Heat Transfer under Multiple Slot Jets Impinging on a Permeable Moving Surface

Local and average heat transfer were measured for a system of multiple jets impinging on a moving permeable surface at which there may be throughflow. Multiple jets were confined by a hood, as is required industrially for thermal efficiency. Exhaust ports located symmetrically between the jet nozzles eliminated crossflow, otherwise a strongly detrimental effect. Impingement surface motion decreases average heat transfer, by 17% at industrially relevant values of the surface motion parameter, M_{vs}. Enhancement of impingement heat flux by throughflow is linearly additive. Expressed as $\Delta\overline{St}$, this enhancement depends only on the throughflow parameter, M_{us}, with $\Delta\overline{St}/M_{us} = 0.17$. For typical operating conditions, withdrawal as throughflow of only 10% of the jet flow increases mean Nusselt number by over 50%. Industrial design modeling for the potential process of combined impingement and throughflow drying of wet webs such as paper is demonstrated.

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Introduction

This investigation concerns a transport phenomena problem with boundary conditions that have previously precluded experimental investigation, that is, impingement heat transfer to a rapidly moving surface with throughflow at the convective surface. The study derives from the industrial importance of systems of multiple impinging jets for achieving high rates of heat transfer at rapidly moving surfaces. In practice, moving impingement heat transfer surfaces may also be permeable, as in the drying of textiles and paper. This provides the possibility of inducing throughflow at the surface in order to enhance heat transfer rate. Industrial impingement heat transfer systems involve multiple jets, and these generally cannot be unconfined. Thermal efficiency requirements normally dictate use of a confinement hood to retain the impingement flow, permitting recirculation of the spent flow for heat recovery. The present study is the first to provide convective heat transfer rates for a confined system of multiple slot jets impinging on a permeable moving surface at which there can be throughflow.

In the use of multiple jets one basic design feature is location of the ducting to exhaust spent flow from the impingement system. In the arrangement chosen here the spent flow exhausts at the confinement hood through slot ports located symmetrically between the slot jets at a jet centerline-to-exhaust centerline spacing, S. This design eliminates the strongly detrimental effect of crossflow on impingement heat transfer documented by Saad (1981) and Ahmad (1987). Such a multiple slot jet system consists then of repeating flow cell units, each of width S and depth H, the confinement surface-to-impingement surface spacing.

As three variables S, H and the jet nozzle width, w, define the geometry of this confined jet system, two independent nondimensional ratios define geometrically similar systems. One universally used ratio is the nozzle-to-impingement surface spacing, H/w. In his investigation of confined impingement systems with a stationary heat transfer surface and with slot jets alternating with slot exhausts, Saad (1981) demonstrated that the appropriate choice for the second nondimensional ratio is S/H, the aspect ratio of the repeating flow cell units. He found that for $S/H > 1.5$, multiple slot jets become equivalent to an assembly of noninteracting single jets, while for aspect ratios below a critical value of $S/H = 0.7$, the adjacent impinging jet and exhaust flows are sufficiently close that the flow and heat transfer characteristics are affected even at the stagnation point, i.e., over the entire impingement surface.

Based on data for a wide range of Re, H/w and S/H, Saad's correlation indicated that the maximum value for average Nusselt number, would be obtained for a multiple jet system with geometrical parameters $H/w = 5$, $S/H = 0.5$. In view of the critical value of the aspect ratio, $S/H = 0.7$, the predicted

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condition for maximum $\overline{Nu}$, i.e., $S/H = 0.5$, evidently corresponds to a system with entering jets and leaving spent flows interacting strongly.

The present study examines the effect of throughflow at a surface moving from negligible to high speed under confined impinging slots jets. The system geometrical parameters H/w and S/H chosen were those predicted by Saad to give maximum impingement heat transfer rate. The range of other variables was selected so that the results would be relevant for an important industrial application, the drying of paper. Local heat transfer rate was measured with a unique heat flux sensor, Polat et al. (1991), for a moving surface at which there may be throughflow. The principal fixed dimensions were $H = 50$ mm, $S = 25$ mm, $w = 10$ mm, giving $H/w = 5$ and $S/H = 0.5$. The range of other parameters was $10 < (T_s - T_j) < 24°C$, $8,000 < Re < 26,000$, $14 < u_j < 48$ m/s, $0 < u_s < 0.4$ m/s, and $0.5 < v_s < 8$ m/s.

Experimental Facility

General description

The experimental facility, Figure 1, was designed for the discharge of three closely-spaced slot jets (1) on a moving porous impingement surface (3). The impingement flow was retained under a confinement hood (4) concentric with the impingement cylinder and flush with the nozzle exits. In order to provide steady-state operation of the overall system a heating jet (10) was directed on the rotating impingement cylinder at 180° from the multiple cooling jets. Thus the impingement cylinder assumed a steady temperature, intermediate between that of the heating and cooling jets. With this temperature, 46–60°C for the conditions used, the ΔT for the multiple jets was in the range 10–24°C. Spent flows from the heating and cooling jets were kept separate by skimmer plates (6).

Multiple jets

The multiple jet system (1) consists of three slot jets, each 10 mm wide × 0.2 m long in the axial direction, with a nozzle-to-nozzle centerline spacing $2S = 50$ mm. Flow to each jet is supplied via a nearly parallel channel (2), 90 mm wide at its inlet and 0.4 m long, terminating flush with the confinement surface. The jets impinge on the porous surface spaced at $H = 50$ mm from the nozzle exit.

The fraction of nozzle exit flow not withdrawn as throughflow is exhausted through ports at the confinement surface. For the symmetrical multiple exhaust arrangement used, each of the two interior ports is 20 mm wide while the two side exhaust ports are each 10 mm wide. For $2S = 50$ mm there is therefore a 10 mm confinement surface between each nozzle and exhaust port. As proven by Saad (1981) for a stationary impingement surface with this symmetrical exhaust flow configuration, the middle jet of such an array of three identical jets is representative of a slot jet in a multijet array.

Air supply to the multiple jets: Air at about 40°C (due to blower heating) enters the multiple jet section via a 0.25 m diameter pipe (not shown). After a 4.75 m flow measuring section, a 1.45 m long diverging section adapts this pipe to a plenum box (5) 0.4 m × 0.2 m in cross-section, 0.3 m deep in the flow direction. Two 0.19 m high movable flow dividers serve to provide equal flow to each jet. A 100-mesh screen at the plenum box entrance reduces the turbulence level and flow is straightened by a 50 mm thick honeycomb (4.7 mm cells).

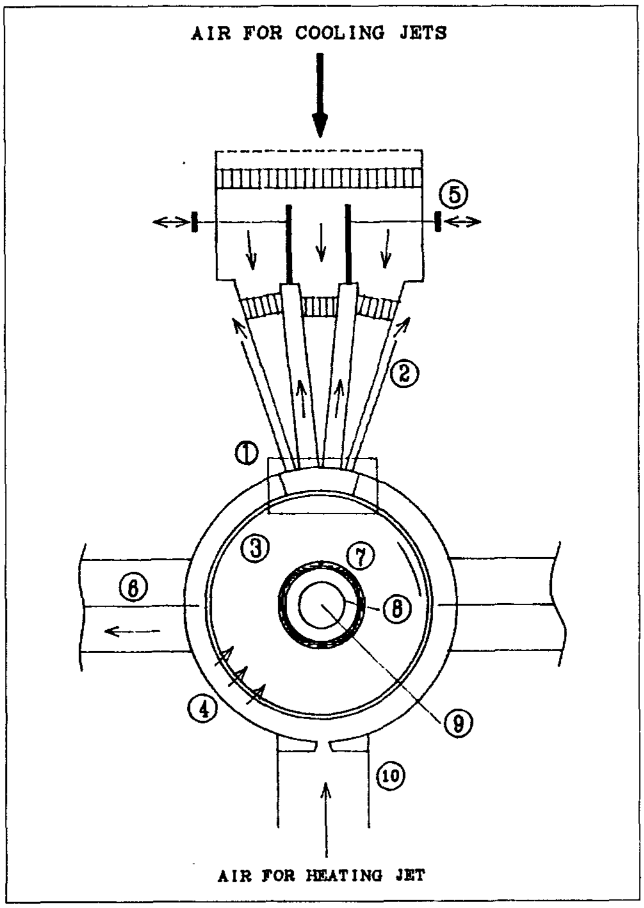

Figure 1a. Experimental facility.

1. Multiple jet system
2. Nozzle and exhaust channels
3. Impingement surface
4. Confinement hood
5. Plenum chamber
6. Skimmer plates
7. Porous cylinder
8. Hollow shaft
9. Suction
10. Heating jet

Heat transfer surface

The impingement surface is the external wall of a rotating porous cylinder, 0.48 m diameter, 0.2 m long. This cylinder, of 13 mm wall thickness, was custom fabricated from 50 μm uniform size particles, grade 55 "porous glass" of the 3M Company, with porosity 30%. The surface of the cylinder was very smooth, 1–2 μm equivalent sand roughness.

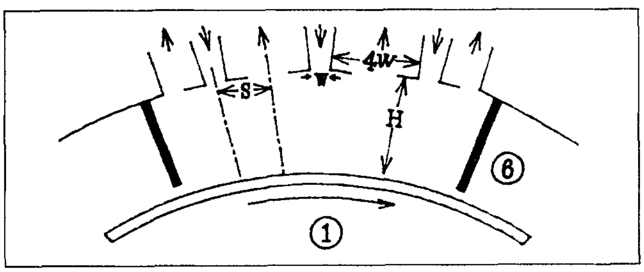

Figure 1b. Multiple jet system.

Suction system

As suction is applied for experiments with throughflow at the heat transfer surface, the impingement cylinder is mounted on a hollow shaft, a 0.114 m O.D. and 0.37 m long perforated pipe (8). To obtain radially uniform throughflow a 0.165 m diameter cylinder (7), of porous glass of higher porosity (3M Company, grade 155) than the impingement cylinder, is located as shown in Figure 1. The suction line connection is by a 102 mm I.D. flexible stainless steel line (not shown) via a Rulon sealed bearing. Throughflow velocity at the porous cylinder surface is calculated from throughflow rate. Local uniformity was confirmed by measurements with various fractions of the surface blocked. A pressure drop across the cylinder wall of 3.5 kPA gives a throughflow velocity of about 0.4 m/s, the maximum used.

Heat flux measurements

The unique permeable heat flux sensor used was a thin-film resistance thermometer, a 70 mm long × 1 mm wide gold filament deposited about 0.15 μm thick on a substrate, 100 mm long × 10 mm wide, identical to the porous glass wall of the heat transfer cylinder. For measurement of substrate temperature during in-situ calibration, two thermocouples were glued flush with the substrate surface, 5 mm from the gold film ends. This porous sensor, flush mounted centrally in the 0.2 m long porous wall of the 0.48 diameter heat transfer cylinder, Figure 1, monitored the instantaneous local surface temperature.

Leads from the heat flux sensor and thermocouples pass through a low noise slip ring assembly (IEC Corp., Model IEC-TX-14) to the signal conditioning unit consisting of a Wheatstone bridge (J.C. Biddle Co., Cat. No. 601022), a low noise differential amplifier (DANA Model 2820), and a tuneable low pass filter (Frequency Devices, Inc., Model 901F1). About 500 samples per rotation were acquired via an IBM-PC based data acquisition system containing the 27.5 kHz A/D board (Data Translation, DT2801-A). Local surface temperature was obtained from the sensor resistance using the sensor calibration. With a resistance of 186 ohms, the sensor sensitivity was 459 μV/°C, and its response time was in the order of 10^{-10} s. With the electronic instrumentation and data acquisition system used, a surface temperature resolution of 0.0005°C was obtained.

Van Heiningen et al. (1985) describes the principles by which an impermeable thin-film sensor works. The unique feature of the present sensor is that its substrate is permeable. From the surface temperature measurements, therefore, instantaneous values of local Nusselt number at the permeable surface, with and without throughflow, could be obtained via iterative solution of the equation:

$$\rho_P C_{Pp} \frac{\partial T}{\partial t} = k_{\text{eff}} \frac{\partial^2 T}{\partial x^2} - \rho_a C_{Pa} u_s \frac{\partial T}{\partial x} \tag{1}$$

and the relations

$$q_s = k_{\text{eff}} \frac{\partial T}{\partial x}; \quad h = \frac{q_s}{(T_j - T_s)}; \quad Nu = \frac{hw}{k_j}.$$

The alternation between heating and cooling at the exterior surface of the impingement cylinder was sufficiently rapid that the temperature waves never reach its interior surface. This provides the second boundary condition necessary for the solution of Eq. 1. Considering all contributions to experimental error, uncertainty in local Nusselt number was estimated to be ±5%.

In derivation of Eq. 1 it is assumed that within the substrate the local average substrate temperature equals the local air temperature. The applicability of this assumption was checked by calculating Nu distributions for a particular case twice, once from the above equation, once from simultaneous solution of the heat transfer equations for air and substrate with an estimated convective heat transfer coefficient in the coupling term. The two Nu distributions were indistinguishable.

The permeable sensor results were validated by the following tests:

1. The $\overline{Nu}$ results were ΔT-independent in the range $10 < \Delta T < 27$°C.
2. Local *Nu* profiles under a single impinging jet obtained with this sensor under no throughflow and negligible surface motion conditions agreed with the results of van Heiningen (1985) and Cadek (1986), who used an entirely different type sensor.
3. The extent of shift in position of the off-stagnation maxima of local impingement *Nu* profile at a rapidly moving surface measured with this sensor agreed with that measured by van Heiningen et al.

It is the development of this sensor, detailed by Polat et al. (1991), which has opened to experimental study the class of local transient heat transfer problem with throughflow at a moving convection surface.

Heat Transfer without Throughflow or Impingement Surface Motion Effects

Measurements

The profiles of local *Nu*, Figure 2, correspond to three jets located at positions, y, of $-5w$, 0 and $5w$ ($-1H$, 0 and $1H$) from the middle jet centerline, with exhaust ports symmetrically at $-7.5w$, $-2.5w$, $2.5w$ and $7.5w$ ($-1.5H$, $-0.5H$, $0.5H$ and $1.5H$). The off-stagnation minima and maxima seen in local profiles of single jets at this *Re* with $H/w = 5$ do not occur for jets as closely spaced as with $S = 2.5w = 0.5H$. For a stationary

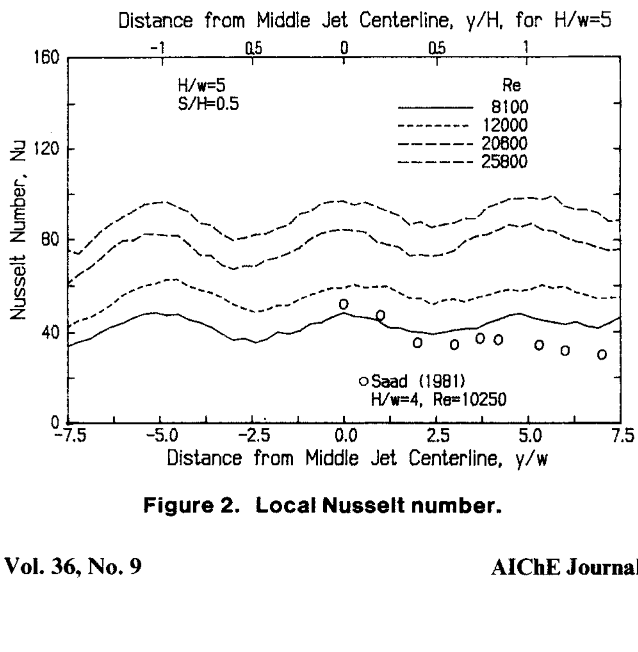

Figure 2. Local Nusselt number.

impingement surface Saad (1981) showed that in flow cells of aspect ratio as narrow as $S/H = 0.5$, interaction between jets and adjacent exhaust flows depresses jet centerline Nu by $\backsim 10\%$ and enhance exhaust centerline Nu by $\backsim 25\%$ relative to Nu for a single jet. Figure 2 demonstrates these two effects by reference to the Nu profile of Saad for a single slot jet at $H/w = 4$, very close to that used here, $H/w = 5$. Because of these two effects the Figure 2 profiles are more uniform than for the equivalent single jet. Thus a system of such closely spaced jets yield values of Nu_0 and Nu not different by more than 10%.

$\overline{Nu}$, the main interest in design, is obtained by integrating Nu for the middle jet over the heat transfer surface of width $2S$, i.e., from $-2.5w$ to $2.5w$ ($-0.5H$ to $0.5H$).

Analysis and comparison

In Figure 3 the results for $\overline{Nu}$ at $H/w = 5, S/H = 0.5$ as a function of Re are displayed with the results of four earlier correlations. Saad's correlation is for $H/w \geq 8$ but the Figure 3 line corresponds to his recommendation that at $H/w = 5$, $\overline{Nu}$ is 5% greater than at $H/w = 8$. Gardon and Akfirat's correlation is limited to $H/w \geq 8$ and to a percent open area of nozzles $f \leq 6.25\%$. The values here $H/w = 5, S/H = 0.5$ or $f = 20\%$ are outside these limits. As $\overline{Nu}$ is not much affected by H/w over the range $4 < H/w < 8$, one may approximate $\overline{Nu}$ at $H/w = 5$, $S/H = 0.5$ by $\overline{Nu}$ at $H/w = 8, S/H = 0.5$ ($f = 12.5\%$) which are the conditions for their Figure 3 line.

Saad, Martin and Schlünder and the present study used confined jets while others used unconfined jets. Martin and Schlünder exhausted the spent air along the edge which is in line with the two ends of the slot jets, thereby introducing a deleterious crossflow effect. Thus as would be expected, their results give anomolously low heat transfer rates, Figure 3. Their type of crossflow exhaust arrangement is unacceptable for those industrial drying applications where uniformity of the dried product is essential, for example in paper drying. The other studies exhausted the spent air between the jets, without crossflow, normal to the impingement surface.

Heat transfer data from unconfined jets generally provide an unreliable basis for designing the confined impingement systems normally used in industry for thermal efficiency. Entrainment of large amounts of the surrounding air makes unconfined jet heat transfer subject to several equipment specific effects, i.e., the relationship between three temperatures—nozzle exit, impingement surface, ambient—as well as equipment dimensions near the nozzle exit that influence the amount of ambient air entrained. The closer the jets in a multiple jet system, the less is the deviation of unconfined jet results from the performance characteristics of a confined jet system. With a flow cell aspect ratio, S/H, as small as in the present case, the unconfined jet results, Figure 3, are within 10% of those with a confinement hood. As larger internozzle spacings are common industrially, this confined/unconfined difference is typically much larger than 10%.

Correlation

The present results at $H/w = 5, S/H = 0.5$ without surface motion effects correlate as

$$\overline{Nu} = 0.094\, Re^{0.68} \tag{2}$$

for $8{,}100 < Re < 25{,}800$. This equation is the limiting form for $M_{vs} = 0$ of Eq. 5, which includes the surface motion effect. The Re exponent of all previous studies ranges from 0.6 (Schuh and Pettersson) to 0.8 (Saad).

From his correlations Saad predicted that $\overline{Nu}$ would be a maximum at $H/w = 5$ and $S/H = 0.5$, but made no measurements with this H/w–S/H. For narrow flow cells, S/H of 0.375–0.75, he used larger spacings, H/w of 8–24. For $H/w = 4$, his narrowest aspect ratio was $S/H = 1.5$, and multiple jets spaced this far apart act as single jets. For $Re = 10{,}000$, Figure 4 shows that Saad's prediction that the maximum in $\overline{Nu}$ occurs around $H/w = 5$ and $S/H = 0.5$ appears substantiated by the present measurements. The $H/w = 5$ line presumably parallels the lines on Figure 4 for other values of the H/w spacing.

Heat Transfer with Throughflow

Measurements

For single slot jets the effect of throughflow on Nu and $\overline{Nu}$ was studied by Saad (1981) for a stationary heat transfer surface,

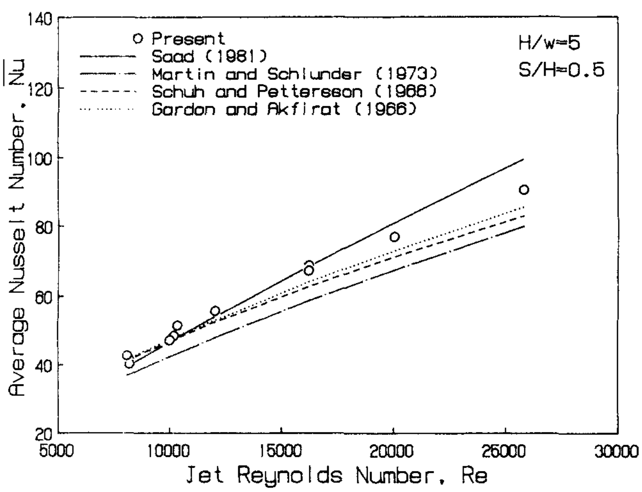

Figure 3. Effect of Reynolds number on average Nusselt number.

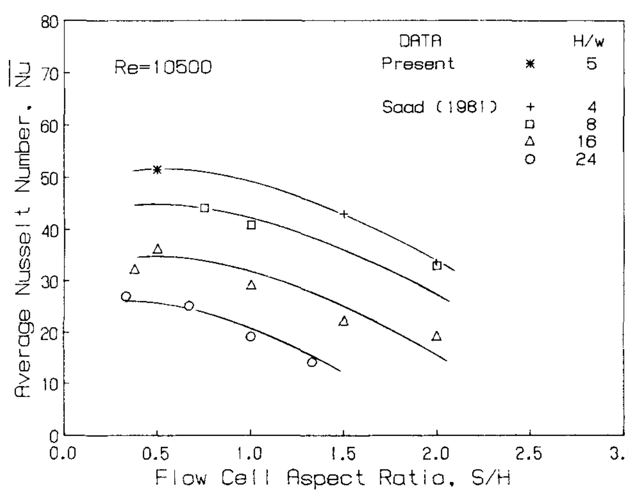

Figure 4. Effect of flow cell aspect ratio on average Nusselt number.

and by Polat et al. (1990a,b) without and with the effects of impingement surface motion. Saad varied throughflow velocity, u_s, from 0 to 0.3 m/s with Re of 10,200–29,100 for $S = 2.25H$ ($18w$). Polat et al. varied u_s from 0 to 0.4 m/s with *Re* of 16,400–57,700 for $S = 6.4H$ ($16w$). For the extention here to multiple jets the throughflow parameter, M_{us}, was varied over the range 0 to 0.0235 for Re of 7,900–25,800. These nondimensional limits correspond to u_s of 0–0.35 m/s and u_j of 15.6–47.7 m/s. At the maximum value of the throughflow parameter, $M_{us} = 0.0235$, only about 12% of the jet flow is removed through the heat transfer surface.

For each value of M_{us}, every profile of *Nu* under multiple jets, when compared to profiles determined with no throughflow, was found to increase by a uniform amount over the entire impingement surface. The linearly additive characteristic of throughflow on local convective heat transfer for multiple jets is consistent with that found for single jets as reported by Saad and by Polat et al. Profiles of Nu at various values of M_{us} are not shown because, visually, they are indistinguishable from profiles with no throughflow, Figure 2.

Correlation and Comparison

The enhancement of impingement heat transfer by throughflow, expressed on Figure 5 in terms of Stanton number, is a linear function of the throughflow parameter, M_{us},

$$(\overline{St})_{\text{with throughflow}} - (\overline{St})_{\text{without throughflow}} = \Delta\overline{St} = 0.16\ M_{us} \quad (3)$$

The linearly additive nature of enhancement by throughflow, seen first in the profiles of local heat transfer, is reflected above for enhancement of average heat transfer. Moreover, the linear relationship between $\Delta\overline{St}$ and M_{us} is independent of *Re* for this multiple confined slot jet system at $H/w = 5$, $S/H = 0.5$, just as found by Polat et al. (1990a) for a single slot jet. Comparison of Figure 5 and Eq. 3 with results for a single confined jet at $H/w = 2.5$ and $S/H \leq 6.4$ ($S/w \leq 16$) shows that essentially the same proportionality factor, $\Delta\overline{St}/M_{us} = 0.16$, applies for the single and multiple jet cases.

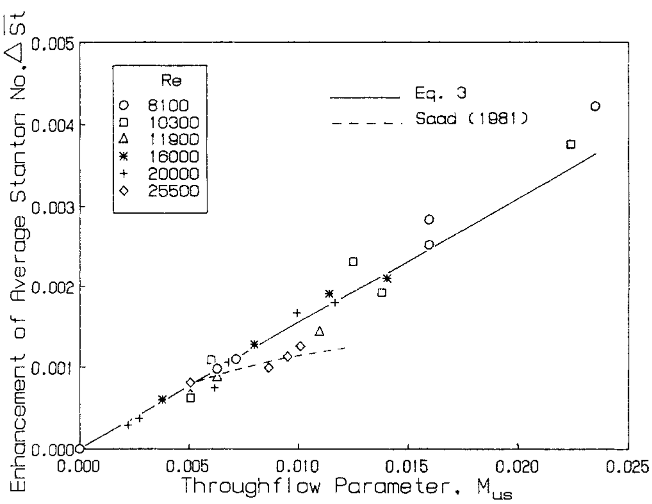

Figure 5. Effect of throughflow on enhancement of average Stanton number without surface motion effects.

Figure 5 shows that the single jet results of Saad agree with Eq. 3 at low M_{us} but diverge at higher M_{us}. His results are considered reliable only at low M_{us} because of an equipment specific error. It is striking that the $\Delta\overline{St}/M_{us}$ proportionality is essentially the same for closely spaced multiple jets as for a single jet, over a wide range of *Re* and width of heat transfer surface, for industrially attractive nozzle spacings, H/w, of 2.5–8.

As enhancement of heat transfer due to throughflow increases linearly with throughflow velocity, u_s, the fraction of jet flow rate which must be removed through the surface, $Q_s = M_{us}/f$, to produce a particular amount of heat transfer enhancement is inversely proportional to the fraction open nozzle area, f. Hence, the present arrangement of closely spaced multiple jets, $S/H = 0.5$, $S/w = 2.5$, $f = 0.2$, chosen for the objective of achieving the maximum average impingement heat transfer rate, has also the advantage of producing a high ratio of heat transfer enhancement to the fraction of jet flow removed as throughflow, Q_s. At $Re = 25{,}800$ for example, with as little throughflow as $Q_s = 12\%$, $\overline{Nu}$ is enhanced by 75% for the closely spaced nozzles used here compared to 16% enhancement in a jet system of low ratio of nozzle area to heat transfer surface area, $f = 0.03$, $S/H = 6.4$ (Polat et al., 1990a).

As the throughflow effect on convective heat transfer is linearly additive, $\overline{Nu}$ with throughflow for a multiple confined slot jet system at $H/w = 5$, $S/H = 0.5$, obtained by combining Eqs. 2 and 3 with the relation $St = Nu/Re\,Pr$, is

$$\overline{Nu} = 0.094\ Re^{0.68} + 0.16\ M_{us}RePr \quad (4)$$

for $8{,}000 < Re < 25{,}800$ and $0 < M_{us} < 0.0235$. The effect of throughflow is discussed further when the combined effects of throughflow and surface motion are treated.

Heat Transfer at a Moving Impingement Surface without Throughflow

Measurements and correlation

For the effect of impingement surface velocity on multiple jet heat transfer surface velocity, v_s was varied to 8 m/s to achieve values of the nondimensional surface motion parameter, M_{vs}, in the range of 0.019–0.38 for $8{,}100 < Re < 16{,}200$. At the maximum v_s the heat flux sensor output was sampled at 2,500 Hz in order to obtain a minimum of 16 values of *Nu* per complete cycle which occurs over a distance of $2S = 50$ mm. With the close internozzle spacing chosen, $S/H = 0.5$, profiles of local *Nu* reveal nothing concerning heat transfer mechanism at the impingement surface, hence are simply the basis of determining the measurement of prime interest, $\overline{Nu}$.

$\overline{Nu}$ determined for $H/w = 5$, $S/H = 0.5$ by integrating the experimentally determined *Nu* profiles, was correlated in the form:

$$\overline{Nu} = 0.094\ Re^{0.68}\,(1 + M_{vs})^{-0.69} \quad (5)$$

over the *Re* range of 8,100–25,800 and M_{vs} values up to 0.38. Eq. 5 converges to Eq. 2 for the limiting case of a stationary impingement surface.

Analysis

Surface motion affects *Nu* profiles through two mechanisms, its effect on wall shear stress and on local ΔT. Wall shear stress is increased on the side where surface motion is towards the nozzle centerline and thereby opposes lateral jet flow. The opposite effect exists on the other side of the nozzle centerline, where surface motion and lateral jet flow are in the same direction. From this mechanism the local Nu on a moving impingement surface should increase in the approach side, decrease in the leaving side.

However the moving impingement surface also alters the local ΔT by dragging a fluid layer of different temperature in the direction of surface motion. This change in local ΔT by surface motion reduces the impingement heat transfer rate on the approach side and enhances it on the leaving side.

Thus on the approach side of the nozzle centerline, local Nu tends to increase because of the wall shear stress effect and to decrease because of the change in local ΔT. These opposing effects each act in the opposite direction on the leaving side. The experimentally determined overall effect of surface motion on $\overline{Nu}$, Eq. 5, is to decrease $\overline{Nu}$, as was found for single jets by Polat et al. (1990b) and van Heiningen (1982). Thus the two mechanisms noted are not linear and, although acting in opposite directions on the two sides of the nozzle centerline, the effects do not cancel.

$\overline{Nu}$ for a single jet, Polat et al. (1990b), was correlated for a heat transfer surface half width, S, from $3.2H$ to $6.4H$ ($8w$ to $16w$), i.e., including the off-stagnation minima and maxima and extending into the wall jet region. For the present jet system S is only $0.5H$, $2.5w$, i.e., within the stagnation region. As pressure forces dominate the flow in the stagnation region at small H/w, the effect of surface motion on Nu is less in this region than in the wall jet region. Thus as would be expected, the exponent of the $1/(1 + M_{vs})$ term, Eq. 5, for the present multiple jet system, 0.69, is smaller than that, 0.89, for a single jet.

Comparisons

Preceding studies of unconfined slot jets impinging on a moving surface provide highly contradictory findings as to the effect on impingement heat transfer of surface motion. Fechner (1971) found that $\overline{Nu}$ under single and multiple slot jets increased slightly with increasing M_{vs} in the present M_{vs} range. Subba Raju and Schlünder (1977) reported 1.2 to 1.5 times higher $\overline{Nu}$ when a single jet impinged on a slowly moving surface instead of a stationary one. At higher surface speeds they observed that $\overline{Nu}$ either stayed constant or, with increasing Re, decreased slightly. For M_{vs} varying up to 0.2 with a single slot jet Baines and Keffer (1979) found no appreciable effect on average wall shear stress over an averaging distance of $32w$. For Re of 1,300–3,300, Haslar and Krizek (1984) found that the average mass transfer rate at a slowly rotating impermeable cylinder under multiple slot jets was 5–15% higher than that at stationary flat surface. Beyond this sudden enhancement in average transfer rate for a very low M_{vs}, they measured a much smaller increase in mass transfer with further increase in M_{vs} to about 0.4.

Contrary to those four studies with unconfined jets, two studies of confined single slot jets impinging on a moving surface, van Heiningen (1982) and Polat et al. (1990b), as well as the present multiple confined jet study agree that surface motion decreases $\overline{Nu}$. The disagreement between the unconfined slot jet studies underlines the significance of entrainment of ambient air in the absence of a confinement hood, an effect which makes such results equipment specific. For reasons of thermal efficiency industrial impingement systems cannot be unconfined, so unconfined jet studies are generally of little value in engineering design.

For the study of heat transfer without appreciable effect of surface motion, M_{vs} was kept less than 0.05. That assumption is now confirmed, as at $M_{vs} = 0.05$, Eq. 5 indicates a negligible reduction in impingement heat transfer, only 3%. One important industrial application of multiple jet impingement systems, the Yankee dryer for paper, operates with quite high values of M_{vs}, about 0.3. Eq. 5 indicates that in design of a multiple slot jet system for $M_{vs} = 0.3$, use of data obtained with a stationary impingement surface would lead to underestimating the required heat transfer area by about 17%, not a negligible error in design of expensive equipment.

Heat Transfer at a Moving Impingement Surface with Throughflow

Measurements

Results are now presented for the most exacting aspect of the present study, the measurement of local and average convective heat transfer at a permeable surface at which throughflow is varied over a wide range and which is moving at up to quite high speed under multiple impinging jets. The effect of throughflow on impingement heat transfer for a broad range of surface velocities was studied for a constant Re, about 10,000, by determining profiles of local Nu for 27 combinations of M_{us} from 0 to 0.022, and M_{vs} from 0.038 to 0.38.

With closely spaced multiple jets, of flow cell aspect ratio $S/H = 0.5$, profiles of Nu provide no indication of flow and heat transfer conditions along the impingement surface. With surface motion these profiles become simply nonsymmetrical equivalents of the Figure 2 profiles, providing the basis of determining by integration the aspect of prime interest, $\overline{Nu}$.

The average impingement heat transfer results with throughflow, again expressed as Stanton number, are differenced at each of the seven levels of M_{vs} used, giving enhancement in average Stanton number, $\Delta\overline{St}$, due to throughflow at the moving impingement surface. These results, displayed on Figure 6, show

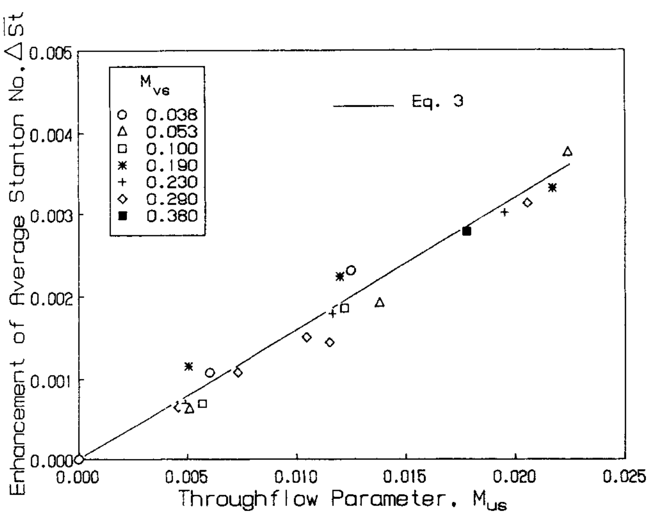

Figure 6. Effect of throughflow on enhancement of average Stanton number at a moving impingement surface.

that for multiple jets the linear relation between $\Delta\overline{St}$ and M_{us}, found for operation without surface motion effects, applies also for impingement surfaces moving at speeds up to that corresponding to $M_{vs} = 0.38$. Thus the proportionality factor, $\Delta\overline{St}/M_{us}$, is shown to be independent of both M_{vs} (Figure 6) and Re (Figure 5). Hence a single equation, Eq. 3, correlates all the experimental data for $\Delta\overline{St}$, in a relation independent of Re and M_{vs}, over the range of variables $8{,}000 < Re < 25{,}800$, $0.038 < M_{vs} < 0.38$ and $0 < M_{us} < 0.0235$. As noted earlier, Eq. 3 applies for single jets at $H/w = 2.5$, Polat et al. (1990a,b), and at $H/w = 8$, Saad (1981). Thus for multiple jet systems it appears that Eq. 3 may also be used at values of spacing both larger and smaller than that of the present experimental test at $H/w = 5$.

Comprehensive correlation

Because throughflow provides a linearly additive effect on convective heat transfer, the comprehensive relation

$$\overline{Nu} = f(Re, M_{vs}, M_{us}, Pr)$$

for multiple jets at $H/w = 5$, $S/H = 0.5$ impinging on a moving heat transfer surface where there is throughflow may be given as

$$\overline{Nu} = 0.094\, Re^{0.68}(1 + M_{vs})^{-0.69} + 0.16\, RePrM_{us} \quad (6)$$

valid for $8{,}000 < Re < 25{,}800$, $0 < M_{vs} < 0.38$ and $0 < M_{us} < 0.0235$. With throughflow at a high speed impingement surface, the excellent agreement between experiment and correlation is displayed in Figure 7.

The effect of throughflow on convective heat transfer has now been documented for single and for multiple confined slot jets impinging on heat transfer surfaces moving at up to quite high speeds. In all cases throughflow provides a linearly additive enhancement of convective heat transfer, by an amount uniform over the entire impingement surface from the stagnation point out, and varying directly with the throughflow parameter M_{us}. The proportionality, $\Delta\overline{St}/M_{us}$, is independent of both Re and M_{vs}. Moreover, the proportionality factor, $\Delta\overline{St}/M_{us}$, is 0.16 for closely spaced multiple jets at $H/w = 5$, $S/H = 0.5$, is 0.175 for

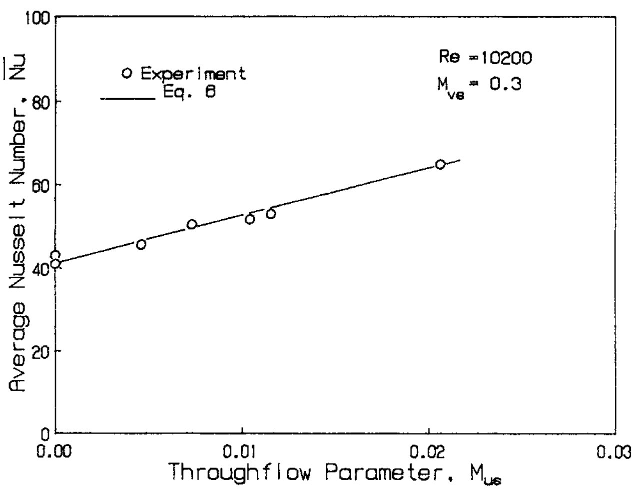

Figure 7. Effect of throughflow on average Nusselt number at a moving impingement surface: correlation and experiments.

single jets at $H/w = 2.5$, Polat et al. (1990a,b), and for the low throughflow rate results of Saad (1981) at $H/w = 8$. These findings indicate that the enhancement of average convective heat transfer by throughflow for all confined impinging slot jets, from single jets to closely spaced multiple jets, can be adequately expressed by

$$\Delta\overline{St} = 0.17\, M_{us} \quad (7)$$

at H/w spacings from 2.5 to 8 in the Re range of 8,000–58,000 and with impingement surfaces that can be either stationary or moving under the jets at high speed, up to $M_{vs} = 0.38$.

Industrial Process Application

If the permeable impingement surface were a sheet of moist paper moving at paper machine speed, then impingement without throughflow corresponds to conditions in a Yankee dryer, while throughflow without impingement corresponds to through dryers of paper. Hundreds of these two types of dryers exist currently in paper mills around the world. Conditions of the last case treated here would describe a combined impingement and through dryer for paper. Although that is not a process used in the paper industry, results have been reported by Burgess et al. (1972a,b) for experiments with a prototype of a combination dryer, called the "Papridryer," operated at conditions listed below for comparison with those of the present study.

	This Study	Papridryer
u_s, m/s	0–0.4	0.05–0.3
u_j, m/s	14–48	13–97
v_s, m/s	0.5–8	2–3.6
Re	8,000–26,000	1,000–3,000
M_{us}	0–0.024	0.0005–0.0054
M_{vs}	0.04–0.38	0.04–0.48
Q_s, %	0–12	3–34

Design modelling for the potentially important industrial process of combined impingement and throughflow drying of paper has to date rested on some important assumptions, unavoidable in the absence of experimental data. Thus in the models of Crotogino and Allenger (1979) and Randall (1984) the effect of a high speed impingement surface on impingement heat transfer was omitted. The present multiple slot jet study shows that for heat transfer surface speeds giving M_{vs} values in the range attained in paper machines, this is not a negligible effect. For example a reduction of 17% in impingement heat transfer is found at $M_{vs} = 0.3$, a value corresponding to conditions in the impingement hood of a Yankee dryer.

Again, in the absence of experimental data on the effect of throughflow on convective heat transfer, Crotogino and Allenger (1979) assume that the rate of convective heat transfer by the multiple confined impinging jets would decrease linearly with the fraction of jet flow which became throughflow, Figure 8. A comparison of this assumption with the experimental data now available may be made for this multiple jet system with $M_{us} = 0.0235$, for which $Q_s = 12\%$ of the jet flow is removed as throughflow. By the linear proportionality assumption of Crotogino and Allenger the average convective heat transfer by impingement would be reduced by 12% relative to the no

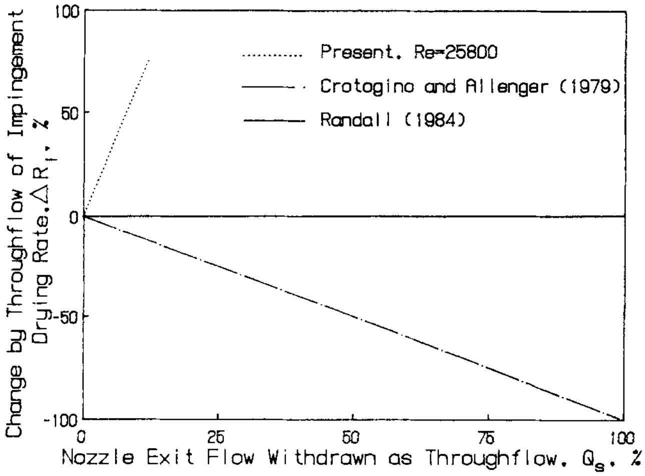

Figure 8. Change by throughflow of impingement drying rate with multiple jets.

throughflow case. The present study establishes the opposite trend, i.e., that for $Q_s = 12\%$ average impingement heat transfer would be enhanced from 53% to 76% in the Re range tested. Thus Crotogino and Allenger's model would underestimate the impingement heat transfer contribution to drying by 65% to 88% for this Reynolds number range.

Randall (1984) assumes that impingement and throughflow transport phenomena take place in sequence, with impingement heat transfer unaffected by the amount of throughflow, i.e., dependent only on nozzle exit conditions. Thus Randall's model would underestimate the impingement heat transfer contribution to drying by 53% to 76% in the *Re* range tested, Figure 8.

In techno-economic assessment of potential commercial processes for which required data do not exist, design models play an essential role. For a multiple jet system the present results provide the first experimental data concerning the effect on convective heat transfer rate of impingement surface speed and of throughflow rate, two essential elements in the potential process of combined impingement and throughflow drying of wet webs such as paper and textiles.

Conclusions

Industrial heat transfer applications of impinging jets typically involve an impingement surface moving at high speed under a system of multiple jets housed under a confinement hood for thermal recovery from the spent flow. Several experimental deficiencies limit the industrial significance of the few reported investigations of multiple jet heat transfer: use of a stationary impingement surface; use of unconfined jets, for which entrainment effects on heat transfer may be large and indeterminant; use of an exhaust flow arrangement producing crossflow of the spent air under the jets, strongly deleterious to heat transfer. The present study provides comprehensive correlations, Eqs. 6 and 7, for average convective heat transfer valid over $8{,}000 < Re < 25{,}800$, $0 < M_{us} < 0.0235$ and $0 < M_{vs} < 0.38$ for a confined multiple slot jet system with $H/w = 5$ and $S/H = 0.5$ and without crossflow. The proportionality constant for the effect of throughflow, $\Delta\overline{St}/M_{us} = 0.17$, and the $(1 + M_{vs})^{-0.7}$ factor for effect of surface motion are applicable in a much wider *Re*, H/w and S/H range, 8,000–58,000 2.5–8 and 0.5–6.4, respectively.

The effect of throughflow on convective heat transfer for jet systems impinging on moving surfaces has previously not been open to experimental study for lack of a heat flux sensor applicable to this difficult combination of conditions. With the unique heat flux sensor developed in this laboratory, Polat et al. (1991), the effect of surface throughflow on impingement heat transfer is shown to be linearly additive for the confined multiple jet system of closely spaced jets, $S/H = 0.5$, $H/w = 5$, as was established by Polat et al. (1990a,b) for a single jet at $H/w = 2.5$ for heat transfer widths up to $S/H = 6.4$. The independence of the throughflow enhancement ratio, $\Delta\overline{St}/M_{us} = 0.17$, to major changes in geometry, i.e., from a single jet to closely spaced multiple jet systems, is a finding of considerable scope which implies that this throughflow enhancement factor applies over a range of parameters much broader than tested here.

The present investigation provides data on the effects of impingement surface speed and throughflow rate required to construct more realistic design models for the assessment of a potentially attractive industrial process, combined impingement and throughflow drying of wet webs such as textiles and paper.

Notation

C_{Pa} = specific heat of air, J/kg · K
C_{Pj} = specific heat of air at nozzle exit temperature, J/kg · K
C_{Pp} = specific heat of the permeable substrate, J/kg · K
f = fraction nozzle open area, $w/2S$
H = nozzle-to-impingement surface spacing, mm
h = convective heat transfer coefficient, W/m² · K
k_{eff} = effective thermal conductivity of substrate, W/m · K
k_j = thermal conductivity of air at nozzle exit temperature, W/m · K
M_{us} = mass velocity ratio, u-direction at the impingement surface, $\rho_s u_s/\rho_j u_j$
M_{vs} = mass velocity ratio, v-direction at the impingement surface, $\rho_s v_s/\rho_j u_j$
Nu = Nusselt number, hw/k_j
Nu_0 = stagnation Nusselt number
$\overline{Nu}$ = average Nusselt number
q_s = heat flux at the impingement surface, W/m²
Q_s = fraction of jet flow removed as throughflow = M_{us}/f
Re = Reynolds number, $\rho_j u_j w/\mu_j$
ΔR_i = enhancement by throughflow of impingement drying rate, %
S = nozzle centerline-to-exhaust port centerline spacing, mm
St = Stanton number, $h/u_j\rho_j C_{Pj}$
$\Delta\overline{St}$ = enhancement of average Stanton number by throughflow
T = temperature, K
T_j = air temperature at jet nozzle exit, K
T_s = heat transfer surface temperature, K
ΔT = temperature driving force, K
u_j = velocity at jet nozzle exit, m/s
u_s = superficial throughflow velocity, m/s
v_s = velocity of heat transfer surface, m/s
w = nozzle width, mm
x = distance in the substrate normal to heat transfer surface, mm
y = extent of heat transfer surface from nozzle centerline, mm

Greek letters

μ_j = viscosity of air at nozzle exit temperature, kg/m · s
ρ_a = density of air, kg/m³
ρ_j = density of air at nozzle exit temperature, kg/m³
ρ_p = density of substrate, kg/m³
ρ_s = density of air at heat transfer surface temperature, kg/m³

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