Investigations of Heat-Transfer Coefficients for Air Flow Through Round Jets Impinging Normal to a Heat-Transfer Surface

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In this paper, empirical equations of local as well as average heat-transfer coefficients of single jet system were derived. Two aspects of multiple jet systems have been studied. One concerns mainly the uniform distribution of heat-transfer coefficients and economy of power consumption. The other concerns the high magnitude of heat-transfer coefficients and the interference among jets. The experiments were conducted at Reynolds number from 10^3 to 10^4 and hole size from $^1/_8$ to $^1/_4$ in. diameter. An attempt was made to correlate empirical data to render practical application possible.

Introduction

The application of high velocity air jets to heat or cool a normal heat-transfer surface has tremendous engineering potential in various industries. However, due to the complexity of the system, no theoretical treatment has yet been evidenced and only very scant experimental data are available in literature for reference. Vickers [1][1] studied local heat-transfer coefficients of fluid jet impinging on a normal surface at the laminar flow region. His empirical equations applied to Reynolds number 250 to 950, the distance between the jet orifice and the surface of the target plate ranging from $8D$ to $20D$, and the values of X/D ratio 1 to 3.2 (diameter of the jet orifice being held at $D = 0.0507$ in.). Gordon and Cobonque [2] measured local as well as average heat-transfer coefficients for single and multiple nozzle jets at a Reynolds number of 7000 to 112,000 with the nozzle diameter 0.125 to 0.354 in. Freidman and Mueller [3] reported their experimental data regarding average heat-transfer rates of multiple jet air flow from slots, holes, and nozzles parallel to, or impinging on, the heat-transfer surface. They studied the effects of injection angle, spacing, hole size, free (open) area of jets, plate width, and so on, on heat-transfer coefficients. Perry [4] made further studies on the effect of impingement angles on heat transfer, the maximum gas temperature and velocity being 750 deg F and 250 ft/sec. Daane and Han [5] investigated the interference of the impingement air flow with the spent air exhaust flow in a multiple jet system.

[1] Numbers in brackets designate References at end of paper.

Contributed by the Heat Transfer Division of The American Society of Mechanical Engineers and presented at the ASME-AIChE Heat Transfer Conference and Exhibit, Houston, Texas, August 5-8, 1962. Manuscript received at ASME Headquarters; April 30, 1962. Paper No. 62—HT-31.

The present work was stimulated by both practical and academic reasons. The designer of industrial jet cooling or heating equipment wants to know the correlation between physical characteristics of the system so as to obtain uniform distributions as well as high magnitude of heat-transfer coefficients. The academic interest lies, of course, in gaining a deeper insight into the complexity of this problem.

The purposes of this investigation are:

1 To study the single jet system with an extended range of heat-transfer conditions, to obtain test data on local as well as average heat-transfer coefficients of the system, and to derive empirical equations.

2 To supply sufficiently detailed data about multiple jet systems for practical applications. Both a simple multiple jet system and a general multiple jet system have been treated, the former dealing mainly with the uniform distribution of heat-transfer coefficients and economy of power consumption and the latter being designed to simulate a rotating cylinder.

3 To compare the experimental data obtained from this work with those of other researchers.

Test Apparatus

1 Tests of a Single Jet. The test unit as shown in Fig. 2 consists of an insulated plenum chamber 12 in. on a side and a frame for supporting the heat-transfer plate. Hot air is introduced into the plenum chamber from a separate fan-burner system (Fig. 1). The chamber is insulated with 1-in-thick marinite to insure a constant air temperature. A $^1/_8$-in-thick steel plate with different hole configurations is attached to the bottom of the chamber. A sliding plate containing a submerged test block is mounted on the frame at a controlled distance from the jet plate. The test block is made of a fine silver piece of known size (1 in. sq, $^1/_4$ in. thick)

Nomenclature

A = area

A_f = open area of jets, ratio of total orifice cross-sectional area to heat-transfer area, percent

C_p = specific heat of air

C_{ps} = specific heat of silver probe

D = diameter of hole

h = heat-transfer coefficient

$\bar{h}$ = average heat-transfer coefficient

h_0 = heat-transfer coefficient at zero interference

h' = surface coefficient of heat transfer for free convection of an enclosed space

I = the amount of interference

K = unit conversion factor

k_p = thermal conductivity of probe material

L = maximum distance along the heat-transfer surface between a point of impingement and the spent air removal edge of the impingement system

l_p = distance between two thermocouple wires in probe

l_s = thickness of silver probe

m, n = experimentally determined exponents

P = air power per unit area of heat-transfer surface

p_s = static pressure

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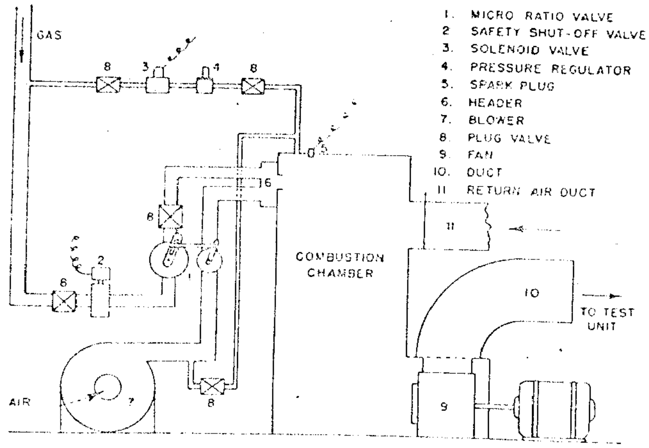

Fig. 1 General layout of the apparatus

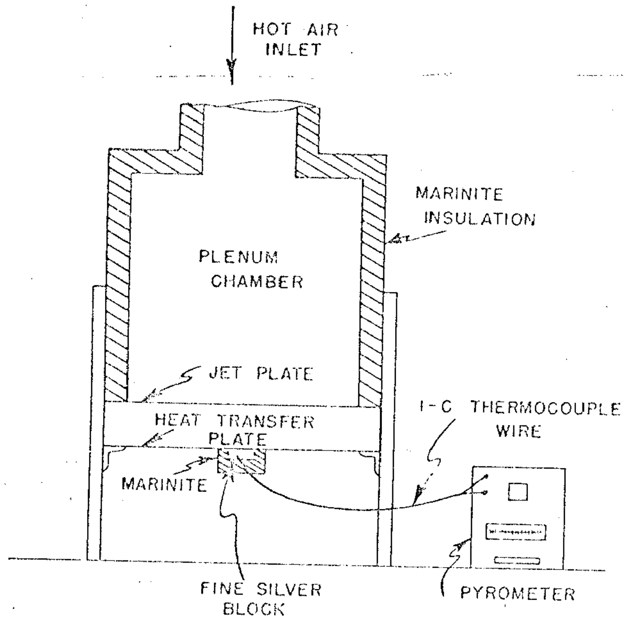

Fig. 2 Test unit for single jet

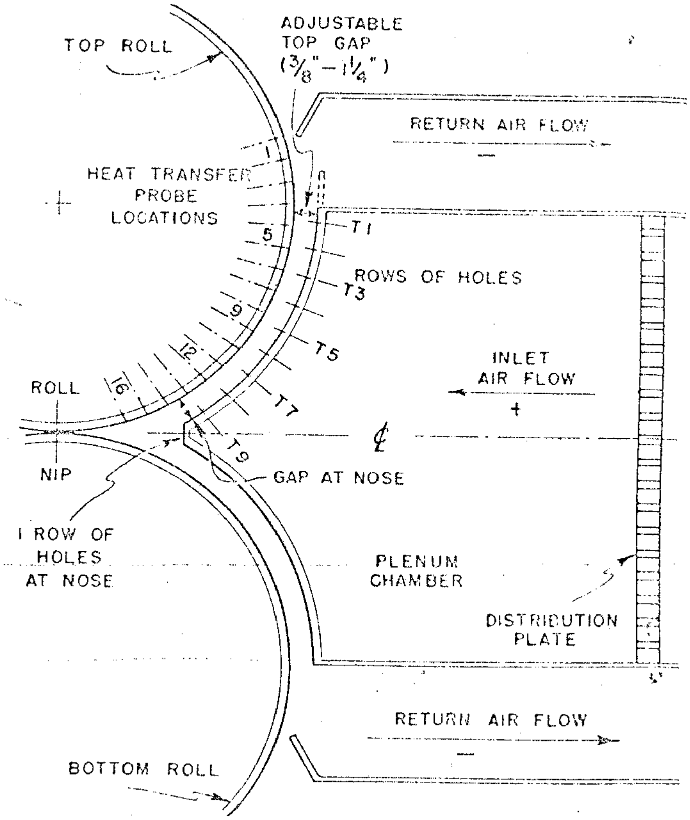

Fig. 3 Test unit for multiple jets

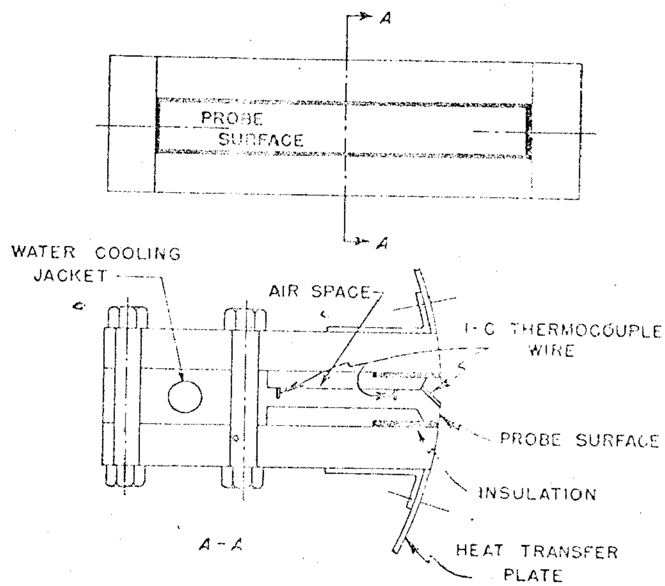

Fig. 4 Heat-transfer probe for multiple jets

and mass and submerged in a marinite insulator (3 in. × 3 in. × 2 in.). The top of the silver block is flush with the insulator and the heat-transfer plate to form a flat surface for the impinging air. The temperature of the test block is determined by means of a thermocouple wire embedded in the silver block. The impact velocities of the air on the surface are determined by means of a separate sliding plate and air pressure probe system.[2]

[2] Pressure probe consists of a pitot tube and a manometer. Impact velocity is derived from $U_c = \sqrt{\frac{2(p_t - p_s)}{\rho}}$ where p_t and p_s are measured values.

Nomenclature

- p_t = total or stagnation pressure
- Δp = velocity pressure, $p_t - p_s$
- Q = heat flux
- S = spacing between jet plate and heat-transfer surface
- T_A = air temperature
- T_s = probe surface temperature
- T_1 = initial temperature at $\theta = 0$
- T_2 = final temperature at $\theta = \theta$
- $T_3 - T_4$ = temperature drop in probe
- $\Delta T_1 = T_A - T_1$
- $\Delta T_2 = T_A - T_2$
- ΔT_m = logarithmic mean temperature differences, $\dfrac{\Delta T_1 - \Delta T_2}{\ln \dfrac{\Delta T_1}{\Delta T_2}}$
- U = air velocity
- U_c = impact velocity
- X = distance along the heat-transfer surface from the center line of round jet
- Y = distance between the center lines of two neighboring rows of holes
- Nu = Nusselt number, Dh/k
- Re = Reynolds number, $DU_c\rho/\mu$
- Pr = Prandtl number, $C_p\mu/k$
- δ = boundary-layer thickness
- θ = time
- μ = viscosity of air
- ρ = density of air
- ρ_s = density of silver probe
- τ = the height of roughness elements or the depth of knurl
- α = coefficient

2 Tests of Multiple Jets. The apparatus as shown in Fig. 3 consists of a plenum chamber with a distribution plate to regulate air flow for insuring uniform input air velocity to the jets. In front of the chamber is the curved jet plate which simulates segments of a circular roll surface. The hot air is introduced to the system from the unit as shown in Fig. 1.

A heat-transfer probe[3] (Fig. 4) is submerged into and flush with the surface of the dummy roll which simulates a cylindrical heat-transfer surface.

An adjustable spacer is used to vary the distance between the roll surface and the jet plate. The temperature of the heat-transfer surface is measured by means of thermocouple wires imbedded in the heat-transfer probe. The impact velocities at different probe positions are determined by a separate dummy roll and air pressure probe system.

Test Procedures

1 Single Jet. A jet plate with a particular hole configuration is installed for testing. By setting up a fixed distance between the jet plate and the heat-transfer plate, the following data are taken:

(*a*) Air temperature
(*b*) Initial temperature at zero time
(*c*) Final temperature at certain time intervals
(*d*) Time intervals

Knowing thickness, density, and specific heat of the silver test block in addition to the data obtained as listed above, the heat-transfer coefficient (h) can be calculated from the following equation for each test.

$$h = \frac{\rho_s l_s C_{ps}}{\theta} \ln \frac{\Delta T_1}{\Delta T_2} \qquad (1)^4$$

2 Simple Multiple Jets. The test procedure used is the same as for the single jet. However, the design of the jet plate is different and the test probe is submerged in the heat-transfer plate in such a way that it can travel a longer distance along the surface.

3 Multiple Jets. The heat-transfer coefficient is determined by the following steps for each test:

(*a*) Stabilize air temperature at about 350 deg F.
(*b*) Set the distance between the jet plate and the heat-transfer surface as desired.
(*c*) Position the probe axial to the jets.
(*d*) Stabilize the probe surface temperature.
(*e*) Obtain data on temperatures.

Knowing the thermal conductivity of the probe material and the distance between two thermocouples in the probe, based on steady-state conduction, the heat-transfer coefficient can be calculated from

$$h = \frac{k_p}{l_p}\left(\frac{T_3 - T_4}{T_A - T_s}\right) \qquad (2)^5$$

A separate set of air pressure probes is used to measure total air flow pressure and static pressure.

[3] The design of a heat-transfer probe is based on the principle of conduction heat transfer, the free convection effect being considered negligible.

[4] Equation (1) is derived from $Q = \frac{A l_s \rho_s C_{ps}}{\theta}(T_2 - T_1) = hA\Delta T_m$, where the following conditions are satisfied:

1 The gradient in the block is small or Biot number $(hl/k) < 0.1$.
2 Heat losses from block to insulation during transient are negligible.
3 The ΔT_{wall} is small as compared to $T_{air} - T_{wall}$.
4 Mach number < 0.3.

[5] Equation (2) is derived from the general conduction heat-transfer equations $Q = hA(T_A - T_s) = k_p A/l_p(T_3 - T_4)$. Total heat losses are negligible because with Grashof number (based on air gap) below 2×10^3, natural convection is suppressed and h' is dependent on air gap.

Heat Transfer Under a Single Jet

This phenomenon is clearly one of forced convection. Therefore, according to the classical forced convection heat-transfer approach, a group of dimensionless parameters such as Nusselt number, Reynolds number, and Prandtl number can be correlated in the following form:

$$\text{Nu} = \alpha(\text{Re})^m(\text{Pr})^n \qquad (3)$$

Where n is usually assumed to be $^1/_3$.

It has been noticed that the existence of turbulent air flow in our experiments is due to the high turbulence in the jet and the roughness of heat-transfer surface. According to Liepmann's [6] observation, surface roughness does produce immediate transition from laminar to turbulent flow when $\tau > 0.92\delta$ which holds true in our case. Vickers [1] estimated that the critical Reynolds number at which a round jet becomes turbulent is in the range of 1000 to 2000. Since, in these experiments, the impact velocity of air jets is between 100 and 300 fps, Reynolds number 10^3–10^4 and air temperature 300 to 350 deg F, the jets are highly turbulent.

When comparing our measured values of heat-transfer coefficients with the calculated ones from Colburn's [7] equation of the turbulent boundary layer

$$\text{Nu} = 0.0292\,(\text{Re})^{4/5}\,(\text{Pr})^{1/3} \qquad (4)$$

the results show that the measured values are about 20 percent higher with Reynolds number as low as 10^3. This evidently indicates that vertical impinging flow, rather than parallel flow, can bring about higher heat transfer.

In this work we investigated axial ratios (S/D) from 1 to 12 with hole diameters $^1/_8$, $^5/_{32}$, $^3/_{16}$, $^7/_{32}$, and $^1/_4$ in. In order to express the dependence of heat-transfer coefficient on S/D, we equated $f(S/D)$ to α in equation (3) or

$$\text{Nu} = (\text{Re})^m\,(\text{Pr})^n\, f\,(S/D) \qquad (5)$$

Using the experimental data as shown in Fig. 5, we obtain the value $m = 0.87$ when $n = {}^1/_3$.

Fig. 6 shows the variation of local heat-transfer coefficient under the center line of a round jet. The variations of S/D to the average heat-transfer coefficient over a distance ratio X/D from 0 to 20 are shown in Fig. 7. These data indicate hardly any change in heat-transfer coefficient when the S/D ratio is less than 6.

It is observed that the empirical equations for a single jet impinging on a flat heat-transfer surface can be written as

$$\text{Nu} = 0.0233\,\text{Re}^{0.87}\,\text{Pr}^{0.33} \qquad (6)$$

for local heat-transfer coefficient, and

$$\text{Nu} = 0.0180\,\text{Re}^{0.87}\,\text{Pr}^{0.33} \qquad (7)$$

for average heat-transfer coefficient.

The above equations show that under the same air impingement system, the local heat-transfer coefficient is about 25 percent higher than the average heat-transfer coefficient over a distance ratio (X/D) 0 to 20 and an S/D ratio 1 to 10. These limitations are well within the range of interest for air impingement heating or cooling work.

In order to arrive at a more general equation, we obtain α from Fig. 5, resulting in the following form

$$\text{Nu} = 0.0220\,\text{Re}^{0.87}\,\text{Pr}^{0.33} \qquad (8)$$

When compared with equation (4) and with Kern's [8] empirical equation of fully developed pipe flow for gas oil heating,

$$\text{Nu} = 0.0115\,\text{Re}^{0.9}\,\text{Pr}^{0.33} \qquad (9)$$

the heat-transfer coefficients derived from equation (8) are 18 to 35 percent higher than those derived from (4) or (9) with Reynolds number 10^3 to 10^4.

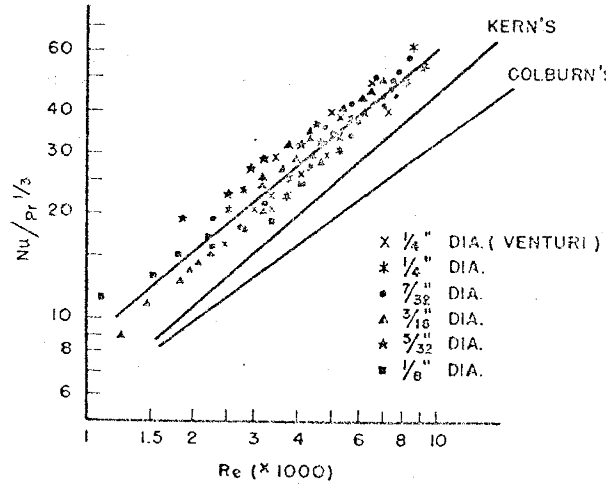

Fig. 5 Correlation of $Nu/Pr^{1/3}$ and Re

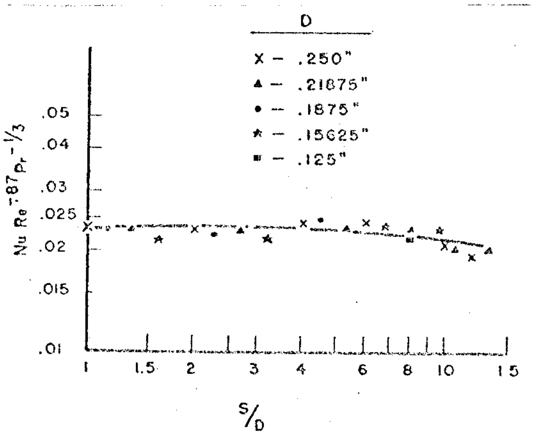

Fig. 6 The variation of local heat-transfer coefficient under the center line of round jet with axial distance ratio S/D

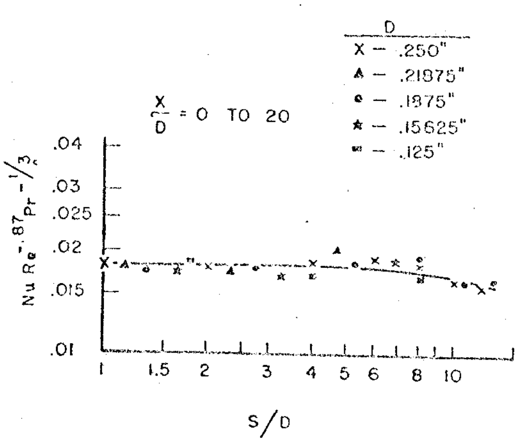

Fig. 7 The variation of average heat-transfer coefficient under a round et with axial distance ratio S/D

The difference between equation (4) and equations (6), (7), (8) lies in the definition of characteristic size. The advantage of using hole size instead of the distance along the heat-transfer surface from the centerline of jet as in equation (4) is twofold. Unlike the case of a flat plate immersed in a parallel stream of uniform flow, we do not have a free stream at an appreciable distance above the plate surface. Therefore the heat transfer is governed by the turbulence generated during the emergence of the jet from the orifice. The heat transfer is affected by the pressure drop $(p_t - p_s)$ which gives a high velocity to thin out the boundary layer upon impinging on the plate. The velocity is proportional to pressure drop across the jet plate and the mass flow rate depends on the hole size.

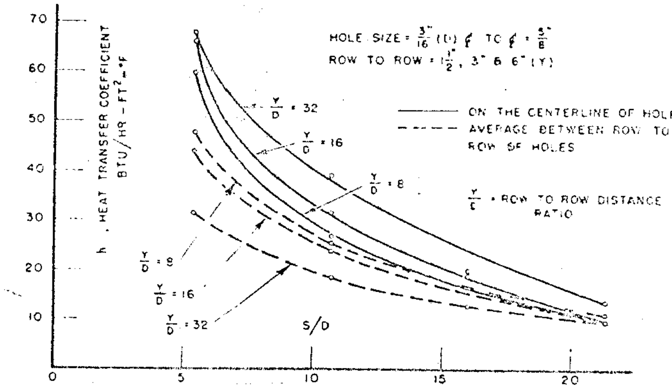

Fig. 8 The variation of heat-transfer coefficient under simple multiple jets with axial distance ratio S/D

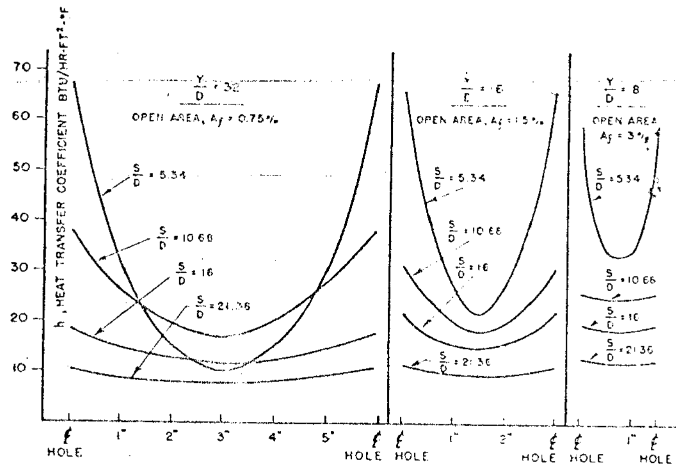

Fig. 9 The heat-transfer coefficient distribution between row to row of holes; 0.1875-in-dia hole, air velocity at the hole = 150 ft/sec; air temp = 300 deg F

Heat Transfer Under Multiple Jets

1 Simple Multiple Jets. Based on our test results of single jet experiments, we found that heat transfer reached its highest with a $^3/_{16}$-in-dia hole and an X/D ratio less than 3. Therefore, in designing simple multiple jet plates, we decided on using $^3/_{16}$-in-dia holes with a $^5/_8$ in. center-to-center distance. The plates were made of 12 in. × 12 in. × $^1/_8$ in. steel sheet with different row to row distances, namely, $1^1/_2$, 3, and 6 in. because of the fact that for simple multiple jet system, 0.75 to 3 percent open area constitutes the most logical range in practical application. The variation of heat-transfer coefficient with the S/D ratio was correlated as shown in Fig. 8. The data indicate that when S/D ratio becomes larger than 20, all curves will eventually meet at one point where the heat-transfer coefficient is identical in all cases. However, at S/D ratio below 20, the local heat-transfer coefficient on the center line of hole is directly proportional to the Y/D ratio, whereas the average heat-transfer coefficient between rows of holes is inversely proportional to the Y/D ratio.

In the engineering design of simple multiple jet systems, the holes are sparsely distributed and S/D ratio is usually large and, consequently, interference among jets can be neglected. Moreover, the uniform distribution of heat-transfer coefficient, rather than its magnitude, is of greater importance in this system. In order to obtain heat-transfer rates of best uniformity and satisfactory magnitude with least power consumption, the selection of proper hole configuration is crucial. Fig. 9 shows that in all cases the heat-transfer coefficient curves nearly become straight at an

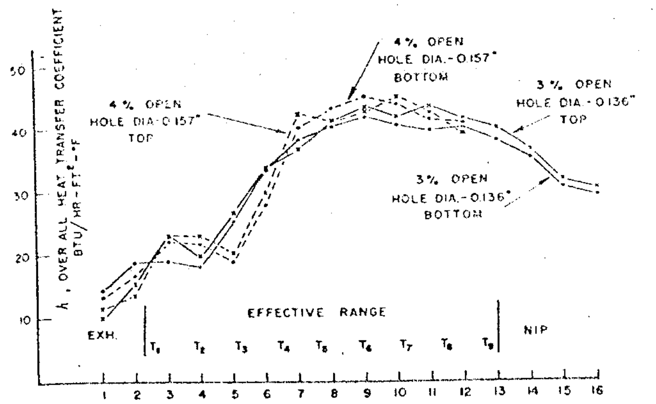

Fig. 10 Heat-transfer coefficient distribution curves; top and bottom sections

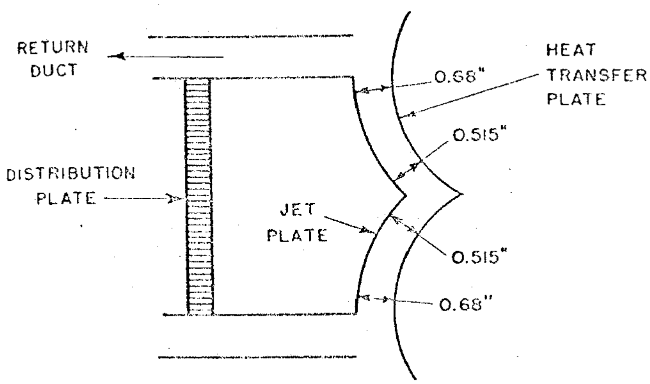

Fig. 10(a)

S/D ratio equal to 21.36. With 0.75 percent open area, the average heat transfer at this S/D region is about 9 Btu/hr-ft²-deg F as compared to 11.5 Btu/hr-ft²-deg F with 3 percent open area, yet in the latter case, the volume flow rate is four times greater.

The power consumption is proportional to the product of volume flow rate and pressure drop through the orifice. This pressure drop, in turn, is proportional to U_c^2. Thus, for round jets, we have

$$P = KU_c^3A_f \tag{10}$$

The above equation interprets the effect of open area A_f on unit power consumption.

2 *Multiple Jets.* In most cases, high magnitude of heat-transfer coefficient is as equally important as its uniform distribution over the entire heating surface. Further experimental work on multiple jets was carried out on rolls which were designed to simulate the actual heat-transfer conditions on a rotating cylinder. Test data were taken on both top and bottom rolls (Fig. 3). No significant difference in overall heat-transfer coefficients was observed under the same conditions as shown in Fig. 10. Later tests were made on the top roll only. The gap (S) between the jet plate and the heat-transfer surface and open area (A_f) were the principal variables in these tests. Fig. 11 shows the heat-transfer distribution curves with a fixed open area (4 percent) and various gap settings, and Fig. 12 shows the curves with a fixed gap setting ($^1/_2$ in.) and varied open area. Both figures indicate the presence of interference among the jets. The heat-transfer coefficient drops rapidly toward the edge, whereas its value remains high in the midst of the plate or near the nip. The trend clearly demonstrates how the impingement flow of one jet is cut by the spent air flow from the neighboring jet when both are active as jets. The impinging flow from the jet at the edge of the plate suffers most as a result of the accumulation of spent air exhaust flow from all other jets. According to [5], the amount of interference I is governed by the following equation:

$$I = A_fL/S \tag{11}$$

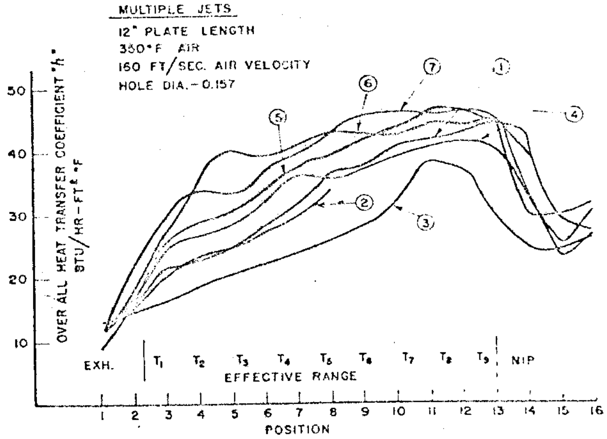

Fig. 11 $\bar{h}$ average versus rows of holes

NO.	A	B	% OPEN AREA
1	3/4"	1/2"	4
2	"	"	4 1" FENCE IN RETURN DUCT
3	1"	"	4
4	7/8"	"	4
5	1/2"	"	4
6	1/2"	"	4 5% FOR FIRST TWO ROWS
7	3/8"	"	4 "

Fig. 11(a)

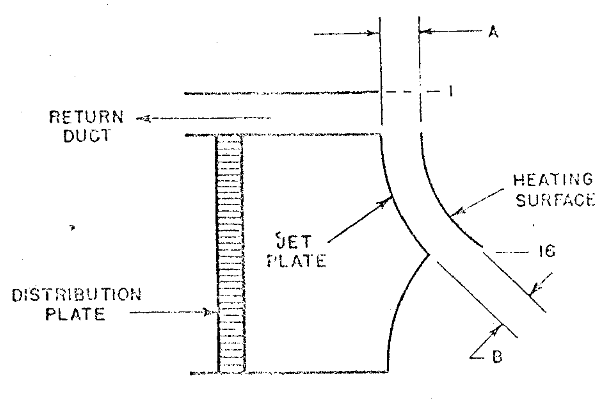

Fig. 11(b)

In order to make I small enough so that there will be absolutely no interference, it is necessary either to reduce A_f and L to the minimum, or to increase S to the maximum. However, it is not feasible in practice to reduce A_f and L below certain limits. On the other hand, an increase in S decreases the impact velocity and reduces the Reynolds number at the stagnation point on the heat-transfer surface which brings about a lowering of heat-transfer coefficient. The purpose of this experiment is to combine A_f, L, and S in such a manner that a uniform distribution of heat-transfer coefficient over the heat-transfer surface together with a high magnitude can be achieved.

Further correlation (Fig. 13) of the average heat-transfer coefficient with the S/D ratio at different hole diameters indicates that at S/D ratios between 2 and 4, the heat-transfer coefficients remain high for all hole sizes ($^1/_8$ to $^1/_4$ in.). When the S/D ratio is larger than 12, the same heat-transfer coefficient can be obtained regardless of hole sizes.

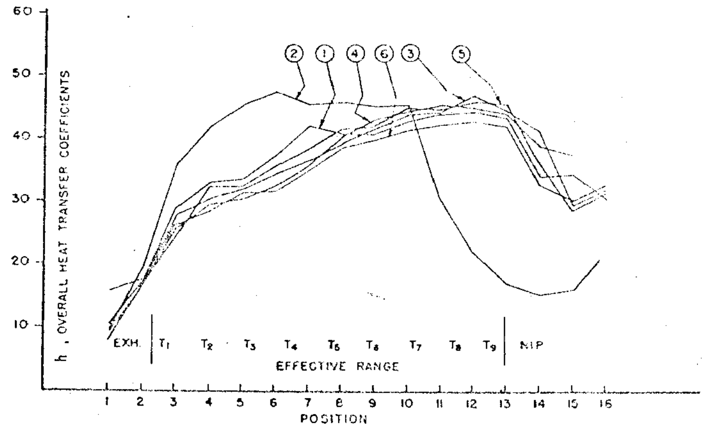

Fig. 12 The distribution of heat-transfer coefficients versus percent of open area; air temp 350 deg F; air velocity 160 ft/sec

CURVE NO.	% OPEN AREA
1	7 (T_1 & T_2) 6 (T_3) 5 (T_4) 4 ($T_5 T_6 T_7 T_8 T_9$)
2	SAME AS NO. 1 EXCEPT $T_7 T_8$ & T_9 ARE PLUGGED
3	7 (T_1 & T_2) 6 (T_3) 5 ($T_4 T_5 T_6 T_7 T_8 T_9$)
4	7 (T_1 & T_2) 6 ($T_3 T_4 T_5 T_6 T_7 T_8 T_9$)
5	7 (ALL ROWS)
6	8, 7 (ALL ROWS)

Fig. 12(*a*)

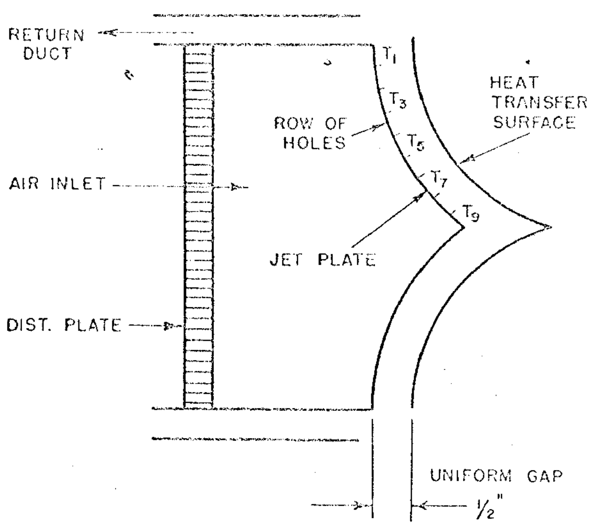

Fig. 12(*b*)

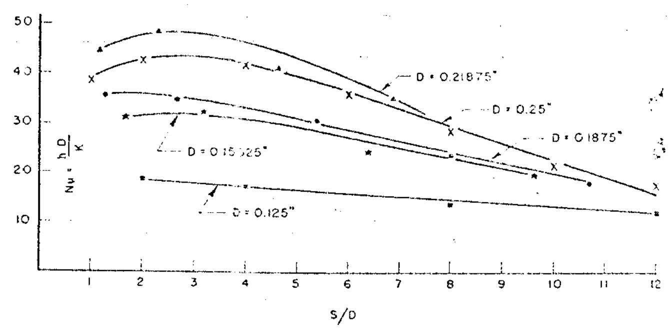

Fig. 13 Correlation of heat-transfer coefficients to spacing and dia ratio

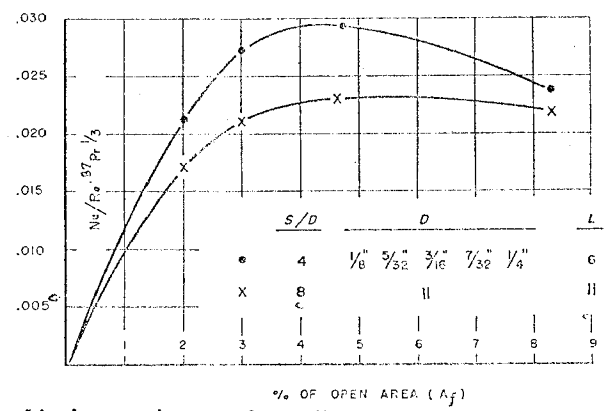

Fig. 14 Average heat-transfer coefficient under multiple jets versus total open area

Fig. 14 shows the effect of open area on heat-transfer rate. As A_f increases beyond 4 percent, the average heat-transfer coefficient reaches a maximum level and then declines. (D = from $^1/_8$ to $^1/_4$ in. S/D = 4 and 8.) However, at an S/D ratio equal to 4, the heat-transfer coefficient increases about 30 percent over that at an S/D ratio equal to 8.

Freidman and Mueller [3] correlated their test data of the effect of spacing and open area on the heat-transfer rate. Their findings were based on hole sizes from $^1/_4$ to $^5/_8$ in., and the results showed that the best type of air flow from perforated plates was the one with an open area (they termed free area) of 2 to 3 percent and spaced at a distance of 4 to 6 hole diameters from the heat-transfer surface. It is interesting to note from our work that with hole sizes from $^1/_8$ to $^1/_4$ in. at S/D ratio between 2 to 4 and an open area of 3 to 5 percent, the highest average heat-transfer coefficient can be obtained.

Conclusions

The heat transfer of high speed impingement jet flow in heating or cooling a normal heat-transfer surface is not yet thoroughly explored at the present time. The problem of single jet impingement has often been treated by applying the theoretical analysis of forced convection heat transfer. To define those dimensionless groups in equation (3) in terms of impact velocity and hole diameter used as characteristic size underlines the fundamental differences between our empirical equations (6), (7), and (8) and those of other researchers [4 and 7]. Our preference seems

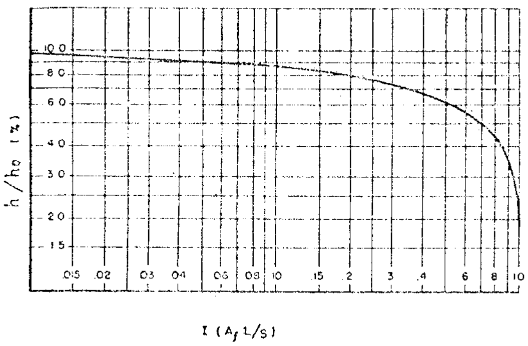

Fig. 15 Interference (I) versus the ratio of heat-transfer coefficients (h/h_0)

to be more appropriate in a system where the turbulence in the impinging stream is generated during its emergence from the orifice and where the velocity at the stagnation point under the jet is directly related to the effectiveness of impingement.

The plots illustrating the variations of local as well as average heat-transfer coefficients under a single round jet with an axial distance ratio S/D (Figs. 6 and 7) are in agreement with those of Daane and Han [5] for their two-dimensional jet. Their data for local heat-transfer coefficient at the same S/D ratios are about 20 percent lower than those of our round jets, and for average heat-transfer coefficient, ours are about 30 percent greater. However, in both cases heat-transfer coefficients remain constant up to $S/D = 6$, but start to decline when $S/D > 6$.

In multiple jet system, the flow pattern becomes much more complex. Neither existent experimental data nor theoretical treatment are available to help obtain a desired magnitude and uniformity of heat-transfer coefficient in engineering design. This work is an attempt to correlate empirical data in such a manner that practical application can be made possible. Furthermore, the interference among jets was actually measured (Fig. 15).

Two aspects of multiple jet systems have been studied. One concerns mainly the uniform distribution of heat-transfer coefficient and economy of power consumption. Our test results indicate that uniformity can be achieved at an S/D ratio larger than 21; and 0.75 percent open area is most economical in power consumption. The other concerns the feasibility of simulating a rotating cylinder in design. Our test results show that the best type of air flow through round jets impinging on a normal heat-transfer surface is at S/D ratio from 2 to 4 with an open area of 3 to 5 percent.

The data obtained from tests on the multiple jet system are most helpful in designing a cylindrical model of a rotating heat-transfer device. Given the heat-transfer rate required, the optimum S/D ratio and the percentage of open area can be determined. The distribution of heat-transfer coefficients over the entire surface can also be predicted.

Acknowledgments

The author wishes to express his gratitude to the following persons whose help made this paper possible:

Professor J. H. Potter, Stevens Institute of Technology; and Professor M. J. Levy, Newark College of Engineering, for their valuable comments and suggestions.

The Management of the United States Rubber Company for permission to publish this work and especially to Messrs. T. J. Rhodes and R. H. Hugger for their encouragement and advice.

Mr. D. N. Curtiss for his collaboration in all phases and in particular for his excellent design of the heat-transfer probe.

Messrs. A. S. Glowacki and D. W. Morgan of the Test Laboratory in the Mechanical Engineering Research Department, who were responsible for building the apparatus and taking the data; Miss E. G. Trumpoldt for her assistance in arranging and typing the manuscript.

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DISCUSSION

R. A. Daane[6]

Air impingement heat transfer is of considerable importance to the paper industry as well as others, and additional data leading to better understanding of such systems are welcome, even if only to confirm our own previous work. Mr. Huang's results, covering a wider range of variables than we covered, confirm our own findings particularly on heat transfer under a single jet and with respect to trends for multiple jet systems.[7]

However, there are large quantitative discrepancies between his heat-transfer values for multiple jet systems and those that we obtained. Mr. Huang's multiple jet systems used a different hole arrangement than we worked with. In our case, the holes were spaced in an equilateral triangular array, which we expected would give the most coverage with the least interference and thus would provide the maximum average heat-transfer coefficient for a given open area, hole size, S/D ratio, air temperature, and velocity. His holes are spaced in rows, with the row-to-row distance much larger than the distance between holes in each row. Contrary to our expectation, he shows higher heat-transfer coefficients for comparable conditions. This result is also contrary to his own data for a single jet. Comparing his Figs. 6 and 14, it appears that the average heat-transfer coefficient under a multiple jet system is higher than the peak heat-transfer coefficient directly under one of the same jets operating individually with no inhibiting effect of interference between jets.

In order to make sure that something very unexpected was not actually going on, we set up an experiment to determine local and average heat-transfer coefficients under a single row of holes using dimensions equal to those included in Mr. Huang's work. The results of this experiment fall about 40 percent lower than those shown in his Fig. 14. Also, in our previous work, we found that the peak heat transfer occurs with a lower value of open area. In this study, as well as in our previous investigations, we either worked with nozzles having a negligible jet contraction or based our computation of open area on the vena contracta diameter determined from measurements of orifice discharge coefficient. It appears that the open area values given in the subject paper are based on total hole area not considering the jet contraction which must have been present.

I think that there are a few points of interpretation which also warrant discussion. Mr. Huang's considerations of heat-transfer uniformity are not necessarily significant for heat transfer to a moving web of material. For example, jets on an equilateral triangular array can be set sufficiently askew to the direction

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[7] R. A. Daane and S. T. Han, "An Analysis of Air Impingement Drying," *Tappi*, vol. 44, no. 1, 1961, pp. 73–80.

of travel so that each point of the moving web experiences the same average heat-transfer rate while passing through the system, even though the local heat-transfer coefficient varies considerably over a fixed surface under such an array of jets.

According to the author, the best system is that which corresponds to the peak of a curve in Fig. 14. This does not consider the air blower power required. It can be shown that a system with less open area than that corresponding to the peak of a curve in Fig. 14 can achieve the same average heat-transfer coefficient with much less air per unit area using a somewhat higher velocity, but with less horsepower as compared with a system at the peak of the curve.

Darryl E. Metzger[8] and John J. Schauer[8]

This paper helps provide design criteria for the use of jet cooling and heating in modern applications. The transient technique used by the author to obtain heat-transfer coefficients is a convenient and inexpensive method for obtaining these coefficients, and it is particularly suited to the impinging jet problem. Some recent experimental work at Stanford University [9][9] utilized a similar transient method to obtain heat-transfer coefficients for both slot and circular jets impinging on flat surfaces. The results of this work indicate that the local heat-transfer coefficients depend not only on the ratio S/D, but also on the ratio X/D and the jet nozzle or orifice design.

These discussers feel that the importance of X/D, as an independent parameter affecting local heat-transfer values, should be clarified. Fig. 6 presents the local heat-transfer coefficient at the jet stagnation point; however, it is apparent from the description of the experimental apparatus that this is really an average heat-transfer coefficient in a region around the stagnation point. The size of the region depends upon the hole size. For example, since the silver target block was 1.0 in. square, the use of the 0.250 and 0.125-in. holes would result in X/D ratios at the block edge of 2.0 and 4.0, respectively. It should be pointed out that in the range $0<X/D<4$ the local heat-transfer coefficient varies as much as 50 percent, for $S/D<6$, as reported by Gardon and Cobonque (author's references [2]). This strong dependence of local heat-transfer coefficient on X/D probably explains the large scatter of the data of Fig. 5.

A comparison of the values presented in Fig. 7 with those of Fig. 6 indicates that the data points of Fig. 7 must have been obtained by averaging data taken at different X/D ratios out to $X/D = 20$. Since the author did not specify the method of obtaining the various X/D ratios, it is presumed by the discussers that these were obtained by moving the silver block out to various radial positions with respect to the stagnation point of the jet.

If the block was moved with respect to the stagnation point, then caution must be used in applying these results, for they may apply directly only to the particular heat-transfer conditions of the test. If the block surface and surrounding insulation surface are at the same temperature as the adjacent heat transfer plate, then the block will not affect the thermal boundary layer on the plate, and the measured heat-transfer coefficients will be those which would be obtained on the flat plate without the silver block. However, if the block surface is at a different temperature than the adjacent plate, then the thermal boundary layer on the block will depend not only on the position of the block on the plate, but also on the temperature difference between the block and adjacent plate. The local heat-transfer coefficients recorded for the silver block in these two cases may be quite different [10] and would not necessarily vary in the same manner with position. Unless the author has considered the effect of the block temperature on the measured local coefficients, his conclusions might not be entirely valid for the practical case of heating or cooling a more or less uniform temperature surface. For the tests with multiple jets with the heat-transfer probe it is explained in the paper that the probe has been moved to various positions with respect to the holes. Thus the temperature of the probe surface is important in the multiple jet tests as well as in the single jet tests.

The qualitative results for the jet interaction presented by the author should be of definite value in the design of multiple jet arrays from single jet data.

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Alfred H. Nissan[10]

This paper is very timely. There is a great deal of interest in such systems for the purpose of drying sheet materials like paper or textiles. Several times it happens that a multiple jet installation does not live up to expectations; almost invariably, subsequent investigations reveal the cause to be neglect of factors found important in the present paper by Huang and by a previous communication by Gordon and Cobonque (reference [2] in the paper). This fact makes it very important to understand these two papers and, perhaps of equal importance, to compare them. This brings me to my question.

Comparing the results of Huang with those of Gordon and Cobonque, one finds certain discrepancies. One must remember that the two sets of experiments were not identical: one laboratory used isothermal plates while the other adiabatic; one used fluid colder than and the other hotter than the plate; one used nozzles and the other orifices. Still, it is interesting that, whereas Huang finds a good correlation between the local values of Nu and the 0.87th power of Re, Gordon and Cobonque find good correlations between local Nu values and the half power of Re. Again, in the present paper, Fig. 7 shows no effect of X/D (varying from 0 to 20) or S/D (varying from 0 to 8) on the average heat-transfer coefficients, and Fig. 6 shows no effect of S/D variation up to $S/D = 6$, on local heat transfer. The detailed studies by Gordon and Cobonque, on the other hand, would lead one to expect quite a marked influence on these parameters for $X/d \leqslant 5$ or $S/D \leqslant 6$. Would the author care to comment and perhaps clear up these difficulties?

Similarly, comparisons by the author between his results and those of Gordon and Cobonque on multiple jets would be most instructive. For example, the use of two bases for the Reynolds number by the two laboratories is a little inconvenient for designers and research workers; one's own preference would be for the somewhat simpler approach of Huang. Perhaps the author would care to comment whether the use of higher linear speeds by Gordon and Cobonque necessitates the use of their basis for the Reynolds number or whether his relationships cover the entire range of their results.

Author's Closure

The author is thankful to the discussers for their interest in this work and for their valuable comments.

To Mr. Daane, the author is grateful for his criticism on the discrepancies in the experimental data. It is always a problem for experimenters to agree on the test results since each has his own test method and, moreover, the test conditions can hardly

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are identical. However, the fact that the local heat-transfer coefficient for a single jet (Fig. 6) is lower than the average heat-transfer coefficient under a multiple jet system (Fig. 14) probably results from using two different heat-transfer probes. The one for single jet is a fine silver block and the other for multiple jet system is a probe elaborately designed as shown in Fig. 4. Consequently, the equations for calculating the heat-transfer coefficients for these two systems are different, Equations (1) and (2). Any unintentional delay, and it is likely to occur, to record the time interval between initial and final temperatures by the experimenter will result in a more conservative heat-transfer coefficient for single jet system. The author had no intention to correlate the data of these two separate systems in his work.

Mr. Daane's effort "to make sure that something very unexpected was not actually going on" in Fig. 14 is appreciated. However, the test results in Fig. 14 have been verified by rerunning the tests. To our regret, Mr. Daane did not describe what kind of heat-transfer probe he used and how he had arrived at his data. Furthermore, the data in Fig. 14 were not obtained from using a certain jet plate with a single row of holes as in Mr. Daane's case. As a matter of fact, the multiple jet plate was designed with staggered holes of rhombus shape, the row to row distance being about 14 percent less than the centerline to centerline distance. Only the jet plate in our simple multiple jet system is similar to the one in Mr. Daane's tests. His data should probably be compared with those in Fig. 8 instead of Fig. 14. Obviously, the difference between simple multiple jet system and multiple jet system was not clearly understood.

The author does agree with Mr. Daane that when a moving web travels through the system, multiple jets spaced in an equilateral triangular array will give the best result. The hole arrangement in our multiple jet system was designed with this very notion in mind. From our tests, we find that there exists a lower limit of open area, below which the average heat-transfer coefficient drops drastically. Unless Mr. Daane can prove to the contrary, we believe that an open area from three to five percent will achieve the best overall results in multiple jet system.

The author agrees fully with Messrs. Metzger and Schauer on the importance of X/D ratio as an independent parameter to local heat-transfer coefficient. It is more appropriate, for practical reasons, to define the local heat-transfer coefficient as the heat-transfer rate of the whole impinging strength of a jet on a heat-transfer surface rather than that of the partial strength of a jet at the stagnation point. The impinging flow gains its full strength, not at the stagnation point, but at the point where the flow first comes in contact with the heat-transfer surface. Due to the sweeping effect, the boundary layer thickness is minimum at this point and is greatly increased when the sweeping strength is gradually weakened. Turbulence is generated from this impact and yet the laminar core remains at the centerline of the flow. The sweeping radius of a jet varies with the impinging velocity, S/D ratio, and the hole size. We have observed from our tests that the sweeping radius is about two to four hole diameters.

In single jet tests, the author was aware of the importance in maintaining the temperature of both the test block surface and its surrounding insulation surface the same as that of the adjacent heat-transfer plate, when the test block was moved with respect to the stagnation point. Same caution was taken for multiple jet tests.

The discrepancies Professor Nissan finds in this work and that of Gordon and Cobonque [2] in regard to the effect of X/D and S/D ratios on local and average heat-transfer coefficients may be explained by the fact that in order to present the true strength of a jet, our local heat-transfer coefficient was obtained by averaging its values in the region of X/D ratio from 0 to 2 or 4; whereas, in Gordon and Cobonque's case, it was taken at the stagnation point. Since our data are relatively more consistent, the X/D and S/D ratios at their lower range exert less influence on the heat-transfer coefficient in single jet system.

For multiple jet system, the choice of characteristic size accounts for the only marked difference in defining Reynolds number in the two sets of experiments. Gordon and Cobonque used the spacing between nozzles and we the hole size. The advantage of our preference has already been discussed in the paper. Since spacing between nozzles is always equal to or larger than hole size, the Reynolds number obtained by them is probably higher as compared with that derived by us. Our definition of impact velocity is equivalent to their interpretation of arrival velocity. Their mass flow rate in terms of air per unit area of nozzle array is comparable to ours in terms of air per unit cross-sectional area of hole combined with the use of percentage of open area as an independent variable. Therefore, if the effect of Reynolds number on geometrical variations (our holes and their nozzles) is negligible, our relationships should cover their results.