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To cite this article: Somayeh Davoodbadi Farahani & Farshad Kowsary (2017): Heat Transfer from Pulsating Laminar Impingement Slot Jet on a Flat Surface with Inlet Velocity: Sinusoidal and Square Wave, Heat Transfer Engineering, DOI: 10.1080/01457632.2017.1338868

To link to this article: https://doi.org/10.1080/01457632.2017.1338868
Heat Transfer from Pulsating Laminar Impingement Slot Jet on a Flat Surface with Inlet Velocity: Sinusoidal and Square Wave

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ABSTRACT

Heat transfer from a pulsating laminar impingement slot jet on a flat surface was investigated numerically and experimentally. Inlet velocity was considered sinusoidal velocity and square wave velocity. Experimental studies were done only for the sinusoidal velocity state. An inverse heat conduction method, conjugated gradient method with adjoint equation, was used for the experimental estimation of the local heat transfer coefficient along the target surface. Effect of the square wave velocity of the laminar impingement slot jet was studied numerically. The results show pulsations in flow change flow patterns and the thermal boundary layer thickness because of the newly forming thermal boundary layer is extremely small each time the flow is resumed. Heat transfer rate in this state enhances due to pulsating inlet velocity in comparison with steady state. Heat transfer increases with increasing pulsation amplitude. Enhancement in mean heat transfer on the target plate for sinusoidal velocity is rather than square wave velocity.

Introduction

Jet impingement is an effective technique which provides higher heat transfer rate in comparison with other methods through the formation of a thin thermal layer on the impingement surface [1–3]. Because of this characteristic, it is used in many industries such as cooling of gas turbine and electronic equipment, drying of paper and textiles.

For the most part, previous studies [4–6] have focused on optimizing the transport processes associated with steady impinging jets. Parameters that have received attention include the jet-impingement surface spacing, the impingement angle and turbulence intensity.

Several researchers investigated heat transfer rate of pulsating jets. Zumbrunnen and Aziz [7] investigated experimentally the effect of periodic on/off flow on convective heat transfer to a planar water jet impinging and enhancement in heat transfer was recorded. Sheriff and Zumbrunnen [8] investigated experimentally the effect of flow pulsation on cooling performance of an impinging water jet. Mladin and Zumbrunnen [9–11] studied how heat transfer in impinging jets is influenced by a pulsation. Results showed both increase and decrease in surface heat transfer compared to steady jet impingement, depending on the Strouhal number (St) of the pulsating jets. With St = 0.108, 0.022, and 006, surface heat transfer increased, decreased, and remained the same, respectively, compared to that from corresponding steady jet impingement. The deductions were that the enhancement was caused by large vortex structures, whereas the decrease was due to nonlinear dynamic effects within the boundary layers. Mladin and Zumbrunnen [12] investigated theoretically the influence of the pulse shape, frequency and amplitude on instantaneous and time-averaged convective heat transfer in a planar stagnation region using a detailed boundary layer model. However, Sailor et al. [13] used a Strouhal number between 0.009 and 0.042 and still recorded significant enhancement of heat transfer in stagnation line for pulsating flow. Fallen [14] has found no influence of pulsation on heat transfer in laminar flow. Hofmann et al. [15] experimentally investigated the flow structure and heat transfer from a single pulsating submerged round jet impinging perpendicularly on a flat plate. Their results show that convective heat transfer can be influenced by pulsations in the mean velocity and heat transfer can be enhanced by the pulsation when the pulsation frequency is in the order of magnitude of the turbulence. They reported that there exists a threshold Strouhal number, below which no significant heat transfer
enhancement was obtained. Hofmann et al. [16] studied numerically pulsating submerged jets. Lewkongsataporn et al. [17] reported heat transfer rate of impinging jets with square-shaped wave velocity increases in comparison with steady jets. Poh et al. [18] studied numerically heat transfer performance of a confined pulsed laminar impinging jet. Heat transfer of an externally perturbed impinging hot jet at Re = 1000 was numerically examined by Jiang et al. [19]. Hewakandamby [20] investigated numerically the effect of pulse velocity on heat transfer for two jets on a flat surface. Xu et al. [21] examined numerically the effect of large temperature difference between a sinusoidal impinging jet and a flat target surface. Demircan and Turkoglu [22] reported that the Nusselt number value oscillates with time in the wall jet region due to the formation of circulating flow in the pulsed jets on the target surface. Mohammadpour et al. [23] investigated numerically the effect of intermittent and sinusoidal pulsed flows on the heat transfer rate from a slot jet impinging to a concave surface. Mohammadpour et al. [24] studied numerically a composite design consisting of steady as well as intermittent (on/off) or sinusoidal jets in different combinations of four slot jets.

Few works on laminar pulsating impinging jet flows have been reported and the unsteady characteristics of impinging heat transfer are not yet fully understood. Thus, the effect of the periodic flow on heat transfer from the laminar impingement slot jet was investigated. Investigations were performed for sinusoidal velocity and square-wave velocity. An experimental study for sinusoidal velocity was done and heat transfer coefficient was estimated using the inverse heat conduction method. The estimation of the convective heat transfer coefficient is a nonlinear inverse problem. Thus, conjugated gradient method with adjoint equation was used. In this method, radiation heat transfer and lateral conduction in target plate was considered. For more study, the effect of the periodic flow on heat transfer from the laminar impingement slot jet was examined numerically. The effect of parameters such as pulse shape, frequency, amplitude pulsation, Reynolds number and nozzle-to-jet-to-surface spacing and nozzle width was defined as H and w, respectively. Boundary conditions on the computational domain are shown in Figure 1a. Velocity profiles at inlet jet were applied:

\[
\begin{align*}
\text{Square wave velocity:} & \quad \begin{cases} 
    u = u_{avg}(1 + A) & n = 0, 1, \ldots \\
    \rightarrow \frac{n}{f} < t < \frac{(n + 1/2)}{f} \\
    u = u_{avg}(1 - A) & \frac{(n + 1/2)/f} < t < \frac{(n + 1)/f}
\end{cases}
\end{align*}
\]

\[u = u_{avg}(1 + A \sin(2\pi ft)) \quad (2)\]

where A is the amplitude of pulsations in velocity. Square wave velocity and sinusoidal velocity are shown in Figure 1b. In the present study, the fluid is air as an incompressible and Newtonian fluid.

**Problem statement**

According to Figure 1a, the problem of the present study is a two-dimensional geometry semi-confined impinging slot jet. A heated surface \(q_w\) with no-slip condition was considered for the impingement surface. The confinement wall was specified to be adiabatic. The
Governing equations

The governing equations in the computational domain (Figure 1a) are expressed as follows:

\begin{align}
\text{continuity} : \quad \frac{\partial (u_i)}{\partial x_i} &= 0 \quad (3) \\
\text{Momentum} : \quad \rho \frac{\partial u_i}{\partial t} + \rho \frac{\partial (u_i u_j)}{\partial x_j} &= -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \rho g_i \quad (4) \\
\text{Energy} : \quad \rho c_p \left( \frac{\partial T}{\partial t} + \frac{\partial (u_i T)}{\partial x_i} \right) &= \frac{\partial}{\partial x_i} \left( k \frac{\partial T}{\partial x_i} \right) \quad (5)
\end{align}

Where $i,j$ in the above equations are 1,2. The continuity, momentum and energy conservation equations were solved numerically to simulate the flow and thermal field in the computational domain. The computational domain is symmetric and boundary conditions are described completely in Figure 1a. Along the upper and lower walls, the velocities satisfy the no-slip condition. Thermal boundary condition on the lower wall is $q = q_w$. Thermal boundary condition on the upper wall is $q = 0$. The boundary condition on the exit plane is an outflow. Eq. (1–2) show the velocity profile for inlet velocity. Pressure and temperature in the computational domain were $P_\infty$ and $T_\infty$. 

**Figure 2.** (a) Experimental set-up and (b) Schematic of pulsating valve mechanism.
Numerical solution

The finite volume method (computational fluid dynamics (CFD), Fluent Software) was used for solving the flow and thermal fields. A second upwind discretization scheme was used for advective terms and the SIMPLEC algorithm [25] was employed for pressure-velocity coupling. The y-direction grid was chosen to be nonuniform and the grid is refined near the target surface. In order to both guarantee the numerical accuracy and reduce the computational cost, the grid independence is studied by employing different fine and coarse meshes and the optimum grid size is determined. The grid refinement is continued until halving the grid size resulted in less than 1% change in the computed local Nusselt number (Nu). A grid density of $150 \times 50$ (nonuniform grid) provides satisfactory solution. Converged solution was deemed to have been achieved and iterations terminated when all the residuals were less than 0.0001. The relaxation factors were employed to promote the smooth convergence of the discretized equations. The relaxation factors were 0.7, 0.7, 0.3 and 1 for $u$, $v$, $P$ and $T$, respectively.

The time step is also varied in accordance with the frequency of pulsating jets. Convergence criteria were set for all the residuals being less than 0.0001. The time-step independence step for the unsteady state was checked. Small time steps were not required in the implicit procedure. Convergence could be guaranteed when time step size is set at $1/(\text{pulsation frequency})/10$.

This numerical method has been validated by comparison with experimental results for sinusoidal velocity ($H/D_h = 4$ at $Re = 900$, $A = 0.5$ and $f = 0, 80$ and 300 Hz) in Figures (3b, 4a and 4b), presented in the results section.

The hydraulic diameter of the nozzle is:

$$D_h = \frac{2wl}{w + l} \quad (6)$$

where $w$ is the slot width and $l$ is the slot length.

Reynolds number for impingement jet is:

$$Re = \frac{u_{avg}D_h}{v} \quad (7)$$

where $v$ is the inlet velocity and $\nu$ is the fluid dynamic viscosity. Nusselt number is given by:

$$Nu(x, t) = \frac{h(x, t)D_h}{k}$$

$$= \left( - \frac{\partial T}{\partial y} \bigg|_{y=0} \right) D_h \left( T_s(x, t) - T_{jet} \right) \quad (8)$$
In the pulsating flow case, the Nusselt number depends on time and position, therefore, time-average Nusselt number for each point on the target surface is given by:

$$\overline{Nu}(x) = \int_{0}^{t_f} \frac{1}{t_f} Nu(x, t) \, dt$$  \hspace{1cm} (9)

where $t_f$ is final time and overall Nusselt number along target plate can be calculated, as follows:

$$\overline{Nu} = \int_{0}^{L} \left( \int_{0}^{t_f} \frac{1}{t_f} Nu(x, t) \, dt \right) \, dx$$  \hspace{1cm} (10)

where $L$ is the length of target plate, The frequency is expressed by the Strouhal number ($St = f/d$), where $f$ is the pulsation frequency and $u_{avg}$ is the averaged velocity.

### Experimental apparatus

A schematic of the experimental apparatus is shown in Figure 2a. Compressed air flows through a series of two air filters, an air filter/regulator and then a flow meter (Rotameter) into a plenum chamber. A pressure gauge, connected before the plenum chamber, is used to correct the flow rate. Baffle plate in the inlet of the plenum chamber allows the flow to decelerate and expand. A row of stainless steel mesh screens and honeycomb, which is used to produce uniform velocity profile at the nozzle outlets, are mounted in the bottom half of the plenum chamber in order to dissipate large and small eddies, respectively. A programmable 3-D stage controller, used to adjust the position of the hot-wire sensor, was positioned at the right measuring place within 15 μm spatial resolution. The most deviations of the mean velocity between hot wire sensor and Rotameter was 0.4% for all cases.

Top of Plexiglas plenum's surface is screwed to the other Plexiglas plates which contact zones are sealed with rubber gaskets and silicon glue to prevent any probable air leakage. The jet nozzle is placed at the bottom of the plenum chamber. The length of two-dimensional slot jet is considered longer than the target plate to avoid the entrance of three-dimensional vertical structures of the slot corner into the heated zone. The dimension of nozzle is $76 \times 7 \text{ mm}^2$. A stainless steel (AISI-304) plate $250 \times 70 \times 5 \text{ mm}^3$ was used as a target surface. A silicon heater of less than $3 \text{ mm}$ thickness, placed at the bottom of the plate provides a uniform heat flux of $2000 \text{ W/m}^2$.

The total power supplied was monitored using two digital multi-meters; one was used for voltage and the other for the current. The emissivity of the plate surface, which is polished by grinding process to minimize the radiation heat transfer, measures 0.75 via an infrared thermometer gun. Wooden end caps with thermal conductivity of $0.12 \text{ W/m.K}$ were attached to small sides of the plate to minimize the end effects.

The surface temperatures in target plate were measured over a distance of seven times the slot width from the stagnation line. 11 type-K thermocouples (TP-01) were used, one at the stagnation line, 9 on the left hand side and 1 on the right side of the symmetry line to check the symmetry of the heat transfer distribution. Distance between each two adjacent thermocouples is 5.25 mm. Three layers of materials were used to insulate the backside of the target plate and the heater assembly. The first layer was a wood insulation of $16 \text{ mm}$ thickness, (thermal conductivity $0.12 \text{ W/(m.K)}$), the second layer was a fiber glass of $12 \text{ mm}$ thickness, (thermal conductivity $0.035 \text{ W/(m.K)}$) and the third layer was an elastomer of $6 \text{ mm}$ thickness (thermal conductivity $0.036 \text{ W/(m.K)}$). Thermocouples were inserted through holes of $4.5 \text{ mm}$.
Figure 5. Velocity Stream function contours of pulsating jets with sinusoidal inlet velocity (Numerical solution) for \( Re = 900, H/D_h = 4 \) and \( f = 80 \) Hz (St = 0.9) at time (a) \( t = 0.05 t_s \), (b) \( t = 0.5 t_s \), and (c) \( t = t_s \).

depth machined through the thickness of the metal plate. Also, the wood-heater surface temperatures in target plate were measured using 11 type-K thermocouples, with the same arrangement on the metal surface. Heat flux distribution of the heater was estimated using an inverse method and the wood-heater surface temperatures. The pulse air jet is generated using a pulsating valve (Figure 2b). This valve generates a pulse similar to sinusoidal shape, thus, inlet velocity to nozzle is sinusoidal form. The velocity of the jet exit air close to the nozzle was measured with a calibrated hot wire anemometer. Figure 3a shows the generated velocity profile by the pulsating valve, for pulsation frequency, \( f = 30 \) Hz. The varying velocity with time is similar to a sinusoidal velocity.

The convective heat transfer coefficients have been estimated by using the real measured temperatures in plate and inverse method. In this experimental study, the inverse method was conjugate gradient method with adjoint equation. The physical model for target plate (Figure 3b) is defined as:

\[
\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \\
\frac{\partial T}{\partial x} \bigg|_{x=0} = \frac{\partial T}{\partial x} \bigg|_{x=L} = 0 \\
-k \frac{\partial T}{\partial y} \bigg|_{y=0} = q_w \\
-k \frac{\partial T}{\partial y} \bigg|_{y=E} = h(T - T_{jet}) + \varepsilon \sigma (T^4 - T_{jet}^4) \quad \text{and} \quad T \bigg|_{t=0} = T_0
\]

where \( q_w \) and \( h \) are the heat flux of the heater and unknown convective heat transfer coefficient, respectively. Temperature histories in target plate were
recorded. In our experimental study, the inverse method used was the conjugate gradient method along with the adjoint equation [26–28]. This is due to the fact that estimation of the convective heat transfer coefficient is a nonlinear function estimation problem. For more detail about mentioned method see reference [26]. Experimental investigations have been done at three frequencies: 0, 80, and 300 Hz.

The uncertainties are related to measuring devices (Table 1) such as thermocouples, Rotameter, machining process, laser cutting machine, and infrared thermometer gun. Using the method proposed in [29], the maximum range of relative overall uncertainty in estimated Nusselt number in this experiment is 4.7 ±2.2%.

Result and discussion

The purpose of this study was to investigate the effect of the periodic flow on heat transfer from the laminar impingement slot jet. First, the numerical solution scheme used here for sinusoidal velocity was validated by

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<th>Table 1. The error sources.</th>
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<td>Error source</td>
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<td>Thermocouple</td>
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<tr>
<td>Infrared thermometer gun</td>
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<tr>
<td>Rotameter</td>
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<td>Machining process</td>
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<td>Laser cutting machine</td>
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Figure 7. Temperature contours of pulsating jets with sinusoidal inlet velocity (Numerical solution) for $Re = 900$, $H/D_h = 4$ and $f = 80$ Hz ($St = 0.9$) at time (a) $t = 0.05\, t_r$, (b) $t = 0.5\, t_r$, and (c) $t = t_r$.

comparison with the experimental Nusselt number distribution for $H/D_h = 4$ at $Re = 900$, $A = 0.5$ and $f = 0, 80$ and $300$ Hz as shown in Figures (3c, 4a and 4b). It is noted that the flow for $Re = 900$ is laminar [30]. The prediction of time-averaged local Nusselt number in the present study (numerical solution) displays acceptable level of agreement with the experimental data. It is observed that the heat transfer coefficient increases with increasing frequency and there doesn’t exist a threshold frequency for laminar flow, but Hofmann et al. [15] show for turbulent flow that there exists a threshold Strouhal number, below which no significant heat transfer enhancement was obtained. Velocity stream function contours from numerical solution at several different times for $Re = 900$ at a frequency of $80$ Hz ($St = 0.9$) in Figures 5 (a, b, and c) is shown. The motion can be represented with streamlines showing the direction of the flows in different regions. Pulsations will cause disturbance in the jet boundary layer. The wave propagation effect because of pulsations inlet velocity was observed in velocity stream function contours and the streamlines change with time. The pulsation in inlet velocity causes to increase the streamlines density in $x < 0.05\, m$. The density of the streamlines increases as the velocity increases. Thus, a significant enhancement in heat transfer in this region is observed. Time-averaged velocity profile at some distance from the stagnation line, respectively, in Figures 6 (a–e) is shown for $Re = 900$ and $H/D_h = 4$. Figure 6 shows
that the velocity profile for $f = 300$ Hz is broader than the steady state. Wave propagation velocity is dependent of fluid property, jet geometry and flow regime. Temperature contour at several different times for $Re = 900$ at a frequency of 80 Hz ($St = 0.9$) in Figures 7 (a, b, and c) is shown. Figure 7 shows how pulsations propagate in the wall jet zone and change thermal boundary layer shape. In fact, sinusoidal flow pulsations can induce non-sinusoidal responses because of nonlinearities in momentum and energy transfer within the boundary layer. The nonlinear dynamic effects reflect that disturbances associated with flow pulsations do not allow momentum and energy transfer to equilibrate within the boundary layer. Both thermal and velocity boundary layer is not fully formed due to pulsations in flow. The thermal boundary layer is disrupted and renewed periodically. A stagnation point similar to that of steady jet are formed where the wall jet interact. The location of the stagnation point is not fixed and oscillates around a fixed position due to the flow field fluctuations above the wall jet. The wall jet oscillation reduces boundary layer thickness; when the boundary layer begins to establish and grow; the inlet velocity and flow condition are changed and prevents to form a constant boundary layer. Therefore, heat transfer is increased by pulsation due to the minimized time-averaged boundary layer thickness. The flow pattern and temperature distribution in the computational domain are changing with time (See Figures 5 and 7).

Effect of square form of inlet velocity was investigated. Velocity stream function contours and temperature contours at several different times for $Re = 900$ at a frequency of 80 Hz ($St = 0.9$) are shown in Figures 8 and 9.
Figure 9. Temperature contours of pulsating jets with square wave shape inlet velocity (Numerical solution) for $Re = 900$, $H/D_h = 4$ and $f = 80$ Hz ($St = 0.9$) at time (a) $t = 0.05 t_s$, (b) $t = 0.5 t_s$, and (c) $t = t_s$.

respectively. These figures show the effects of the pulsation in inlet velocity and show how pulsations propagate in the domain with time. Figure 8 shows how pulsations propagate in the flow field and how the streamlines density in the flow filed along the target plate changes. This figure shows that the position of the maximum changes in the streamlines density moves along the target plate with time. Thus, the enhancement in the flow velocity is seen along the target plate with time, not only in the stagnation region. Thus, significant enhancement in heat transfer in the wall jet region is observed.

Also, the configuration geometry has an important effect on the heat and mass transfer performances in pulsed impingement jet. Heat transfer enhancement can be described to reflect flow by the confinement plate. That is the fluid can reach the confinement wall after impinging with the impingement wall and possibility induce higher mixing, which can contribute to enhanced heat transfer rate.

Figure 10a shows that local heat transfer coefficient of periodically flow; sinusoidal velocity and square wave velocity. The significant enhancement in heat transfer for square wave velocity and sinusoidal velocity is observed in stagnation region and in the wall jet region, respectively. The impact of pulsations amplitude on the overall Nusselt number (dimensionless heat transfer coefficient) is shown in Figure 10b. When the pulsation amplitude increases, mean velocity (in x-direction) of the inner jet is decreased due to mixing between the jet and the environment is enhanced. Thus, $(T_{jet} - T_{target plate})$ is reduced and heat transfer increases. Mean of time-average Nusselt number on plate increases with increasing pulsations.
amplitude. This enhancement for sinusoidal velocity is rather than square wave velocity. Figure 11 shows the changes of overall Nusselt number (dimensionless heat transfer coefficient) for various Re at $f = 10$ Hz ($St = 0.11$) and $H/D_h = 2$. Enhancement in heat transfer with periodically velocity is seen for various Re. Effect of $H/D_h$ and frequency on heat transfer of a pulsating jet for Re = 900 is shown in Figures (11b and 11c). Heat transfer rate decreases with increasing $H/D_h$ due to the area between jet and environment and therefore the effect of entrainment increased. Figure 11b shows the heat transfer for square wave velocity increases with increasing frequency in each $H/D_h$. Figure 11c shows that maximum enhancement in heat transfer for sinusoidal velocity is at $f = 10$ Hz ($St = 0.11$). It became clear, enhancement in mixing between the jet and the environment is decreased with increasing the pulsation frequency for sinusoidal velocity. Sinusoidal flow pulsations can induce non-sinusoidal responses because of nonlinearities in momentum and energy transfer within the boundary layer.

Figure 11. (a) effect of frequency on $\bar{N}u$ at various Re with square wave shape and sinusoidal wave shape inlet velocity (Numerical solution), (b) effect of $H/D_h$ on $\bar{N}u$ at various frequencys at Re = 900 with square wave shape (Numerical solution) and (c) sinusoidal wave shape (Numerical solution).

Conclusions

A 2-D laminar impingement slot jet with the periodically inlet flow has been studied. Inlet velocity was considered sinusoidal velocity and square wave velocity. For the sinusoidal velocity, an experiment was done. In the
experiment, the inverse method was used as a measurement technique to estimate the convective heat transfer coefficient. In this method, radiation heat transfer and lateral heat conduction in metal plate were considered. Convective heat transfer can be influenced by periodic fluctuations in the mean flow. Pulsations in inlet velocity cause to increase mixing between the jet and the environment. The thermal boundary layer thickness due to the newly formed thermal boundary layer is extremely small each time the flow is resumed. Heat transfer in laminar jet increased when inlet flow was periodical. Also, enhancement in heat transfer of plate for sinusoidal velocity is higher than square wave velocity.

Notes on contributors

Somayeh Davoodabadi Farahani received her PhD in Mechanical Engineering in 2015 from Tehran University, Tehran, Iran. Her research areas of interest include: experimental investigations in heat transfer, the use of numerical and analytical methods of solution of the heat transfer problem, direct simulation of thermal systems, solution of inverse heat transfer problems, and optimization of thermal systems.

Farshad Kowsary is a professor in the field of heat transfer at the University of Tehran, Iran. His research interests are in the area of inverse heat transfer with a focus on inverse radiation and conduction. He has a sizable number of papers in the above subjects in reputable heat transfer journals. He is a major reviewer for the Journal of Quantitative Spectroscopy and Radiative Transfer as well as Heat and Mass Transfer.

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