Impingement Heat Transfer of Reciprocating Jet Array*

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This paper describes an experimental study of impingement heat transfer of reciprocating jet-array with piston cooling application for marine heavy-duty diesel engine. A selection of heat transfer measurements illustrates the manner by which the individual and interactive influences of reciprocating force and buoyancy on heat transfer for the impinging jet-array. It is demonstrated that the reciprocating force coupled with buoyancy interaction causes considerable heat transfer modifications from the static results. The isolated reciprocating force effect could initially reduce heat transfer to a level about 0.45 of static level with weak reciprocation but recovers when the reciprocating force increases. Heat transfer improvement and impediment could be aided by the location-dependent buoyancy effect in addition to the reciprocating force effect. An empirical heat transfer correlation, which is physically consistent, has been developed to permit the evaluation of the individual and synergistic effects of reciprocating force and buoyancy interaction on local heat transfer of the impinging jet-array.

Key Words: Reciprocating Impingement Jet-array, Piston Cooling

1. Introduction

The marine heavy-duty diesel engine, characterized by low speed, long stroke and high power-to-weight ratio, directly drives the propulsive propeller in order to obviate the use of heavy-duty reduction gear. Its hot components constructing the combustion chamber require internal cooling schemes to maintain the material temperature below than the melting point at which level the mechanical properties are appropriate to sustain the mechanical and thermal loads so that the structure integrity of engine could be assured. The values of engine speed, maximum cycle pressure and temperature for this class of engine are, respec-

tively, in the range of 90 - 115 rev/min and about 125 bar and 1 700 K. Due to the high thermal and mechanical loads applied to a piston, which component is also subject to the material loss eroded by the gas swirls in the combustion chamber, the heat transfer augmentation for the cooling system in a piston has always been a design focus for engine development. Figure 1 shows the schematics of a conceptual design for the enhanced cooling networks in a piston. The coolant is

Fig. 1 Conceptual design of enhanced piston-cooling network

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channeled through the annulus in the piston rod into a pressure plenum underneath the piston crown. A high momentum coolant jet-array impinges onto the back surface of piston crown to facilitate impingement cooling. After impingement, the coolant is forced to flow in the spiral cooling passage, which stream cools the upper edge of piston crown and the piston skirt. The spent fluid is then directed toward the concentric passage in the piston rod that convects the residual heat out of the piston for the further usage aimed at improving the overall thermal efficiency of a merchandized ship. It is worth noting that the motion of coolant convected in the system depicted in Fig. 1 is simultaneously affected by the inertial force field created by reciprocating piston. The complex synergistic effects of convective inertia, reciprocating force and buoyancy interaction could interactively and considerably modify the flow and heat transfer from the static scenario. Based on the experimental evidences⁽¹⁾⁻⁽⁶⁾, the convective capabilities of various reciprocating flows could be initially reduced from the static duct references at weak levels of reciprocating force but tend to recover and be improved from the static references when the relative strength of reciprocating force further increases. Such reciprocating-force driven heat transfer modifications are usually accompanied with the considerable buoyancy interaction, which effect in isolation could either impede or improve heat transfer. As the piston motion could significantly affect the cooling performance of impinging jet-array, the confident design measure utilizing the impinging jet-array requires fundamental research efforts probing into the effects of reciprocating forces on heat transfer. This requirement motivates the present study and it is the combined effect of the heat transfer mechanisms associated with an impinging jet-array and the reciprocating motion that the present investigation is concerned.

Neither the flow configurations nor the heat transfer characteristics of a reciprocating impinging jet-array have been previously reported in the open literature. A considerable amount of studies has look into the static scenarios of impinging jet-array heat transfer. For design application, the impacts of stationary impinging jet-arrays on heat transfer are customarily described as functions of separation distance and jet Reynolds number for a set of geometrical specifications of impinging jet-array. Based on experimental results of⁽⁶⁾⁻⁽¹⁰⁾, Lee and Vafai⁽¹¹⁾ developed a procedure to determine the optimal separation distance and nozzle-to-nozzle spaces for an impinging jet-array. The maximum convective capability attainable with an impinging jet-array is significantly affected by the manner of treating the spent flow and

the induced crossflow between nozzle exit and targeting surface⁽⁶⁾. In general, heat transfer is deteriorated by the crossflow but might be enhanced due to the crossflow itself when separation distance is very small^{(9),(11),(12)-(14)}. This crossflow effect depends on the direction of exit spent-flow in a confined system⁽¹⁵⁾; but is minimized by locating the spent air exits through the openings in the impinging surface or in the nozzle plate⁽¹⁶⁾⁻⁽¹⁸⁾. Over the impinging surface, the networks along which the radial spent fluid is convected form the hexagonal and square webs, respectively, for the staggered and in-line jet-arrays⁽¹⁹⁾. The development of such radial network is a result of jet-to-jet interaction that affects the spatial heat transfer distributions considerably. When the jet Reynolds number and separation distance are respectively large and small, a ring of secondary stagnation-heat-transfer contour that surrounds the central confined jet emerges⁽¹⁹⁾. The values of the secondary stagnation heat transfer rate could be sometimes comparable with the stagnant point level⁽¹⁷⁾⁻⁽¹⁹⁾ and increase with the increase of Reynolds number but decrease with the increases of nozzle-to-nozzle spacing⁽⁷⁾ and separation distance⁽¹⁹⁾. This experimental observation typifies an influence of the boundary layer transition on heat transfer owing to the radial flow acceleration and deceleration, which mechanism is affected by the jet-to-jet interaction and the resulting pressure field over the impinging surface. Note, although the cooling scheme depicted in Fig. 1 has utilized the high convective capability of an impinging jet-array, the jet-array flow phenomena, such as the jet entrainment, jet-to-jet interference, cross-flow, wall-jet flow transition and the structures of turbulence and vorticity in the impinging jet flows are further complicated by reciprocation.

This study forms part of the on-going research program into the heat transfer augmentation of reciprocating flow with piston cooling application for marine heavy duty diesel engine. The present paper describes the results of a series of experiments aimed at studying the effect of reciprocation on heat transfer for a 3×3 impinging jet-array arranged in the in-line manner. The local Nusselt number distributions along the horizontal central line and along the diagonal of the square-shaped impinging surface with and without system reciprocation are presented over a range of buoyancy levels, Reynolds numbers and reciprocating frequencies. The impacts of reciprocation on heat transfer are analyzed to assist the derivation of empirical heat transfer correlations for piston cooling application.

Nomenclature

English symbols

- A, B : constant coefficients
 a : reciprocating amplitude (m)
 Cp_0 : specific heat of coolant at jet temperature ($\text{Jkg}^{-1}\text{K}^{-1}$)
 C_s, D_s : unknown functions
 d_j : diameter of nozzle in orifice plate (m)
 f_s : location dependent coefficients
 g : gravitational acceleration (ms^{-2})
 Gr : gravitational Grashof number $= g\beta_0\Delta T d_j^3/\nu_0^2$
 Gr_p : reciprocating Grashof number $= \omega^2 a \Delta T d_j^3/\nu_0^2$
 H : pitch of jets in array $= 4d_j$ (m)
 k_0 : thermal conductivity of coolant at jet temperature ($\text{Wm}^{-1}\text{K}^{-1}$)
 $\dot{m}$: total coolant mass flow rate into plenum chamber (kgs^{-1})
 Nu : reciprocating Nusselt number $= qd_j/(T_w - T_j)k_f$
 Nu_0 : static Nusselt number
 Pu : Pulsating number $= \omega d_j/W_m$
 Q : convective heat (W)
 q : convective heat flux (Wm^{-2})
 R : dimensionless radial coordinate (r/d_j)
 Re : Reynolds number $= \rho W_m d_j/\mu_0$
 S : separation distance $= 2d_j$ (m)
 T_j : jet temperature (K)
 T_w : wall temperature (K)
 W_m : mean through flow velocity (ms^{-1})
 X : dimensionless axial location (x/d_j)

Greek symbols

- β_0 : thermal expansion coefficient of coolant (K^{-1})
 ρ_0 : coolant density (kgm^{-3})
 μ_0 : dynamic viscosity of coolant ($\text{kgs}^{-1}\text{m}^{-1}$)
 ν_0 : kinematic viscosity of coolant (m^2s^{-1})
 ω : angular velocity of rotating disc creating reciprocating motion (rads^{-1})
 ΔT : reference temperature difference $= Q/Cp_0/\dot{m}$

Ψ, χ, ξ, ζ : unknown functions

Subscripts

- 0: static state

2. Experimental Details

2.1 Strategy

The primary tasks of the present study are to generate the impingement heat transfer data of a reciprocating jet-array and to examine the effect of reciprocating motion on heat transfer. The translating velocity of heat transfer assembly, motivated by the rigid slider-crank mechanism of which is simulat-

ing by the present reciprocating facility, is not a constant value but a skewed temporal sinusoidal function. A modified version of Navier-Stokes equations that considers the effect of reciprocation induced translating acceleration of the coordinate system on the fluid motion is usually convenient to describe the dynamic mechanism of the governing forces involved in a reciprocating system. The dimensionless equations derived from these modified flow momentum and energy equations, with the fluid motion referred to a coordinate frame which reciprocates with the flow boundary itself, unravel the governing non-dimensional flow parameters those control the convection of fluid momentum and energy⁽¹⁾. Two additional reciprocation related flow parameters; namely the pulsating (Pu) and reciprocating Grashof (Gr_p) numbers are identified to, respectively, quantify the ratio of pulsating to inertial forces and the relative strength of the reciprocating buoyancy effect⁽¹⁾. For a coolant under the condition that its temperature variation due to heat transfer did not make a considerable change of coolant's Prandtl number, the local Nusselt number (Nu) over the impinging surface of a reciprocating jet-array could be parametrically described by the following equation

$$Nu = \Psi\{Re, Pu, Gr, Gr_p, \text{Boundary conditions}\} \quad (1)$$

All the dimensionless groups appearing in Eq. (1) are defined in the nomenclature section. The study of the individual and combined effects of the non-dimensional groups appearing in Eq. (1) with the attempt to devise a physically consistent empirical heat transfer correlation is the strategic aim. The experimental data with zero buoyancy interaction ($Gr_p=0, Pu \neq 0$) are not feasible to be directly generated by the heat transfer experiments because a real fluid has finite value of thermal expansion coefficient. In such a limiting case having a vanishing small wall-to-fluid temperature difference but the thermal fluids system reciprocates, the solution of Ψ function in Eq. (1) recasts the combined Re and Pu effects on heat transfer without any buoyancy interaction. These zero buoyancy heat transfer data are inferred by extrapolating a family of heat transfer data taken at a specific Reynolds or pulsating number but with different wall-to-fluid temperature differences into the zero buoyancy asymptote. Note that, the buoyancy parameters in Eq. (1) involve the gravitational and reciprocating Grashof numbers, Gr and Gr_p , which feature two physically different buoyancy interactions. The influence of gravitational buoyancy normally develops in the steady flow field among which the differences in fluid temperature are stable and the differences in volumetric gravity force pro-

mote fluid motion. The buoyancy effect, motivated by the reciprocating acceleration, features the complex buoyancy interactions in a temporally periodical flow. It is felt that the buoyancy interactions in the present reciprocating system are originated from their effects on the vorticity. The temporary varied vorticity field affected by the reciprocating buoyancy could lead to a different mode of buoyancy interaction from the gravitation associated buoyancy effects. Thus the gravitational buoyancy is unlikely to exert considerable effect on flow and heat transfer inside a reciprocating flow system, as the flow and temperature fields are never steady. Therefore the gravitational and reciprocating Grashof numbers, Gr and Gr_p , are separately summarized in Eq.(1) that, respectively, dominates the buoyancy interactions in static and reciprocating flow conditions. It is very unlikely to simultaneously develop the gravitational and reciprocating buoyancy interactions.

For the set of boundary conditions simulated by the present heat transfer test module, such as the geometrical features of the jet-array and the heating condition, the effects of those constituent dimensionless groups indicated in Eq.(1) are examined by systematically varying each of the flow parameters involved in Eq.(1). A subsequent data reduction program follows the definitions of the dimensionless groups shown in Eq.(1), which generates the non-dimensional raw data for subsequent parametric analysis leading to the generation of empirical type heat transfer correlation for engineering applications.

2.2 Apparatus

The impinging jet-array test assembly was mounted on a reciprocating test facility, which has been previously described in Ref.(4). A brief description of the reciprocating test facility and a detailed illustration of the heat transfer test module now follow. Figure 2 shows the layout of the reciprocating test facilities and the airflow loop featuring the main components and instrumentation. The test coolant, dry air, was fed from tank (1). The coolant was pressurized by IWATA SC 175C screw-type compressor unit (2) after which the flow was dehumidified and cooled to the ambient temperature through a refrigerating system that was integrated with the compressor. After the tank (1), the test coolant was channeled into the heat transfer test module (3) through a dryer (4), a pressure regulator and filter (5), a pressure transducer and Tokyo Kesio TF-1120 mass flow meter (6) and a needle valve (7) for the control of mass flow rate. The nominal through flow Reynolds number was controlled by adjusting the mass flow rate measured at 150 diameters upstream of the test module (3). Because the coolant properties,

such as viscosity and density, varied with the local fluid temperature, the mass flow rate was adjusted to compensate the fluid-property variations in order to maintain the variations of Reynolds and pulsating numbers at the flow entrance within $\pm 1\%$ of the nominal values. Upon entering the plenum chamber of jet-array assembly in (3), the coolant flowed through a straight flow calming section (8) with an equivalent length of 20-tube diameter, in which the flow could be settled. System reciprocation is created by a crank-wheel-mechanism driven by a 2500 W DC motor (9). Through a reduction gearbox (10), the flywheel (11) could be controlled to rotate at the required speed measured by the optical pick-up (12). A counter balancing weight (13) is fixed on the rotating wheel (11) to maintain the dynamic balance during reciprocation. A slider track (14) is fitted underneath the platform (15) in order to support the heat transfer test module.

The constructional details of heat transfer test module are illustrated in Fig. 3. The test fluid was channeled through a flow calming tube (1) into a settling chamber of dimensions $50 \times 50 \times 80$ mm (2), and ejected from a 3×3 in-line array of nozzles through an orifice plate (3). The separation distance, S , and the jet diameter, d_j , of test module were 10 and 5 mm respectively. The thickness of orifice plate (3) was two jet diameters. Each nozzle had abrupt inlet and exit. The dimensionless jet-to-jet spacing, H/d_j , was 4. As the complete jet-array test module was mounted on the reciprocating test bed, the system reciprocation involved the reciprocations of impinging jet assembly and the flow calming tube (1), for which their reciprocating directions were in parallel with the

Fig. 2 Reciprocating test facility

Fig. 3 Heat transfer test module

main jet flow direction. The spent flow of the impinging jet was encapsulated within the Teflon made cubic-shaped assembly (4), which was eventually reversed toward the opened exit (5) after it impinged onto the thickness of 0.1 mm stainless steel heating foil (6). The heating foil (6) was connected with an adjustable high-current/low-voltage electrical power to generate a basically uniform heat flux heating condition. This 160×160 mm thin heating foil was squeezed between the four Teflon side walls (7) and a 30 mm thick Teflon plate (8) by 24 bolts for additional thermal insulation. The thermocouples measuring the temperatures of impinging surface were installed on the back of heating foil (6) as illustrated schematically in Fig. 3 where the X and R coordinate systems are defined. There were, respectively, 16 and 18 thermocouples welded along the X and R axes with equal intervals. Three additional thermocouples penetrated into the core of the flow path to measure the coolant temperatures at the flow calming tube (1), the plenum chamber of nozzle plate (2) and the discharged ports (5). Also depicted in

Fig. 3 are the constructional details of thermocouple arrangement. Each junction of K type thermocouple was produced by the spark-welding method. To measure the jet temperature, T_j , the thermocouple bead was guided in a 0.5 mm stainless tube that penetrated into the plenum chamber (2). For wall temperature measurement, the thermocouple junction made from two 0.1 mm thick thermocouple wires was sandwiched between the 0.05 mm thick Kapton insulation tape and the layer of ceramic cement. A small piece of Kapton tape was placed on the back of the 0.1 mm thick stainless steel heating foil (6) at location where the wall temperature measurement was made in order to assure the electrical insulation. The ceramic cement provided the essential adhesion to secure the thermocouple bead at the precise location. A Teflon plate over which the guiding grooves for thermocouple wires were machined squeezed the complete construction of wall thermocouple arrangement with the Teflon-made plenum chamber (2). This thermocouple arrangement was attempted to minimize the influences of wall conduction within the

heating element on the temperature measurement. All the temperature measurements were monitored and stored on a PC through a Net-Daq Fluke Hydra 2640A data logger for the subsequent data processing. Prior to the conduct of heat transfer test, all the thermocouples were calibrated in a constant temperature bath to ensure the measurement accuracy of ± 0.1 K. The local coolant properties such as β_0 , Cp_0 , k_0 , and μ_0 , which were used to evaluate the controlling non-dimensional parameters such as Gr , Gr_p , Pu and Re , were evaluated by means of standard polynomial functions with the reference jet temperature T_j measured at the fluid temperature in the plenum chamber (2) as the determined variable. Having acquired the coolant temperatures in the flow calming tube (1) and at the discharged ports (5), a routine procedure was subsequently performed to evaluate the fluid temperature rise through the complete set of jet-array assembly using the net amount of convective heat and the total coolant mass flow rate. Each calculated fluid temperature rise was compared with the actual measurement to check the accuracy of energy accountancy. The data batch was only collected when the difference between the calculated and measured fluid temperature rises was less than $\pm 10\%$.

2.3 Program

The program has three phases. Initially the static flow conditions are examined to generate the database to which the reciprocating results obtained at the second phase could be compared to assess the reciprocating effects on heat transfer. In these two phases, the local heat transfer measurements along X and R axes with and without system reciprocation are initially illustrated. It is followed by a parametric study of the coupled and individual effects of reciprocating force with and without buoyancy interaction on heat transfer aimed at revealing the impingement heat transfer of the reciprocating jet-array. In the third phase, a methodology is proposed to devise the physically consistent heat transfer correlation. The Nusselt number was determined experimentally using Eq. (2).

$$Nu = \frac{qd_j}{k_0(T_w - T_j)} \quad (2)$$

The local heat flux to the fluid at a given measurement spot, q , was determined from the heat flux generated in the heating foil with a correction to account for external heat loss. It was found that the local external heat loss was proportional to the locally prevailing wall-to-ambient temperature difference, with the proportionality constants taken from the static and reciprocating heat loss calibration plots. In actual fact, because the wall temperature distributions varied with the different flow conditions tested, the

locally prevailing wall-to-ambient temperature differences changed with flow conditions, which effects resulted in different heat losses. However, due to the good thermal insulation of the test module, the experimentally processed heat flux distribution based on the wall-temperature measurements detected by the present allocations of thermocouples over the impinging surface was controlled with maximum variation range of $\pm 5.3\%$, which allowed the approximation of basic uniform heat flux boundary condition. The reference temperatures T_w and T_j in Eq. (2) were obtained by actual thermocouple measurements. The characteristic length selected to define the Nusselt, Reynolds, pulsating and Grashof numbers was unified as the jet diameter of array, d_j , with the Reynolds number to be specified as $\frac{4}{9\pi} \frac{\dot{m}}{d_j \mu_0}$.

At each selected Reynolds/pulsating number test, five different levels of heating power, which respectively heated the wall temperature at the central stagnation point of impinging jet-array to the levels of 308 K, 313 K, 318 K, 323 K and 328 K. Note, due to the cubic encapsulation of heat transfer test assembly, the wall temperatures around the edge of heating foil were in the range of 398 – 468 K at the highest heating levels tested. Nevertheless each set of data generated for any combination of Reynolds and pulsating numbers reflected the influences of buoyancy on heat transfer when the heating power or the buoyancy level was systematically increased. The reciprocating tests at each Reynolds number tested were performed at four different frequencies, namely 0.83, 1.25, 1.67 and 2 Hz. Due to the reciprocating force effects, the flow structures, heat transfer rates and local wall temperatures were expected to be temporal functions. For each individual set of tests, the flow and heating level were fixed for about 30 minutes in order to assure that the flow reached an equilibrium state. When the differences between the successive time-averaged wall temperature measurements remained within ± 0.5 K, the temporal-repeated-state was assumed and the on-line data acquisition system was activated to store the instantaneous data for a period of 10 seconds. These raw experimental data were subsequently processed to generate the appropriate Reynolds, pulsating and Grashof numbers; and the individual effect of these parameters on the Nusselt number distribution examined. The range of these non-dimensional parameters is given in Table 1. It is worth noting that the relative small pulsating and reciprocating Grashof numbers were due to the selection of the nozzle exit velocity and the nozzle diameter as the characteristic velocity and length respectively.

An uncertainty approximation of the data reduc-

Table 1 Range of experimental non-dimensional parameters

Non-dimensional parameter	Range
Reynolds number [Re]	4700 – 14700
Pulsating number [Pu]	0.001103 – 0.0067
Gravitational Grashof number [Gr]	125 – 8354
Pulsating Grashof number [Gr_p]	0 – 2619
Buoyancy parameter [$\beta_0(T_w - T_j)$]	0.00895 – 0.45868

tion was conducted⁽²⁰⁾. Because the temporal-repeat-state of the reciprocating flow system was approximated when the variations of local time-averaged wall temperatures were in the range of ± 0.5 K, the maximum uncertainty in temperature measurement was estimated to be ± 0.5 K. These temperature measurements were the major sources to attribute uncertainty. With the temperature difference between the wall and fluid varied from 25 to 75 K, the maximum uncertainty for the Nu , Re , Pu , Gr and Gr_p were about 15.2%, 5.6%, 2.1%, 4.8% and 5.1%, respectively.

3. Results and Discussion

3.1 Static jet-array

The axial and radial distributions of wall temperature obtained at nominal Reynolds number of 13 300 with five heating powers, along with the corresponding fluid reference temperature, T_j , are respectively depicted in Figs. 4-a and 4-b. As shown in Figs. 4-a and 4-b, the increase of heat flux has incurred an attendant increase in wall temperature with enlarged data-spreads in both axial and radial directions. The regionally low wall-temperatures consistently occurred at the stagnation points while the hot spots developed in the regions adjacent to the inner edge of the cubic encapsulation constructed by the plenum chamber. An additional characteristic distribution pattern with which the overall wall temperature levels along both axial and radial axes were elevated in the streamwise direction, which reflected the influences of increased streamwise fluid temperature on wall temperature distributions over a surface with uniform heat flux. These wall temperature distributions revealed in Figs. 4-a and 4-b for static impingement jet array were typical for all the Reynolds and Grashof numbers examined that reflected the effects of local heat transfer variation on wall temperatures. The typical Nusselt number distributions along X and R axes with a nominal Grashof number of 1500 at Reynolds numbers of 4700, 7900, 11200, 13300 and 14500 are shown in Figs. 4-c and 4-d respectively. Local heat transfers are consistently improved when Reynolds number increases systematically. Three jet stagnation points of central and corner jets of array (see Fig. 4-c) and of middle jet in side column of array (see Fig. 4-d) are clearly distinguishable as the

Fig. 4 Axial and radial distributions of static wall temperature and Nusselt number

local maximum Nusselt numbers. Heat transfers decrease away from the stagnation point. The stagnant Nusselt number of central jet is lower than the middle jet in side column of array as shown in Fig. 4-c. Note, each jet in the array is subject to different degrees of confinement caused by jet-to-jet interactions and by the cubic encapsulation constructed by the present test assembly. With the separation distance of 2 jet diameters, the central jet-flow could be significantly confined by the surrounding jets and become submerged flow at very small separation distances^{(21),(22)}. As a result, the stagnant Nusselt number for the less confined middle jet in the side column of array is higher than the central jet value as shown in Fig. 4-c. Due to the confinements provided by the cubic encapsulation of test module, the stagnant Nusselt number of corner jet is consistently

lower than the middle-jet value in the side column of array but is almost as the same value of central jet as shown in Fig. 4-d. In the inter-jet regions between two stagnation points along X and R axes, as respectively depicted in Figs. 4-c and 4-d, the local minimum heat transfers are generated by the jet-to-jet interactions. In this respect, the worse heat transfer scenarios are observed in Fig. 4-c along X axis which reflects the stronger collision of radial spreading flows leading to weakened wall-jet flow momentum and thickened boundary layers. Away from the impinging jet regions, heat transfers decrease monotonically toward the inner edge of cubic encapsulation of test module among which the spent flows are reversed backward to the discharged ports.

The impact of buoyancy interaction on the impingement heat transfer of static jet-array is demonstrated in Fig. 5 where the heat-transfer variations along X and R axes with three ascending buoyancy levels at a fixed Reynolds number are compared. As shown in Fig. 5, the considerable downward data spreads, driven by the increase of Grashof number, appear in the jet stagnation areas where the considerable impeding buoyancy effect on heat transfer is unraveled. When the radial spent-fluids flows away from the jet stagnation areas toward the inner edges of cubic encapsulation, a tendency of data convergence gradually emerges, which could be treated as a weakened buoyancy effect on heat transfer. Thus the

gravitation-driven buoyancy effect appears to be the spatial function that could provide considerable heat transfer impediments in the jet stagnation areas of static jet-array. Because there is noticeable buoyancy effect but negligible change in the coolant Prandtl number covered by the experimental program, the local Nusselt number along the X and R axes of the static impinging jet-array at each measured station is attempted to be correlated by a combined parameter of $\frac{Re^2}{Gr}$ in the form of

$$Nu_0 = A(X, R) \times \left(\frac{Re^2}{Gr} \right)^{B(X, R)} \quad (3)$$

Coefficients A and B in Eq. (3) are functions of axial or radial location, X or R . A set of selective examples revealed in Fig. 6 demonstrates the manner of data convergence when the static Nusselt number is plotted against the combined parameter of $\frac{Re^2}{Gr}$.

Within a finite range of buoyancy levels, Eq. (3) approaches zero at the limiting case of zero Reynolds number. Table 2 summarizes the numerically curve-fitted A and B coefficients. As compared in Fig. 7, 90% of the present experimental static Nusselt numbers were correlated by Eq. (3) within $\pm 10\%$ over the entire range of Reynolds and Grashof numbers examined. Equation (3) is therefore treated as the reference heat transfer solution of static jet-array against which the heat transfer results obtained with

Fig. 5 Static Nusselt number distributions along axial and radial axes with various Grashof numbers at Reynolds numbers of 4700 and 14470

Table 2 Coefficients A and B in Eq. (3)

Axial location (X)	$A(X)$	$B(X)$	Radial location (R)	$A(R)$	$B(R)$
0	9.211	0.229	1.325	10.855	0.198
0.925	7.692	0.204	2.65	15.832	0.163
1.85	10.776	0.161	3.975	7.685	0.214
2.775	62.775	0.059	5.3	8.529	0.190
3.7	35.202	0.118	6.625	48.661	0.057
4.625	12.278	0.192	7.95	24.018	0.137
5.55	9.783	0.157	9.275	27.104	0.113
6.475	8.479	0.139	10.6	21.047	0.129
7.4	7.718	0.119	11.925	20.018	0.123
8.325	8.821	0.094	13.25	12.110	0.157
9.25	4.933	0.133	14.575	11.549	0.161
10.175	5.379	0.1296	15.9	8.868	0.174
11.1	7.563	0.111	17.225	6.982	0.184
12.025	9.729	0.082	18.55	7.347	0.171
12.95	10.966	0.046	19.875	5.161	0.177
13.875	8.534	0.049	21.2	11.359	0.091
			22.525	3.167	0.154

Fig. 6 Variations of static Nusselt number with $\frac{Re^2}{Gr}$ at various axial and radial locations

the reciprocating jet-array are compared to assess the effect of reciprocation on heat transfer.

3.2 Reciprocating jet-array

When the heat transfer test module reciprocates, the jet-flows, radial wall-flows and vortical flow cells developed between the orifice and target plates⁽²²⁾ are affected by the reciprocating forces. These temporally varied flow fields accompanying with the modified structures of turbulence and vorticity could trigger a dynamic process of time-wise variations in heat transfer for the present reciprocating jet-array. After taking the averaged value of the time-wise local Nusselt number data for a period of 10 seconds, the time-averaged heat transfer results along the X and R axes are acquired. Figure 8 compares the Nusselt number distributions along X and R axes obtained

Fig. 7 Comparison of experimental measurements with correlative results of static Nusselt numbers

with a nominal Reynolds number of 4700 at reciprocating frequencies of 0, 0.833, 1.25, 1.67, and 2 Hz, which illustrates the general reciprocating effects on heat transfer. Each plot shown in Fig. 8 contains three sets of heat transfer distributions obtained with three ascending buoyancy levels. In this regard, the increase of reciprocating Grashof number, Gr_p , at any $Re-Pu$ condition examined shown in Fig. 8 generates the upward data spreads in the stagnant jet area. In contrast to the impeding gravitational Gr effect on heat transfer as demonstrated in Fig. 5 for static jet-array, the reciprocating buoyancy effect improves heat transfer at locations under the jet-array. However, there is spatial dependency of reciprocating buoyancy effect on heat transfer, which shows the improved data convergence when the coolant flow travels away from the stagnant jets areas toward the inner edges of the cubic encapsulation. The impact of reciprocating buoyancy interaction on heat transfer is less effective in the regions where the spent fluid flow is forced to reverse toward the discharged ends. In this respect, increasing the buoyancy level impedes heat transfers in the region near the inner edges of encapsulation, although the buoyancy effect on heat transfer in this region is relatively weak. Further

Fig. 8 Variations of reciprocating Nusselt number along X and R axes at Reynolds number of 4700

discussion on this buoyancy complexity is provided in the section of parametric analysis.

The variations of heat transfer distributions along X and R axes caused by the systematic increase of pulsating number at Reynolds number of 4700 are revealed by comparing the heat transfer results

typified from Figs. 8-a to 8-d and from Figs. 8-e to 8-h respectively. As shown in Figs. 8-a, 8-b and 8-c and in Figs. 8-e, 8-f and 8-g, the overall heat transfer levels along X and R axes gradually decrease with the increase of pulsating number when the reciprocating frequency increases from 0.833 to 1.67 Hz. In this

particular range of pulsating numbers examined, the considerable heat transfer impediments from the static references develop in the area covered by the jet-array. A further increase of the reciprocating frequency from 1.67 Hz to 2 Hz, which increases the pulsating number from 0.0056 to 0.0067 as showed in Figs. 8-d and 8-h produces a subsequent heat transfer recovery along X and R axes from the results depicted in Figs. 8-c and 8-g. Thus, when the pulsating number gradually increases, the impingement heat transfer of the reciprocating jet-array is initially reduced. Increasing the pulsating number further then follows a subsequent heat transfer recovery. This observation will be examined in the more details when the experimental data are presented parametrically. It is also interesting to note that the variations of Pu and Gr_p could only provide moderate heat transfer modifications from the static scenarios in the regions where the flow is forced to reverse backward to the exit of heat transfer test assembly.

3.3 Parametric analysis

Figure 8 has demonstrated the various degrees of heat transfer modification from the static reference when the pulsating and reciprocating Grashof numbers vary at the fixed value of Reynolds number, thereby confirming the interactive effect of the pulsating and buoyancy forces on impingement heat transfer of reciprocating jet-array. The synergistic effects of reciprocating force and buoyancy interaction could

either increase or reduce heat transfer from the static condition. The reciprocation induced heat transfer modification could be indexed by the relative Nusselt number, Nu/Nu_0 , which is the ratio of reciprocating Nusselt number, Nu , to the static Nusselt number, Nu_0 , obtained at the same Reynolds number. To accomplish the strategic aim of acquiring the heat transfer correlation for engineering application, it is essential to identify the individual and coupling Re and Pu effects on heat transfer without buoyancy interaction, which results feature the limiting cases with $Pu \neq 0$ and $Gr_p = 0$ in Eq. (1). As a real fluid possesses a value of volume expansion coefficient, it is not strictly possible to conduct a heat transfer experiment at the zero buoyancy condition with finite values of wall-to-fluid temperature difference. The values of relative Nusselt number with zero buoyancy of Re - Pu effect may be inferred by extrapolating a family of heat transfer data obtained with different $\beta_0(T_w - T_j)$ values to the zero buoyancy asymptote at which $\beta_0(T_w - T_j)$ value remains zero for a specified set of Reynolds and pulsating numbers. The procedure of revealing the zero buoyancy asymptotic heat transfer data is demonstrated in Fig. 9 where the relative Nusselt number is plotted against $\beta_0(T_w - T_j)$ for a range of pulsating numbers at Reynolds numbers of 4700 and 11200 at several locations under the impinging jet-array area. Each individual curve at a selected Re - Pu combination is numerically extrapolated back

Fig. 9 Extrapolating heat transfer results into zero buoyancy level

to the origin to determine the zero buoyancy heat transfer data in terms of relative Nusselt number. The zero buoyancy asymptotic heat transfer data collected in each plot of Fig. 9 is reduced below than the static jet-array level at low values of pulsating number but tends to recover at high values of pulsating number. These Pu dependent zero-buoyancy heat transfer levels, indicating in each plot of Fig. 9, vary with Reynolds number and location over the impinging surface. Thus the Pu controlled heat transfer variation with vanished buoyancy interaction at any location over the impinging surface of reciprocating jet-array is Reynolds number dependent. Figure 9 therefore implicitly demonstrates the coupling Re - Pu effect on heat transfer. Also noticed that, after examining all the versions of Fig. 9 at different axial and radial locations, the increase of $\beta_0(T_w - T_j)$ value could correspondingly produce either the increment or impediment in relative Nusselt number at a fixed pulsating number depending on the location examined. The review of all the versions of Fig. 9 shows that the improving buoyancy effect on heat transfer generally occurs at the locations under the impinging jet-array; while the heat transfer impediment due to buoyancy interaction is developed at locations around the inner circumference of cubic encapsulation among which the spent fluids are reversed backward to the exit. As illustrated previously in Fig. 8 that the buoyancy effects on heat transfer are less effective than those experienced at locations under the jet-array, the rates of heat transfer variation caused by the increase of buoyancy level around the inner edges of encapsulation are less than those counterparts at the locations under the jet-array. It is evident that the heat transfer improvement and impediment could be aided by the location-dependent non-zero buoyancy effect in addition to the Re - Pu coupling effect.

A complete set of zero-buoyancy extrapolation heat transfer data is generated using a series of cross plots, based on Fig. 9, but applied to the entire ranges of Reynolds and pulsating numbers tested for all the locations detected. The result of examining the zero buoyancy asymptotic heat transfer data at Reynolds numbers of 4700 and 14700 for several locations along X and R axes are shown in Fig. 10. This manner of data presentation reveals the individual Pu effect on heat transfer at a specified Reynolds number. Note, the value of relative Nusselt number, Nu/Nu_0 , for static jet-array, which corresponds to the data points at $Pu=0$ in Fig. 10 has to be unity in order to recast the static jet-array solution. As showed in Fig. 10, the tendency for heat transfer to be initially impaired relative to the static reference that is followed by the subsequent recovery when pulsating number

increases is evident for all the Reynolds numbers and locations examined. Referring to the different data trends obtained with various Reynolds numbers for any selected location displayed in Figs. 10-a and 10-b or in Figs. 10-c and 10-d, the complete set of Pu controlled zero-buoyancy heat transfer data is found to be sensitive to the Reynolds number. Justified by all the versions of data trend typified in Fig. 10 at any specific pulsating number in the parametric range tested, the relative Nusselt number with vanished buoyancy effect could be reasonably well correlated by the equation of

$$\frac{Nu}{Nu_0} \Big|_{Gr_p=0} = 1 + \xi\{X, R, Re\} \times Pu + \zeta\{X, R, Re\} \times Pu^2 \quad (4)$$

where ξ and ζ functions in Eq. (4) are the numeri-

Fig. 10 Extrapolated variation of heat transfer with pulsating number at zero buoyancy

cally determined coefficients by interpolating a series of curves that correlates the values of ξ and ζ functions at all locations measured. The mathematical structure of Eq. (4) ensures that the relative Nusselt number remains as unity when pulsating number becomes zero, which recovers the static jet-array solution of Eq. (3). The reproduced correlating-curves using Eq. (4) shown in Fig. 10 clearly illustrate the manner of heat transfer modification generated by the isolated pulsating force effect without buoyancy interaction. However, as the convergence of each Pu controlled curve typified in Fig. 10 for different Reynolds numbers tested at any specific location is not achievable, it unravels the inter-correlative relation-

Fig. 11 Variations of ξ and ζ functions with Reynolds number

ship between Pu and Re parameters in Eq. (1). Figure 11 demonstrates the dependency of Pu effect on Reynolds number by revealing the variation manners of ξ and ζ functions with Reynolds number at several locations along X and R axes. A detailed examination of the data trends for all the location versions of Fig. 11 suggests that the values of ξ and ζ functions could be reasonably correlated by the second order polynomial functions of

$$\xi = C_0(X, R) + C_1(X, R) \times Re + C_2(X, R) \times Re^2 \quad (5)$$

$$\zeta = D_0(X, R) + D_1(X, R) \times Re + D_2(X, R) \times Re^2 \quad (6)$$

where C_s and D_s are determined from a regression analysis applied to each ξ and ζ functions respectively. Tables 3 and 4 list the numerically determined curve fits for the C_s and D_s coefficients for each of the location dependent ξ and ζ functions along X and R axes respectively.

The general influence of reciprocation on heat transfer of the present impinging jet-array shows strong coupling effect between Re and Pu controlled flow physics at zero-buoyancy condition. The interactive and combined force effects induced by reciprocation found in the parametric matrix examined by the present study could cause the relative Nusselt numbers to be reduced to the values about 55% of the static levels. By treating the reciprocating buoyancy interaction as a superimposed effect on heat transfer,

Table 3 Coefficients C_0 – C_2 and D_0 – D_2 along X axis

X location	$C_0(X)$	$C_1(X)$	$C_2(X)$	$D_0(X)$	$D_1(X)$	$D_2(X)$
0	-22150	13.85	-0.0007782	44.56	-0.08432	5.149E-6
0.925	-21510	10.38	-0.0005542	53.98	-0.02999	1.761E-6
1.85	-22145	25.95	-0.000134	74.45	-0.05277	2.3E-6
2.775	-1838	6.715	-0.0002753	68.97	-0.06198	3.426E-6
3.7	-106800	32.36	-0.001747	130.3	-0.07732	3.664E-6
4.625	-49420	14.58	-0.0005871	384.4	-0.1376	7.01E-6
5.55	-22740	6.679	-0.0001927	193.2	-0.06731	3.209E-6
6.475	-120700	25.99	-0.001014	844	-0.1989	9.785E-6
7.4	-125700	25.04	-0.0009	897.1	-0.1981	9.558E-6
8.325	-118900	25.63	-0.0009672	889.5	-0.2058	10.06E-6
9.25	-147600	29.5	-0.001014	1137	-0.2434	11.79E-6
10.175	91610	-12.99	0.000698	-306.8	0.01879	0.211E-6
11.1	49910	-0.4491	-0.0001027	-241	-0.008235	1.492E-6
12.025	101000	-13.13	0.000626	-476.9	0.04999	1.845E-6
12.95	130600	-18.48	0.0009396	-565.1	0.06134	-2.481E-6
13.875	-7904	8.157	-0.0003788	-83.61	-0.03807	1.941E-6

Table 4 Coefficients C_0 – C_2 and D_0 – D_2 along R axis

R location	$C_0(R)$	$C_1(R)$	$C_2(R)$	$D_0(R)$	$D_1(R)$	$D_2(R)$
1.325	-78350	20.23	-0.0008569	569.5	-0.1689	8.249E-6
2.65	-31790	-5.031	0.00503	-96.16	-0.1573	0.02821E-6
3.975	-131700	33.36	-0.001499	620.3	-0.1776	8.762E-6
5.3	-113600	28.66	-0.001269	688.4	-0.1959	9.588E-6
6.625	-5533	17.84	-0.0008044	287.1	-0.1234	6.066E-6
7.95	-101900	26.92	-0.001167	523.4	-0.1632	7.776E-6
9.275	-35670	13.49	-0.000515	84.69	-0.07549	3.399E-6
10.6	-24860	11.47	-0.0005286	78.05	-0.07292	3.58E-6
11.925	-53340	19.04	-0.0009327	113.9	-0.0849	4.172E-6
13.25	-81270	22.65	-0.0009915	81.96	-0.06151	2.792E-6
14.575	6728	3.621	-0.0001032	-26.31	-0.04521	2.178E-6
15.9	5420	3.227	-0.00004588	30.86	-0.05732	2.712E-6
17.225	12230	3.115	-0.0001078	-24.51	-0.05226	2.73E-6
18.55	26120	2.35	-0.0001028	-76.84	-0.05396	2.987E-6
19.875	14350	5.824	-0.0002374	-213	-0.02349	1.356E-6
21.2	12320	5.912	-0.0002053	168.2	-0.1195	6.486E-6
22.525	74280	-12.21	0.0007312	-160.2	-0.003597	0.1838E-6

a possible specific form for heat transfer correlation involving the buoyancy effect is assumed as

$$\frac{Nu}{Nu_0} = \{1 + \xi(X, R, Re) \times Pu + \zeta(X, R, Re) \times Pu^2\} + \Phi_{Bu} \quad (7)$$

where Φ_{Bu} quantifies the superimposed reciprocating buoyancy effect on heat transfer. A physical constraint for reciprocating zero-buoyancy solution in Eq. (7) requires that Φ_{Bu} function vanishes when Gr_p approaches zero. In this case, there is still a reciprocating force modified scenario with a consequential effect on heat transfer, which influence is quantified by Eq. (4). Because it has been demonstrated that the reciprocating buoyancy effect on heat transfer is Pu dependent as shown in Fig. 8, the buoyancy function, Φ_{Bu} , in Eq. (7) should involve Pu as a determined variable. Nevertheless, as the isolation of Re variable effect from function of Φ_{Bu} could considerably simplify the task of deriving Φ_{Bu} , an attempt was made by plotting Φ_{Bu} values obtained with the entire range of Reynolds numbers examined against Gr_p for each specific location along X and R axes. The convergence of all the data points of Φ_{Bu} , generating with various Reynolds numbers, toward a single Pu variable controlled curve could verify the weak dependency of Re variable from the reciprocating buoyancy effect on heat transfer. This is demonstrated in Fig. 12 where Φ_{Bu} functions obtained with all the Reynolds numbers tested are plotted against Gr_p for several axial and radial locations. All the

data points are collapsed into a tight data band with a linear-like varying tendency controlled by pulsating number. Each Pu controlled data trend for Φ_{Bu} functions, as revealed in Fig. 12, converges toward zero value as Gr_p approaches zero, which reflects the vanished buoyancy effect at zero Gr_p condition. Therefore, the influence of reciprocating buoyancy on heat transfer is Pu dependent but could be uncoupled from Re effect. A review of all location versions of Fig. 12 with the consideration of the physical constraint for vanished buoyancy effect at zero Gr_p value has led to a correlating equation for the superimposed buoyancy function Φ_{Bu} of the general form

$$\Phi_{Bu}(X, R) = \chi(Pu) \times Gr_p \quad (8)$$

The functional value of χ in Eq. (8) typifies the slope of each correlating line depicted in Fig. 12, which varies with location. The improving and impeding buoyancy effect on heat transfer is, respectively, demonstrated by the positive and negative values of χ function. The varying manners of these Pu -controlled correlating lines with Gr_p shown in Fig. 12 reveals that the buoyancy interactions at different locations could be reversed from improving to impeding heat transfer. Subsequent examination of all versions of Fig. 12 at different axial and radial locations confirms that such location-dependent adverse buoyancy effect could take place in the regions of $X \geq 6.475$ and $R \geq 21.2$. Figure 13 shows the variations of χ function with Pu at several X and R locations. Noteworthy the manner for the χ functions varies with Pu shown in

Fig. 12 Variations of buoyancy function with reciprocating Grashof number at various axial and radial locations

Fig. 13, which could be reversed at different axial locations. The locations under the jet-array generally exhibit the gradual reduction in χ functions from positive to negative values, while the χ function tends to be increased from negative to positive values at the locations along the edge of cubic encapsulation. Led by a detailed examination of all location versions of Fig. 13, it is proposed that χ can be reasonably well approximated by the equation of the following form

$$\chi = f_0(X, R) + f_1(X, R) \times Pu + f_2 \times Pu^2 \quad (9)$$

where the f_s coefficients are determined from the regression analysis applied to each χ function. Table 5 summarizes the correlative coefficients for f_s functions along the X and R axes.

Table 5 Coefficients f_s along X and R axes

X location (X)	$f_0(X)$	$f_1(X)$	$f_2(X)$	R location (R)	$f_0(R)$	$f_1(R)$	$f_2(R)$
0	4.794E-4	-0.1692	16.7	1.325	5.702E-4	-0.157	12.14
0.925	4.478E-4	-0.1882	18.41	2.65	1.106E-3	-0.3401	27.62
1.85	6.682E-4	-0.3486	31.97	3.975	7.015E-4	-0.2441	22.93
2.775	7.862E-4	-0.3172	26.73	5.3	5.702E-4	-0.1736	15.04
3.7	9.087E-4	-0.2837	23.51	6.625	-4.981E-5	-0.05548	7.705
4.625	2.472E-4	-0.05902	4.978	7.95	4.952E-4	-0.1105	5.22
5.55	4.818E-4	-0.1671	15.44	9.275	1.482E-4	-0.01799	4.976
6.475	-3.961E-4	0.1375	-10.58	10.6	5.748E-4	-0.2092	20.46
7.4	-4.983E-4	0.1752	-14.08	11.925	3.978E-4	-0.1422	14.13
8.325	-8.149E-4	0.312	-28.63	13.25	3.997E-7	0.01096	0.1572
9.25	-5.906E-6	0.07827	-10.72	14.575	2.964E-4	-0.06467	4.215
10.175	-3.565E-4	0.1539	-15.36	15.9	5.548E-4	-0.1654	13.45
11.1	-1.742E-4	0.1289	-16.25	17.225	6.616E-4	-0.1999	16.33
12.025	-3.121E-5	0.09636	-14.84	18.55	6.609E-4	-0.1943	15.31
12.95	-2.253E-5	0.1345	-21.55	19.875	9.254E-4	-0.3096	27.14
13.875	4.295E-4	0.004457	-9.902	21.2	9.63E-4	-0.3297	29.64
				22.525	1.38E-4	0.04436	-9.762

An attempt to reveal the heat transfer physics and uncouple the reciprocating and buoyancy forces effects on impingement heat transfer of a reciprocating jet-array with a detailed data analysis have been illustrated. The experimental heat transfer results exhibit a weak inter-correlation between Gr_p and Re controlled heat transfer mechanisms but demonstrate the evident $Re-Pu$ and $Pu-Gr_p$ coupling effects. A net result is the empirical correlation of Eq.(7) that permits the accountancy of synergistic and individual impacts of Reynolds and pulsating numbers with and without buoyancy interaction on local Nusselt number over the impinging surface of a reciprocating jet-array. The overall success of this proposal is indicated in Fig. 14 where the entire experimental data are compared with the correlative predictions evaluated by Eq.(7). The maximum discrepancy of $\pm 20\%$ between the experimental and correlating results is achieved for 85% of the entire data generated. Owing to the complex fluid mechanisms of impinging jet-array and the reciprocating force effect with buoyancy interaction, it becomes a formidable task to predict the local heat transfer. However, justified by the achieved accuracy, Eq.(7) could provide the reasonable heat transfer evaluation to assist the design initiative of adopting the impinging jet-array for cooling application of piston.

Fig. 13 Variations of χ functions with pulsating number at various axial and radial locations

Fig. 14 Comparison of experimental measurements with correlative results of reciprocating Nusselt numbers

4. Conclusions

This experimental program investigates the impingement heat transfer of a jet-array along the axial and radial axes with and without reciprocation. A detailed description of the data correlating process, which complies with the heat transfer physics of the experimentally based observations, reveals the individual and interactive effects of inertial, reciprocating and buoyancy forces on heat transfer. In conclusion the following salient points emerge from this study.

1. The confinements provided by the jet-to-jet interaction and the solid boundary that encapsulates the jet-array undermine the heat transfers of central and corner jets of array respectively. The gravitational buoyancy interaction, which appears to be the spatial function over the impinging surface, generates considerable heat transfer impediments in the stagnation areas of static jet-array. Local Nusselt numbers along the X and R axes of the static impinging jet-array are well correlated by the parameter of $\frac{Re^2}{Gr}$ in Eq. (3).

2. The reciprocating heat transfer results exhibit a weak inter-correlation between Gr_p and Re controlled heat transfer mechanisms but demonstrate the evident $Re-Pu$ and $Pu-Gr_p$ coupling effects. The extrapolating zero-buoyancy results reveal the individual reciprocating force effect, which has shown

that the heat transfer is initially reduced from the static condition but recovered at the higher pulsating number. The reciprocating buoyancy effect in isolation could improve or impede local heat transfer. The improving buoyancy effect on heat transfer occurs at the stagnant jet areas; while the impeding buoyancy effect develops at locations around the inner circumference of cubic encapsulation where the spent fluids are reversed backward to the exit. These two interactive and combined force effects induced by reciprocation could cause the relative Nusselt numbers to be reduced to the values about 55% of the static levels. Such heat transfer impediment has to be particularly considered when the impinging jet-array is adopted for the piston cooling application.

3. The proposed heat transfer correlation is physically consistent with the experimental observations, which permits the individual effects of inertial and reciprocating forces with and without buoyancy interaction on impingement heat transfer of the reciprocating jet-array to be taken in to account.

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