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The localized, high heat transfer rates are accompanied by a rapid decay as the jet spreads away from its center of impact. Furthermore, in the regions between neighboring jets, heat transfer rates are less susceptible to rational correlation of the performance data. It appears that some of the drawbacks can be alleviated by using a series of two-dimensional or line jets impinging onto the inner surface of a blade, resulting in a line of impingement.

In this investigation, the jet spacing, jet height and jet Reynolds number were varied within limits of practical interest commensurate with turbine cooling application. All jets were of equal width. Two series of experiments wer run: one with equal flow for the jets, and the other with increasing flow rates for jets on either side of the symmetry line. A principal feature of the two-dimensional multi-jet configuration is that the heat transfer coefficient shows a less steep decay distribution than a single axis-symmetry jet. Furthermore, by varying the flow rates among the line jets, the surface heat transfer distribution can be modified to a certain extent. Correlations of the average heat transfer coefficients over the region facing the jets are given in this paper.

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IMPINGEMENT COOLING OF TURBINE AIRFOILS BY MULTIPLE TWO-DIMENSIONAL JETS

L.S. Han[1], N. Kayansayan[2], C.W. Dantzer[3]

ABSTRACT

For internal cooling of turbine vanes by convection type-impingement of compressor bleed air, a frequently used arrangement is a multitude of round holes through which cooling air is impacted on the inner surface of a turbine blade. High heat transfer rates at the impingement points are utilized to remove the heat load imparted by external high-temperature gas streams.

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NOMENCLATURE

A	Injection slot width
B	Center-to-center distance between jets
C	Coefficient, equation (1)
H	Channel height
h	Heat transfer coefficient
h_o	Stagnation-point heat transfer coefficient
$\bar{h}_i$	Average heat transfer coefficient in injection section
k	Thermal conductivity
L	Impingement plate length
N_u	Injection-section Nusselt number ($h_i H/k$)
Nu_o	Injection-section Nusselt number (h_o/Ak)
p	Gage pressure
p_o	Stagnation-point gage pressure
Re	Injection Reynolds number ($V_j/H\nu$) or ($\bar{V}_j H/\nu$)
Re_o	Stagnation-point Reynolds number ($V_j A/\nu$)
u	Velocity parallel to impingement surface
V_j	Injection velocity
$\bar{V}_j$	Average injection velocity
x, y	Coordinates, see Figure 2
ν	Kinematic viscosity
γ	Ratio of flow rates between jets

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INTRODUCTION

For internal cooling of turbine airfoils, a scheme combining the features of impingement of two-dimensional slot jets and convection heat transfer in a duct parallel to the airfoil contour may prove attractive under certain combinations of operational parameters. The essential components of such a scheme are illustrated in Figure 1. The method has the advantages: (i) that localized jet cooling with attendant high heat transfer may be tailored to accommodate a particular desired heat transfer distribution and (ii) cooling air after impingement can be further used as a convective cooling medium in the flow channel where the airjets are accumulated.

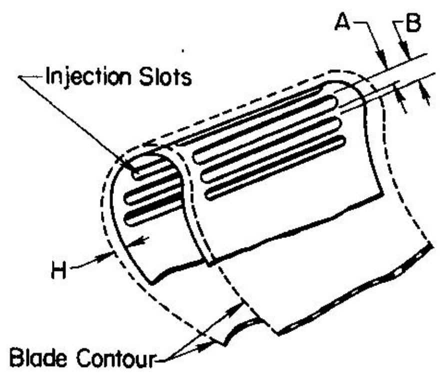

Figure 1. Blade Cooling by Two-Dimensional Jets

To render the contemplated configuration to experimentation and analysis, a first-round modelling is presented in Fig. 2 wherein the channel for flow accumulation around the circumference of the airfoil is unravelled to form a straight flow duct. Its bottom surface denotes the airfoil inner surface and its upper boundary containing two-dimensional slots is the plenum cavity baffle. In this preliminary modelling effort, curvature effects of the airfoil contour are neglected based on the reasoning that near the leading edge of an airfoil, where curvature is invariably severe, the injected air is confined to a thin layer whose dimension is a fraction of the jet width. And since the jet width is comparable to the local curvature in this study, the curvature effects in this thin parallel layer can therefore be neglected. Additionally, as a series of "curtain" jets are impacted into the blade inner surface giving rise to localized thinning of the boundary-layer type flow near the impact points, the resulting phenomena of jet expansion, interactions of the jets with the recirculation zones, and lastly the merges of the jets would be of high-order influence than those caused by the airfoil curvature.

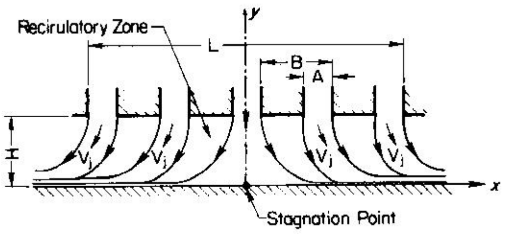

Figure 2. Schematic Representation of Two-Dimensional Jets

In this study, then, the basic objectives are two-fold:

(i) To investigate experimentally the heat transfer distributions on the impacted surface by a series of two-dimensional jets, and

(ii) To ascertain the velocity and frictional characteristics in the flow channel as a result of flow accumulation from the parallel jets.

IMPINGEMENT COOLING

In reviewing heat transfer literature on jet cooling, the circular jets are the most extensively studied configurations, notably by Metzger, et al [1], Sparrow and his colleagues [2], and Cobonpue and Gardon [3]. In these investigations as well as others, an inherent feature of circular jet cooling is the localized nature of heat transfer effectiveness which is confined to the immediate vicinity of the impingement point. In the berm area between two adjacent jets, heat transfer is appreciably reduced because of the interaction between, and annihilation by, two opposing streams on the surface. In a broad sense, the resulting mingling and merging among a matrix of circular jets invariably entails momentum loss which reduces the heat transfer effectiveness.

On the other hand, a two-dimensional jet issued from a narrow slot gives uniform cooling effectiveness along its line of impingement. Structurally speaking, a curtain jet of this nature may be perceived as a series of individual circular jets having been pre-arranged and pre-assembled to impinge onto a flat surface so as to eliminate collisions between adjacent jets, which would occur were they not so pre-assembled. For a single two-dimensional jet, Gardon and Akfirat [4] provided heat transfer data on the plane of impingement. Their heat transfer data expressed by the Nusselt numbers were correlated with the Reynolds numbers. In both of the dimensionless combinations, the initial jet width was used as the characteristic length, which varied from 0.15 to 0.64 cm. Similar results were also obtained by Cadek [5], who extended the range of slot width variation. Douglas et al [6] numerically studied the heat transfer characteristics of a semi-confined laminar impinging slot jet. A significant finding of the works cited above was that the stagnation-point Nusselt number was proportional to the square root of the Reynolds number with the proportionally constant decreasing for downstream positions from the stagnation point.

THE EXPERIMENTS

Flow System. The basic flow apparatus as depicted in Figure 2 consists of five evenly spaced injection slots of dimensions 1.27 cm by 58 cm, the former being the jet or slot width. The magnitude of the transverse dimension of the jet fulfilled the two-dimensionality requirement of the flow configuration which was made symmetrical with reference to the middle jet.

Air flow through each injection slot was individually regulated by a separate air supply plenum mounted on top of each injection slot. The plenum chambers were in the shape of flat approach sections with a flow cross-section of 1.27 cm by 66 cm with the latter dimension overlapping the 58 cm transverse dimension of the slot by 4 cm at each end. The lead flow-length in the plenum was 48 cm, along which two banks of flow straighteners were employed to achieve a uniform discharge velocity at the injector exit. The discharge velocity profile of the injection slot was surveyed along the 58-cm direction and was found to have a maximum deviation of 4 per cent from the mean value.

The spacing between adjacent jets was varied, when necessary, by the placement of spacer blocks flush with the exit plane of the injector openings. The main flow collection channel was completed by plexi-glass extensions beyond the area where the jets were situated.

For the lower surface of Figure 2, two designs were used. In the pressure-friction study, the plate was a 0.6 cm thick aluminum sheet, 66 cm by 100 cm, with the latter dimension parallel to the collected flow direction in Figure 2. On the center line of the plate and along the main flow direction, forty static pressure taps of 0.08 cm diameter were provided. Also on the plate, access holes were drilled to facilitate Pitot velocity probes for measuring velocity profiles across the flow gap in the channel. In the heat transfer study, a fully instrumented heat assembly constituted the lower plate of Figure 2 and is described below:

Heater Plate. An aluminum plate 1.2 cm thick, 66 cm by 100 cm was placed such that the larger dimension of the two was aligned with the flow direction (the x-axis of Figure 2). Along this length, two-thirds of it was heated by fifteen electric heaters housed in fifteen grooves milled transverse to the x-axis on the under-side of the 1.2 cm thick plate. The grooves were 0.32 cm deep, 2.54 cm wide, and 58.4 cm long. Two types of heater strips with power densities of 1 watt/cm^2 and 0.75 watt/cm^2 were used with the larger ones housed in the grooves near the central jet.

Good thermal conduct between the heaters and the aluminum plate was obtained by using a thermal compound. On the side facing the injection slots, seven recesses, 1.6 cm square and 0.038 cm deep, were milled along the centerline of the plate. The recesses were to accommodate seven RDF-20454-3 heat flux sensors to monitor local variations of the heat transfer coefficients. Good thermal coupling between the sensors and the surface was achieved by permanently attaching them to the surface. The aluminum plate, along with the heating strips and heat flow sensors, was mounted in a 2.5 cm thick mahogany board with 5 cm thick Styrofoam located beneath the wood section. The Styrofoam, in turn, was supported by a sheet of plywood. Analysis of heat transfer through the insulation layers and of radiation losses from the plate surface to the upper aluminum wall showed that approximately 5% of the thermal energy from the heaters was lost to the surroundings.

Instrumentation. Air supply to each plenum was metered by a separate glass-tube variable area rotameter of the Schuette and Koerting Type 20-400 with a maximum capacity of 3.5 m^3/min at the standard atmospheric conditions. Regulation of individual air supplies was accomplished by individual valve-controls to the plenum chambers. Static pressures at the top locations on the bottom plate were measured by a Type-570 Barocel pressure transducer and a Type 1173 Analog Barocel manometer. Velocity transverses across the flow channel gap were conducted by a boundary-layer type total pressure probe which had a 0.063 cm sensing diameter but was flattened to a dimension of 0.03 cm at its tip.

For temperature measurement of the heater plate, sixteen 24-gage-copper-coustantan thermocouples were positioned in the wells drilled into the aluminum plate and were located 0.15 cm from the surface facing the injection slots. Temperature signals from these thermocouples were used to adjust the power input to the heater strips for maintaining a uniform surface temperature which was generally held at about 65°C. With air injected at a temperature of 15°C, the resulting temperature difference was in the order of 50°C. The surface temperature variation was controlled to a maximum variation of 3°C. Electrical power to the heating elements was supplied by several Variac auto-transformers which were individually connected to the circuits in conjuction with a wattmeter or a voltmeter-ammeter pair for power measurement.

Procedure. In the present study, the important parameters are the injection Reynolds number, Re, the slot spacing B between jets, and the channel height H. The ranges over which these parameters were varied are as follows:

Re	100	--------------	15000
B	2.54	--------------	5.08 cm
H	1.27	--------------	3.81 cm

In the pressure-friction part of the study, the flow rates through all injection slots were equal, as were for most of the heat transfer part of this investigation. In the heat transfer part of this study, the injection flow rates among the jets were, in a first series of tests, made equal. Subsequently the individual injection rates from the slots were apportioned in a geometrical ratio of γ = 1.5, and then γ = 2.0; i.e. beginning from the middle (stagnation) jet, the next downstream jet had a flow of γ-times of the middle jet and so on.

In the heat transfer runs, the seven heat flux gages on the face of the heater plate were the principal source for determining the heat transfer distributions on the impingement plate. The output signals consisted of milivolt readings across the gage thickness and the surface temperatures of the gages. The heat flux gages were installed on one-half of the heater plate surface, with the center of the first gage aligned with the central jet axis so that in one setting seven heat transfer rates were obtained. By shifting the heater plate relative to the stationary injection slots, additional numbers of heat transfer rates became available.

THE FLOW CHARACTERISTICS

Before heat transfer measurements, representative surveys of the pressure and velocity distributions were conducted on the basis flow configuration. They served the purposes of determining, at least qualitatively, the pressure distribution along the impingement plate, the velocity profiles across the flow channel at selected stations from the stagnation point of the central jet, and also flow symmetry of the basic arrangement.

For checking flow symmetry, the total flow discharge through one of the two open ends was determined by a Pitot probe traverse across the exit section. In principle, the measured flow should be equal to one-half of the injected flow through the injection slots. In reality, a difference between these two sources of flow rate reckonings of 5 per cent was noted but was considered the best achievable.

Typical pressure distribution patterns are shown in Figure 3 for three injection rates, equal for all five slots. The geometrical factors are B/A = 3 and H/A = 2, i.e. the jets are three jet-widths apart and the channel height or clearance is twice the jet width. A significant feature of the curves in Figure 3 is the dip in the distribution at a position beyond the last injection port. The pressure-minimum point generally occurred at high Reynolds number values and was obviously a manifestation of a vena contrata at that position in the flow channel. Subsequent pressure recovery after a distance of two jet widths indicated that the flow channel was again refilled. Hence, from this representative flow picture, it is possible to construe the flow into two parts: one is in the region facing the injection slots or the injection region in which the injected flow from successive jets reinforced the preceeding ones in a manner similar to walljets; the other is downstream of the injection region and is very similar to flow in a flat channel with developing velocity and temperature profiles. Local variances from such a simplistic division can, of course, exist such as the jet contraction phenomenon just discussed; in fact the local flow contractions due to successive cross jets downstream are undoubtedly the agents responsible for heat transfer peaks.

Within the injection region, a further flow demarkation can be attempted. Near the central jet, symmetry dictates that the flow be very much like a single jet bifurcating at the stagnation or symmetry point. Away from the stagnation region, the impingement surface and the free jet contour make the flow very similar to a wall jet, either in a laminar or turbulent mode. By making Pitot tube traverses

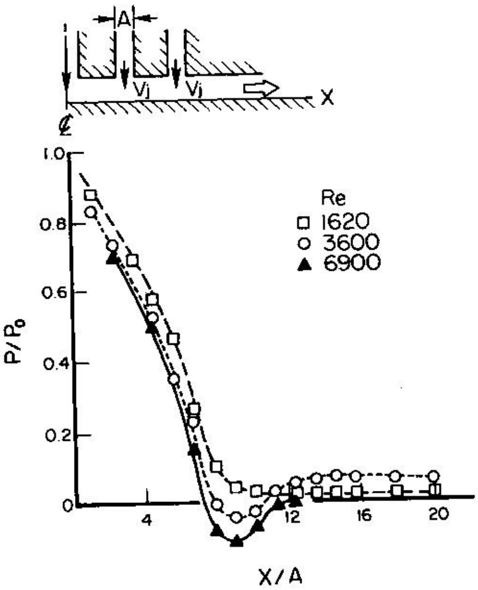

Figure 3. Typical Pressure Distributions on Impingement Surface

across the flow channel, typical velocity (parallel to the impingement plate) profiles obtained are illustrated in Figures 4 and 5. The geometrical configurations for which measurements were made are shown in both figures and are in scale. The jets are spaced four units apart (the jet width equals one unit) and for this separation distance the individual jet characteristics are more distinct than when they are closer to one another. In Figure 4, the channel clearance is one unit or equal to the jet width. The velocity distributions shown in these two figures are at the locations indicated, i.e. near the midpoints between the neighboring jets. All essential characteristics of a wall jet are present: there is a high-velocity region near the wall, and the boundary layer thickness (maximum velocity location from the wall) appears to decrease with increasing Reynolds number. In Figure 5, the channel height is twice the jet width and all other geometrical factors are unchanged. The velocity distributions shown in Figure 5 appear more compressed towards the wall than in Figure 4. This is, however, not quite true in terms of the physical dimension, for the configuration of Figure 5 has its H-value twice as large as the H-value of Figure 4. Hence, in terms of physical coordinates, the velocity distribution in both figures are in reality nearly identical. Indeed, the distributions in Figure 5 are more like those in a wall jet since the impingement plate is farther away from the jet exits.

HEAT TRANSFER RESULTS

Validation Tests. A preliminary round of heat transfer measurements was undertaken to check out the instrumentation used in this study. The configuration chosen was than of a single two-dimensional jet located at a distance of eight times the jet widths from an impingement surface; for this configuration, heat transfer data were made available by

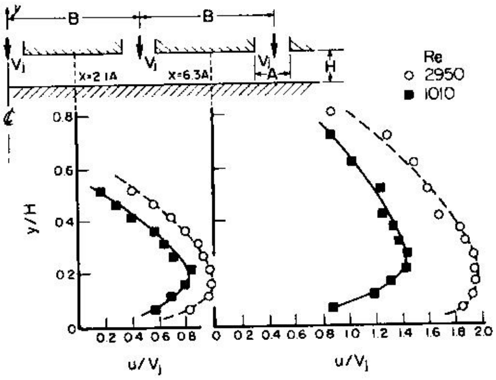

Figure 4. Effect of Injection-Section Reynolds Number on Velocity Profile, $B/A = 4$, $H/A = 1$.

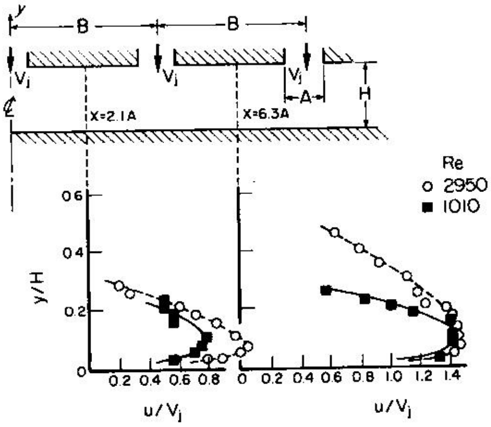

Figure 5. Effect of Injection-Section Reynolds Number on Velocity Profile, $B/A = 4$, $H/A = 2$.

Gardon and Akfirat [4]. By measuring the heat transfer rates with the heat flux gages on the heated plate, average heat transfer coefficients over the heated portion of the surface were obtained; the data so obtained were compared with the correlation equation given in Gardon and Akfirat's paper and are shown in Figure 6, where the agreement was deemed satisfactory and well within the accepted engineering accuracy.

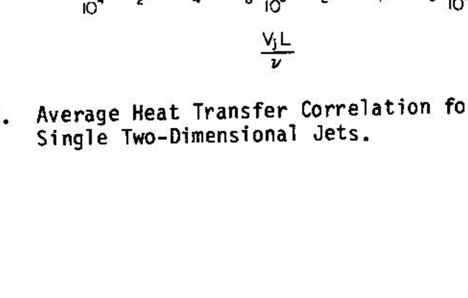

Figure 6. Average Heat Transfer Correlation for Single Two-Dimensional Jets.

The next series of data comparison is the lateral heat transfer distribution on the impingement surface. As the stagnation point heat transfer data have appreciable variances among different experiments because of the initial jet profile, jet turbulence, etc., comparison was therefore made on a normalized basis, i.e., the ratio of the local heat transfer coefficient to the stagnation value. Data obtained in this preliminary round of tests were compared with those given by Cadek [5], and the relative distribution curves are given in Figure 7 for two jet heights at a jet Reynolds number of 4650. Although the curves did not duplicate each other, as no two-experimenters had produced identical distribution curves, the general bell-shaped distribution was evidenced in this illustration.

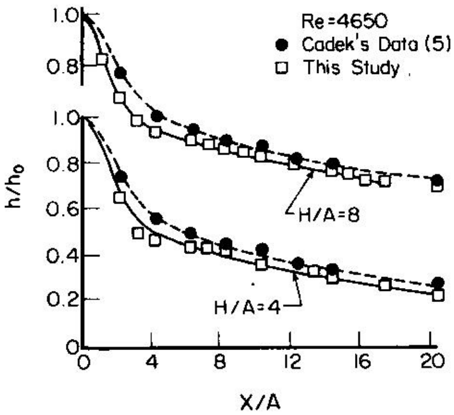

Figure 7. Relative Heat Transfer Distribution Due to a Single Two-Dimensional Jet

A third series of validation tests was undertaken to corroborate the output from the heat flux gage sensors and those from the heating strips underneath the surface plate. This series was conducted for the stagnation region only. Using the electrical input to the heater situated directly under the stagnation jet, a heat transfer coefficient was obtained and compared with the sensor output at the same location. Noting that the value obtained by the heater output was not exactly accurate because of lateral conduction along the heated surface, close correspondence between these two sources, nonetheless, lent credence to the basic instrumentation based on the sensor output. In this series of tests, three height ratios of H/A = 1,2,and 3 were used with all five jets spaced three jet-widths apart.

Stagnation-Point Heat Transfer. The stagnation-region always represents an important criterion to characterize a jet-cooling system. This is particularly true for turbine blade cooling application, for the stagnation-point heat transfer by internal impingement should coincide with the external stagnation so that the maximum heating externally can be matched and mitigated by maximum cooling internally.

For two-dimensional free jets Cadek [5] has analyzed and conducted experiments on the stagnation-point heat transfer; his analytical and experimental results agreed with one another for jet heights up to H/A = 4, i.e., the jets are located up to four jet-widths from the impingement surface. His analytical results can be expressed by an equation:

$$Nu_o = C\sqrt{Re_o} \tag{1}$$

where the coefficient C depends on the jet-height ratio; his values for C are 0.476, 0.467 and 0.451 for H/A = 1,2, and 3 respectively.

In the present investigation, however, the experimental configuration consisted in a multitude of line jets issuing into a common flow channel; and by virtue of symmetry, the impingement point directly underneath the middle jet constitutes the stagnation point. Its heat transfer coefficients were obtained by the heat flux sensor and based on the temperature differentials between the impingement surface and the injected air, as were for all other locations in this investigation. Expressed in stagnation point Nusselt numbers and Reynolds numbers, data for height ratios of H/A = 1, 2, and 3 are presented in Figure 8 in three separate groups; and for each height ratio, at jet-spacing pitches of B/A = 2,3, and 4, identified by different symbols. For Reynolds numbers above 600, a single line of one-half slope can be used to represent the data points on the average for each group of a fixed height; the effect of jet-spacing ratio in each group is however not discernible in the data presentation. The lines in Figure 8 correspond to equation (1) with the coefficient C equal to 0.616, 0.590 and 0.582 for H/A = 1,2, and 3 respectively. The trend of decreasing C-value is consistent with, but higher by 20 per cent than, those obtained by Cadek [5]. From a geometrical consideration, it would be expected that when the jets are spaced wider and wider apart, the stagnation-point heat transfer data would revert back to those of a single jet. However, in the range of B/A up to 4, it was not possible to establish the expected approach towards Cadek's values for single jets.

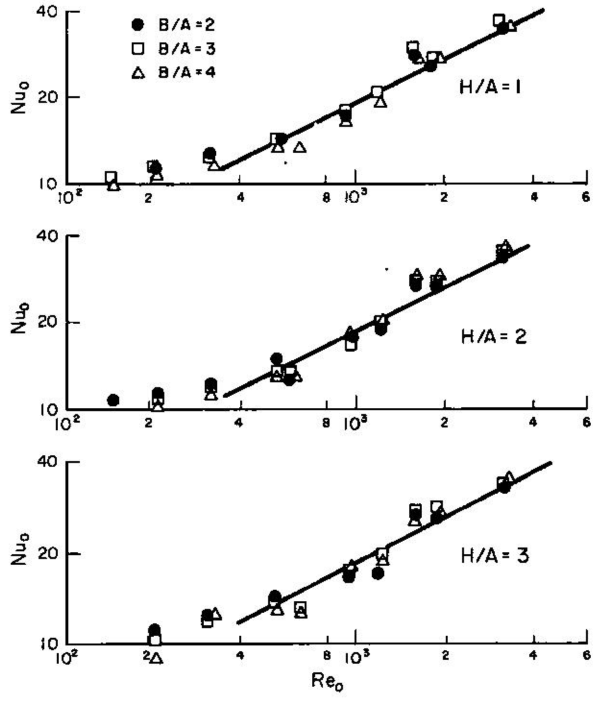

Figure 8. Stagnation-Point Heat Transfer in Multi-Jet Assemblies.

A further note of interest is the data in Figure 8 for Reynolds numbers below 600, deviation and leveling-off from the straight line correlation equation is quite apparent. At low Reynolds numbers, it is not unreasonable to suppose that there is a definite change of the flow pattern, as supported by the velocity survey data in Figure 5 which shows that at a lower Reynolds number, the maximum velocity in the walljet-like distribution is farther away from the impingement surface than at a higher Reynolds number. In other words, the flow becomes less walljet-like and more pipeflow-like.

Patterns of Heat Transfer Distributions (Equal Injection Rate) A distinguishing feature of the present investigation is a number of line jets impinging sequentially onto a surface. The resulting bifurcating flow parallel to the surface may be thought of either as a single jet reinforced by downstream jets, or alternatively as a succession of crossflow jets that depress and intercept the upstream oncoming flow. In the common flow channel shown in Figure 2, momentum considerations lead to the conclusion that subsequent (downstream) jets would be deflected away from their original orientations owing to a pressure differential across the jet contours. The amount of jet deflection from its otherwise normal impingement direction is naturally increasing with increasing jet height. These facets are to a large extent reflected in the patterns of heat transfer distributions found on the impingement surface.

From the above outline of the flow structure, the resulting heat transfer picture may be put together as follows:

Starting from the normal impingement point directly underneath the stagnation jet, heat transfer commences with a high value. Along the surface, the thermal boundary grows resulting in decreasing heat transfer values until transition from laminar to turbulent flow with an attendant rise in heat transfer, or until the base flow is interrupted by a downstream jet, in which case the hydrodynamic and thermal boundary layers are locally reduced, thereby momentarily raising the surface heat transfer value. Thus, a surface heat transfer rise may be caused by either one of these two phenomena discussed. Heat transfer results obtained by Gardon and Akfirat [4] and by Cadek [5] amply illustrated the transition effect in raising the surface heat transfer rate. However, in the present investigation most of the heat transfer peaks observed are attributable to the jet confluence effect with the exception in a few instances where these two effects - jet merging and transition - seem to take place in close proximity to each other.

Figure 9 shows typical heat transfer distribution curves obtained by Kayansayan [7] from whose work a portion of the material in this paper is excerpted; these are normalized with reference to the stagnation-point heat transfer values and are for a fixed jet height H/A = 2 and a fixed Reynolds number of 2200. Starting from the first distribution bor B/A = 6, it should be noted that two peak values are located at x/A = 7 and 13, and are apparently associated with the two downstream jets at x/A = 6 and 12. At about x/A = 16 another mild heat transfer rise is observed which is probably the result of boundary layer transition. The second curve for B/A = 4 exhibits twin peaks at x/A = 5 and 8 which roughly correspond to the jet locations of x/A = 4 and 8. For the third curve, however, only a single hump is apparent, based on the data points available; it is quite dificult to interpret this particular manifestation and is probably caused by transition and jet confluence simultaneously. As the jets are crowding each other, for instance at B/A = 2, the discrete heat transfer characteristics of individual jets become less distinct as is evident in the distribution curve for B/A = 2.

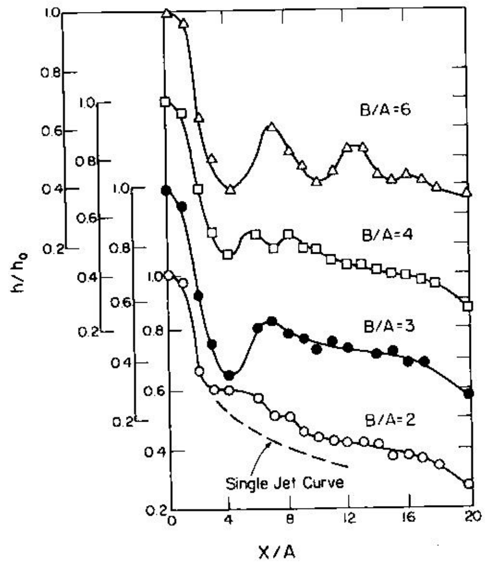

Figure 9. Effect of Jet Spacing on Heat Transfer Distribution, H/A = 2, Re = 2200 (Equal Injection Rates)

Further delineation of the heat transfer distribution is presented in Figure 10, where the jet spacing is constant and the jet height is varied from H/A= 1 to 3. These normalized curves serve to bring out the fact that heat transfer variations from valley to peak are more pronounced for small H/A than for larger values. This particular feature emphasizes impairment of a jet by virtue of deflection and loss of the jet strength at larger values of H/A.

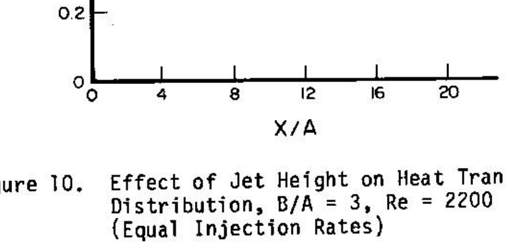

Figure 10. Effect of Jet Height on Heat Transfer Distribution, B/A = 3, Re = 2200 (Equal Injection Rates)

The foregoing discussion applies to a series of normal jet with equal flow rate. A common feature of virtually all combinations of the jet height and jet spacing is the higher heat transfer than a single jet would produce. Counting the stagnation heat transfer as one peak, then there are as many peaks as there are jets, at least for jets fairly apart from each other; these peak values are in a descending order. This observation prompted a natural question whether the rather uneven distribution can be mitigated by controlling the injections at different locations. In some applications it may be desirable to have a more uniform heat transfer distribution than those typified by Figure 9 and obviously a re-distribution of flow such that the injected rates are in ascending order should produce the desired effect. Hence, with this prospect in mind, the investigation of Dantzer [8] was carried out with the flow rates in geometrical ratios from the stagnation jet to the succeeding ones.

Patterns of Heat Transfer Distributions (Unequal Injection Rates) The flow rates in the jets were adjusted so that a constant ratio of flow rates between two neighboring jets was maintained on either side of the symmetry line. In this apportionment, the flow rate of the first downstream jet of the symmetry jet is γ-times that of the symmetry jet and the same ratio prevails between the third and second jet. The purpose is, of course, to raise the successive peaks to the same level as that at the stagnation jet. In Dantzer's [8] experimental work two flow rates γ = 1.5, and 2.0 were used to distribute the injection flows among the jets. Typical resulting heat transfer distributions are shown in Figure 11 for $H/A = 2$ and $B/A = 4$ at an average Reynolds number of 6320 for the three jets on one side of the symmetry line; the three curves are, therefore, for an equal total flow. For $\gamma = 1$, the flows in thre three jets are equal; for $\gamma = 1.5$, the flows in the jets starting from the center and counting outward are respectively 0.631, 0.947, and 1.42 times that of the jet flow when $\gamma = 1$.

At $\gamma = 1$, the heat transfer distribution shows decreasingly peak values at the approximately the local jet impact points. At $\gamma = 1.5$, the central jet flow is reduced to 0.631 of the original flow while the outer-most jet flow is increased to 1.422 times the intial flow rate; the local peak values for the new flow distribution show a less steep decline than for $\gamma = 1$. At $\gamma = 2$, the flow in the central is further curtailed to 0.428 of the original flow, with the resultant heat transfer peaks now arranged in an ascending order. More extensive results obtained by Dantzer [8] indicate that three peaks would have equal heat transfer when γ is 1.75 approximately.

Average Heat Transfer Correlation As in most heat transfer investigations, the average heat transfer coefficient is an important parameter particularly from an application viewpoint. Because of the involved geometry of impingement-convection cooling in this investiation, it appears reasonable to divide the impingement surface into two parts: (1) an area directly opposite to the group of injection slots, defined as the region from the line of symmetry to the extreme right edge of the right-most injection slot, and (2) the remainder of the surface which forms a boundary of the convection-channel downstream of the injection slots. In the first portion, the fluid dynamic process is characterized by a succession of normal impinging jets interacting with one another, while in the succeeding portion the principal feature is the simultaneous developments of flow and temperature profiles, commencing from the interface section between the two portions. In view of the objectives of this study, the focus is on the injection-section average heat transfer.

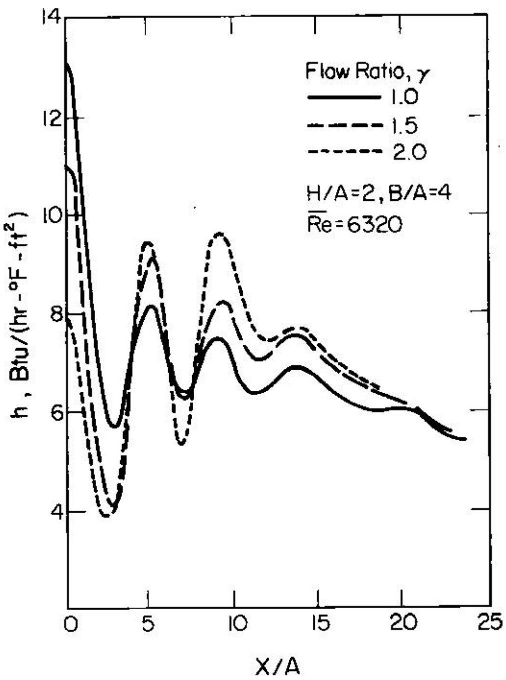

Figure 11. Effect of Uneven Jet Flow Rates on Impingement Surface Heat Transfer Distribution.

Considering the principal factors which influence the overall heat transfer on the impingement surface, the channel Reynolds number characterizing the channel height H and the jet velocity V_j (or $\overline{V}_j$ in the case of unequal jet flow rates) must be a significant parameter. It is in reality a hybrid parameter taking into account the effects of the channel as well as the jets and is organized by $(\overline{V}_j H/\nu)$. Similarly, the averaged heat transfer coefficient is correlated through the Nusselt number $(\overline{h}_j H/k)$. Other ratios describing the jet-spacing B/A and the channel aspect ratio H/A become auxiliary parameters.

One important observation with regard to the average heat transfer coefficient is that it is virtually unaffected, though not surprisingly, by the unequal flow rates among the jet cluster. The average heat transfer correlation for $\gamma = 1$ (equal flow rates among jets) obtained by Kayansayan [7] is undistinguishable from the data for γ = 1.5 and 2 obtained by Dantzer [8]. A single equation relating the two principal parameters given by:

$$(h_j H/K) = 0.064 (\bar{V}_j H/\nu)^{0.75} \qquad (2)$$

is found to be valid for the ranges of the parameters indicated by: (i) γ = 1 to 2.5, (ii) B/A = 2 to 4, (iii) H/A = 1 to 3 and (iv) Re > 500. The exponent of the Reynolds number is a reminiscence of turbulent flow in channel for which an exponent of 0.80 is an established experimental value. Equation (2) represents an average correlation between the two primary dimensionless numbers; however, the influences of the other parameters - flow rate ratio between jets, jet spacing, and jet height - cannot be discerned from the experimental data over the ranges investigated.

As to the first factor, apportionment of the flow among the jets results in an adjustment of the heat transfer peaks as illustrated in Figure 11, but not much change in the average heat transfer over the injection region. To understand the jet-spacing effect, consider a single jet impinging the flat surface of Figure 2. The jet contour starts to thin out as it turns after impingement; the thinning process continues for some distance and then the reverse takes place because of friction until the entire channel is filled. The boundary layer, starting from the impingement point, grows eventually into the channel height. Succeeding transverse jets are therefore more effective in reducing the boundary layer, thereby increasing heat transfer locally, when the layer is thick; in other words, when a jet is located farther away from the preceding one. Thus, in Figure 9, there are two distribution curves, one for B/A = 6 and the other for B/A = 2. When succeeding reinforcing jets are far apart (B/A = 6), the heat transfer peaks are distinct as compared with the lower curve for B/A = 2 when the jets are almost adjacent to one another. In terms of injection-section heat transfer coefficient as defined, the values for these two configurations, for example, were obtained by averaging over a length from x/A = 0 to x/A = 13 for B/A = 6, and from x/A to x/A = 5 for B/A = 2 respectively. Because of the unique way the average heat transfer coefficient is defined, the effect of B/A is not explicitly present in equation (2).

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