefficient to an average absolute error of 12.7%, with an SEP of 16.4 mm/s.

Keywords: Impingement; Modeling; Condensation; Heat transfer; Mass transfer

1. Introduction

Impingement heat transfer has a variety of industrial applications, including: annealing of non-ferrous sheet metals, tempering of glass, drying of paper and textiles, the cooling of electrical components and turbine blades, and the processing of food products. Impingement flows appeal to these industries, because impingement produces heat transfer coefficients an order of magnitude higher than do other gaseous heat transfer methods (Saad, Mujumdar, & Douglas, 1980). Additionally, the combined effects of air impingement and water vapor condensation can be particularly valuable for certain applications, such as manufacturing ready-to-eat (RTE) meat and poultry products.

In terms of just convection, the heat transfer rates due to impingement flows can be varied either by adjusting flow rates and temperature differentials or by adjusting several geometric parameters. One of the most important parameters is the choice of jet configuration. There are four primary configurations: single round nozzle (SRN), single slot nozzle (SSN), array of round nozzles (ARN), and array of slot nozzles (ASN). The heat transfer rate achieved under ASN impingement, which is the focus of this study, is also affected by the geometry of the slot array, including jet width (w), jet-to-surface distance (H), and jet center-to-center distance ($2S$). General reviews of impingement heat transfer were presented by Mujumdar & Douglas (1972), Martin (1977), Obot, Mujumdar, & Douglas (1979), & van Heiningen (1982).

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Nomenclature

A	product surface area, m^2
Bi	Biot number, $hE/2k$
C	water vapor concentration, kg/m^3
c_p	specific heat, J/kg K
d	product diameter, m
D	mass diffusivity of water vapor in air, m^2/s
E	disk thickness, m
f	fraction nozzle open area, $w/2S$
H	nozzle-to-impingement surface spacing, m
h	convective heat transfer coefficient, $W/(m^2 K)$
h_{fg}	latent heat of condensation, J/kg
h_m	mass transfer coefficient, m/s
k	thermal conductivity, W/(m K)
m	mass, kg
M_v	impingement fluid moisture content, %
Nu	Nusselt number, hw/k
Pr	Prandtl number
q	heat transfer (kJ)
Re	Reynolds number, $uw\rho/\mu$
S	heat transfer surface half width, m
Sc	Schmidt number, $\mu/\rho D$
Sh	Sherwood number, $h_m w/D$
T	Block temperature, K
u	gas velocity, m/s
v	heat transfer surface velocity (perpendicular to impingement flow), m/s
w	nozzle width, m

Greeks

μ	viscosity, kg/m s
ρ	density, kg/m^3
λ	sub-equation in the Martin (1977) model

Subscripts

avg	average
D	dew point
E	over the edges
eff	effective
imp	impingement
j	at the jet exit
∞	bulk impinging fluid
i	at time step i, or component i
m	mass transfer
s	at product surface, or steam
T	total

Correlations to predict heat transfer coefficients under ASN flow (Table 1) were developed by Gardon & Akfirat (1966), Martin (1977), & Saad et al. (1980). Gardon & Akfirat (1966) studied heat transfer from an unconfined array of slot jets. In contrast, Martin (1977) studied mass transfer from an array of confined slot jets, in which spent fluid was forced to exit laterally between the jets. Saad et al. (1980) studied heat transfer from an ASN configuration using three slot nozzles, but used symmetrical exhaust ports alternating with the jet nozzles to eliminate the effect of spent fluid interference, as occurred in the Martin (1977) study.

If operating conditions also result in simultaneous vapor condensation during impingement processes, then the heat transfer rates can be significantly higher than in convection-only conditions. However, mass transfer in impingement flows has been studied only on a limited basis. Kroger & Krizek (1966) studied mass transfer rates by measuring the evaporation rate of naphthalene plates under impingement flows. As expected, the mass transfer rates were analogous to heat transfer rates, taking the form

$$Sh = KRe_j^{0.77}Sc^{1/3} \tag{1}$$

Table 1
Heat and mass transfer (Nu, Sh) correlations

Reference	Flow condition	Correlation and conditions
Martin (1977)	ASN[a]	$\frac{Nu}{Pr^{0.42}} = \frac{Sh}{Sc^{0.42}} = \frac{2}{3}\lambda^{3/4}\left(\frac{2Re_j}{f/\lambda+\lambda/f}\right)^{2/3}$; where $\lambda = (60 + 4(H/w - 2)^2)^{-1/2}$ $1500 < Re_j < 40{,}000$; $0.008 < f < 2.5\lambda(H/S)$; $2 < H/w < 80$
Saad et al. (1980)	ASN	$Nu = 0.14Re_j^{0.775}(H/w)^{-0.286}f^{0.314}$; where $0.0156 < f < 0.0833$, $4 < H/w < 24$, $3000 < Re_j < 30{,}000$
Gardon and Akfirat (1966)	ASN	$Nu = 0.66Re_j^{0.62}(H/w)^{-0.31}f^{0.38}$; where $0.0156 < f < 0.0625$; $H/w > 7$; $7000 < Re_j < 120{,}000$
Bejan (1995)	Turbulent parallel	$\frac{Nu_E}{Pr^{1/3}} = \frac{Sh_E}{Sc^{1/3}} = 0.037Re_E^{4/5}$

[a] ASN = array of slot nozzle impingement flow.