Heat Transfer of Confined Impinging Jet onto Spherically Concave Surface with Piston Cooling Application*

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This paper describes a detailed experimental investigation of heat transfer for confined impinging jet onto spherically concave surface with piston cooling application. The experimental conditions involved several nominal Reynolds numbers ranging from 17 500 to 43 200 with five different reciprocating frequencies, namely 0, 0.833, 1.25, 1.67 and 2.08 Hz. It resulted in the pulsating number, which represented the ratio of reciprocating force to inertial force effect, varying from 0 to $1.85E-4$. Along the spherically concave heating surface the evolution of flow structure from separation point into laminar and then turbulent annular flows was demonstrated by examining the variations of power index suffixed to Reynolds number in the non-reciprocating Nusselt number correlations. The typical effects of flow reciprocation on heat transfer were then illustrated by examining the distributions of temporal Nusselt number variation along the heating surface. When test section reciprocated, the Nusselt numbers at measurement locations periodically oscillated and the amplitude of such temporal Nusselt number variation was mostly pronounced at the stagnation point and became location and Reynolds number dependent. At the stagnation point, there were coupling effects of Reynolds and pulsating numbers on heat transfer and, in general, the heat transfer level increased with the increase of Reynolds or pulsating numbers while kept one of these two flow parameters unchanged. At Reynolds and pulsating numbers of 40 000 and $8.41E-5$, the reciprocating Nusselt number value at the stagnation point could be increased up to a level about 3.2 times of the non-reciprocating level. Less reciprocating effects were found in the spherically concave heating area; and reciprocation reduced the time-averaged heat transfer to the levels about 80% - 95% of non-reciprocating heat transfer values after flow passed the concave surface. As the reciprocating force significantly modified the heat transfer from non-reciprocating situation, it needed to account for the effect of reciprocating motion on the heat transfer within the coolant channels of piston in order to achieve the optimum design of cooling system.

Key Words: Reciprocating Confined Impinging Jet Flows, Heat Transfer Measurement, Piston Cooling

1. Introduction

For marine propulsive diesel engines, character-

ized by low speed and extremely high power output, the piston crowns must be internally cooled as the maximum gas temperatures in the combustion chambers are well above the material's melting point of piston. As an illustrative example of piston of crosshead engines, the bottom of the piston for Sulzer RTA-58T engines as shown in Fig. 1 is sealed, and lubricating oil is circulated through the cooling passages below the crown, brought in and out through internal borings in the piston rod. In this piston crown, which is shaped to aid the combustion process,

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Fig. 1 Sectioned view of piston of Sulzer RTA-58T marine heavy diesel engine

there are several internal blind bores within which coolant is circulating to convect heat conducted from the combustion chamber. As an attempt to enhance heat transfer, several jet nozzles are installed to inject high momentum coolant streams towards the blind bores. In each jet-bore cooling passage, the annular wall confines the impinging liquid jet. Manifolding for such an arrangement involves a concentric discharge blind tube, which redirects the spent coolant 180 degree following impingement on a concave surface and routes it through an annular passage in counterflow with the jet. In addition, we also notice that the piston reciprocates and there is considerable distance kept between nozzle and impinging surface for assembling purpose. In the past, although there were experimental studies addressed single-phase convection heat transfer from a heating surface to a confined impinging jet⁽¹⁾⁻⁽⁴⁾, they were performed non-reciprocally under conditions for which the impinging surface was flat and the dimensionless nozzle exit to heater separation distances were less than 15. Based on their results, tighter confinement brings influence from reversed flow of spent fluid on impinging jet which enhances turbulence production in the adjoining mixing zone and reduces the length of the potential core of jet⁽⁴⁾. Near the impinging surface, re-circulating cells emerge from reversed flow of spent fluid and the instability of jet boundaries causes entrainment of spent fluid into the central core of jet, enhancing fluctuations already present in jet efflux⁽⁵⁾. However, for piston cooling application, not only the heat trans-

fer near stagnation point plays significant role for such cooling arrangement shown in Fig. 1, the cooling performance along the inner surface of each blind bore is important as hot gas always traps in the gap between upper piston crown and cylinder. Because pistons reciprocate when engine is running, the reciprocating motion of piston further modifies the flow and heat transfer within the cooling passages illustrated in Fig. 1 from their stationary situations.

Although the study of reciprocating heat transfer of confined impinging jet is very rare in the open literature, the relevant studies which investigated the flow and heat transfer of non-reciprocating pulsatile stagnation flows were available^{(6),(7)}. For these non-reciprocating studies, the flow pulsation with sinusoidal or square waveform was provided before the flow entering the nozzle rather than induced by system reciprocation. In Sheriff and Zumbrunnen⁽⁷⁾, the frequency and amplitude of flow pulsation superimposed were respectively controlled in the range of 5 - 280 Hz and 0.5 - 100% of mean bulk-flow velocity. Their results suggested that, due to the nonlinear dynamic response of the hydrodynamic and thermal boundary layers to the pulsating force, the temporal variations of stagnation heat transfer were typical for pulsatile stagnation flows. In correspondence with this experimental phenomenon, Maladin and Zumbrunnen⁽⁶⁾ analytically demonstrated the oscillations of thermal and hydrodynamic boundary layers when temporal variations of incident flow velocity were imposed to a planar jet. The instantaneous stagnation Nusselt number increases when the thickness of boundary layers decrease while decreases if the boundary layer thickness increases. For both studies^{(6),(7)}, they concluded that time-averaged stagnation heat transfer could reduce about 17% of the non-pulsatile value when pulsation was imposed with large amplitude and low frequency. In contrast to such heat transfer impediment, small heat transfer enhancements were also observed for high frequency pulsation of small amplitude. However, it is possible that further increase of relative strength of imposed pulsatile flow could result in jet intermittency. Under this particular circumstance, the periodical boundary layer renewal along the impinging surface took place. Depending on the frequency of jet intermittency, stagnation heat transfer levels could be increased or decreased from the steady impinging jet situation, and, the heat transfer generally increased with high frequency of intermittency. The stagnation Nusselt number levels at the highest intermittency frequency tested exceeded the values for steady jets by a factor of two⁽⁸⁾. Their experimental results strongly suggested that such heat transfer enhancements were pre-

dominantly due to boundary layer renewals and not to the turbulence effects.

So far it is clear the imposed flow pulsation considerably influence the stagnation heat transfer for a planar jet in addition to the effects of confinement when the cooling geometry shown in Fig. 1 is considered. As we expect the reciprocating confined impinging jet flows share certain degree of similarity with stationary pulsatile stagnation flows, the influences of piston reciprocation on heat transfer within the cooling passages depicted in Fig. 1 are foreseeable. For piston-cooling application, an experimental program aimed at investigating the effects of system reciprocation on heat transfer for confined impinging jet flow was formulated and a brief description of the experimental apparatus and methods of data reduction and presentation now follows.

Nomenclature

English symbols

- A : Coefficient
 c_p : Specific heat of fluid
 D_c : Inner diameter of confining tube
 D_n : Outer diameter of jet nozzle
 d : Hydraulic diameter of test section
 f : Reciprocating frequency of test module
 k : Thermal conductivity of fluid
 L : Active heating length of test module
 n : Power index of Reynolds number
 Nu : Reciprocating Nusselt number
 Nu_0 : Non-reciprocating Nusselt number
 Nu_{Deff} : Non-reciprocating Nusselt number based on effective diameter
 P^* : Dimensionless flow pressure

- p : Flow pressure
 Pu : Pulsating number
 Pr : Prandtl number
 q : Convective heat flux
 r : Radius of crank arm
 $\vec{R}$: Position vector of crank arm
 Re : Reynolds number
 T_f : Bulk flow temperature
 T_w : Wall temperature
 v : Flow velocity vector
 $\vec{V}$: Reciprocating velocity vector
 W_m : Flow velocity on the exit plane of nozzle
 z : Downstream coordinate
 Z : Dimensionless downstream coordinate (z/L)

Greek symbols

- $\vec{\omega}$: Rotating vector of crank arm
 $\tilde{\omega}$: Dimensionless rotating vector of crank arm
 Ω : Angular velocity of rotating crank
 ρ : Fluid density
 μ : Fluid dynamic viscosity
 $\tilde{\epsilon}$: Dimensionless crank arm
 η : Dimensionless flow temperature
 Ψ : Unknown function

2. Experimental Apparatus and Methods of Data Presentation and Reduction

2.1 Reciprocating test facility

Figure 2 shows the schematics of the reciprocating test facility in which a gravity tank (1) with the equivalent pressure head of 15-meter water-height provides the required pressure head to facilitate forced convective flow into the heat transfer test section within the reciprocating module (2). Prior to entering the heat transfer module (2), cooling water

Fig. 2 Systematic diagram of experimental facility

flows through two control-valves (3), (4), a float-type volume flow meter (5) and a flow calming tube (6), which has straight length of about 60 equivalent diameter of test duct. As a result the flow is hydrodynamic developed before entering the test section. In the upstream or downstream direction of the flow meter (5), a straight section with length about 20 tube-diameter is arranged. During experiments, mass flow rates required to achieve the predetermined Reynolds number values are obtained by adjusting the needle valve (4).

A crank-wheel-mechanism driven by a 1500 W DC electrical motor (7) is selected to create the reciprocating motion of the test module (2). Through the pulley-belt unit (8) and the reduction gear box (9), the fly wheel (10) can be controlled to rotate at any desirable speed which consequently drive the heat transfer test module (2) to reciprocate at any preset frequency. The inertia of fly wheel (10) also assists to create the "smooth" reciprocating motion and, on this wheel, a counter balance weight (11) is fixed on the opposite side of heat transfer module (2) in order to remain the dynamic balance during the reciprocating tests. An optical pick-up (12) is installed beside the shaft that transmits the speed signal into the digital display on control console in order to measure the rotating speed and therefore the reciprocating frequency. All the temperature measurement signals sending from the thermocouples are connected with a Fluke Hydra 2620A-100 data logger from where these measurements are sent into a DX 33 PC in which an on-line data acquisition program is

installed to monitor the experimental conditions. Nevertheless, it is worthy of noting that, when the heat transfer module reciprocates, the time varied heat transfer due to the unsteady nature of reciprocating force is expected. As it is intended to reveal the amplitude of instantaneous heat transfer variation against time for each measured spot during reciprocating tests, the time varied wall temperature values are detected and stored using the on-line data acquisition system. With the instantaneously lowest and highest wall temperature values at any axial location been sorted out, the ranges of temporal wall temperature and Nusselt number variations for all measured locations along the heated surface are detected using this data acquisition system. The electrical transformer connected with power supply, voltmeter and ammeter adjusts the amount of electrical heating power feeding into the heat transfer test section.

2.2 Heat transfer test modules

The constructional details of heat transfer test module is illustrated by Fig. 3 in which a 18 mm wide circular nozzle (1) with jet diameter of 2 mm issues the high momentum water jet impinging onto the concave spherical surface. This semi-spherical cup (2) and the cylindrical barrel with confinement diameter, D_c , of 45 mm (3) are made of bronze-copper having a thermal conductivity of 50 W/mK at temperature 20°C. Distance between nozzle jet and stagnation point of the impinging wall is 22.5 jet-diameter. This assembly simulates the cooling system of piston for Sulzer RTA 58T engine. However, due to the effect of

Fig. 3 Constructional details of heat transfer test module

pulsation on the water head that facilitates the impinging-jet cooling there will be a periodically temporal pressure wave created by system reciprocation within the nozzle tube (1) and consequently affect the flow characteristics of jet. Because above situation similarly occurs in each "impinging bore" within a piston of marine diesel engine as shown in Fig. 1, the flow within nozzle tube (1) is designed to reciprocate with test section in order to reflect the effect of water-head pulsation on heat transfer. On the outer surface of semi-spherical cup (2) and cylindrical barrel (3), 1 mm deep grooves with equal interval of 2 mm were machined for installation of Ni/Cr resistance wire (4) to provide electrical heating power. As shown in Fig. 3, the downstream coordinate system is defined from the stagnation point toward the flow exit (6) and ten thermocouples (5) in Z direction are embed with 2 mm depth into the heated wall to measure the axial wall temperature distribution. As we consider the flow fields investigated are symmetrical about the stagnation line, nine thermocouples for water-temperature measurement (7) are inserted into the flow passage at positions in opposite to the wall-thermocouples (5) as shown in Fig. 3. However, to avoid the disturbance of flow at the stagnation point due to the insertion of thermocouple, the flow bulk temperature corresponding to this stagnation point is defined and measured at the flow entrance (8). To minimize the external heat loss, thermal insulation material (9) was wrapped around the test unit and encapsulated within the casing (10).

2.3 Methods of data presentation and reduction

As described by the experimental apparatus, the reciprocating motion of heat transfer test module produces modifications of the flow fields from the non-reciprocating situation and these reciprocating effects must be taken into account parametrically for accurately describing the heat transfer relationships of reciprocating flows. For the piston cooling application, the crank arm with radius r , which rotates at speed Ω , created the reciprocal velocity in terms of $\vec{\omega} \times \vec{R}$ where $\vec{\omega}$ and $\vec{R}$ are the rotating vector of crank shaft and the position vector of crank arm respectively. It is worth noting that the direction of position vector of crank arm is temporally varied following the rotation of crank arm. Therefore, the translating velocity of reciprocating test module inherits a characteristic of temporally periodical variation. The unsteadiness of such reciprocating velocity is therefore resulted and we expect the flow and heat transfer with system reciprocation should vary with time. In order to study and parametrically characterize the flow and heat transfer for a reciprocating flow system, it is convenient to refer the fluid motion to an observer

fixed relative to the reciprocating system. Thus, for convenience, the flow system investigated is referred to a reciprocating reference frame with reciprocating motion created by the crank-arm system. When considering the flow inside a reciprocating conduit with fluid motion to be specified by this reciprocating reference frame, the inertial term appears in the momentum equation needs to be modified in order to include the reciprocation associated accelerations. Based on this approach⁽⁹⁾, we have derived the dimensionless flow momentum equation written as

$$\frac{DV}{Dt} = -\nabla P^* + \frac{1}{Re} \nabla^2 V + Pu(\vec{\omega} \times \vec{\varepsilon}) \cdot \nabla V \quad (1)$$

All symbols in Eq. (1) are defined in nomenclature.

As shown in Eq. (1), the motion of fluid is governed parametrically by two dimensionless groups, namely Reynolds number, Re , and pulsating number, Pu . The convective flow inertia that motivates the fluid motion is characterized by Reynolds number, Re , representing the ratio of inertial to viscous force effect. In the customary manner for forced convection, it is defined as

$$Re = \frac{\rho W_m d}{\mu} \quad (2)$$

in which W_m is the exit velocity of the jet, d is the hydraulic diameter of test duct, ρ is the density of fluid and μ is the coolant viscosity.

With reciprocation, the pulsating number, Pu , which represents the ratio of reciprocating force to inertial force effect, is defined as

$$Pu = \frac{\Omega r}{W_m} \quad (3)$$

The pulsating number, Pu , represents the relative strength of the time-averaged force ratio between the reciprocating and forced convective inertial forces. The numerator of Eq. (3) derives from the translating velocity of the test section and can be treated as an indication of the time-averaged strength of reciprocating force. However, for the stationary forced convective tests, the pulsating number becomes zero, representing zero reciprocating effect.

The convective heat transfer produced at any specified location over the surfaces of a reciprocating duct may be expressed in the customary manner by the local reciprocating Nusselt number, Nu , and defined experimentally as

$$Nu = \frac{qd}{k(T_w - T_f)} \quad (4)$$

Where q is the convective heat flux carried by flow, k is the thermal conductivity of coolant and T_w and T_f represent the local wall temperature and the flow bulk temperature respectively.

The variation of heat transfer along the reciprocating test module also depends on the geometrical

features and the nature of the thermal boundary conditions. Thus, in view of the time averaged heat transfer variation for this reciprocating system, the local heat transfer may be parametrically governed by an equation of form :

$$Nu_s = \Psi\{Re, Pu, Pr, Z, \text{boundary conditions}\} \quad (5)$$

where Ψ is an, as yet, unknown function to be determined and Pr is the coolant Prandtl number. The experimental program undertaken involves the examination of the interactive influences of these parameters shown in Eq. (1) on heat transfer. The Reynolds numbers tested vary from 17 500 to 43 200 with five different reciprocating frequencies, namely 0, 0.833, 1.25, 1.67 and 2.08 Hz. Based on the flow velocity on the exit plane of nozzle, the pulsating numbers for this set of experiments is in the range of 0 to 1.85E-4. When the dimensionless parameters listed in Eq. (5) are evaluated, the constituent fluid properties are calculated in accordance with the local coolant temperature measured.

An uncertainty estimation of the data reduction was performed in accordance with the editorial policy of JHT⁽¹⁰⁾. Because the local fluid properties required to constitute the dimensionless groups shown in Eq. (5) are estimated from the fluid's bulk temperature, one of the major sources to attribute uncertainty is the temperature measurement and the uncertainty in temperature measurement is estimated to be $\pm 0.5^\circ\text{C}$. The maximum percentages of error for the coolant's mass flow rate, thermal conductivity, the temperature difference of wall-to-fluid, the diameter of nozzle and the heating area are estimated as $\pm 5\%$, ± 0.4 , $\pm 6\%$, $\pm 0.6\%$ and 1.6% respectively. With the temperature difference of wall-to-fluid varied from 15 to 53°C , the maximum uncertainty of Nusselt number is about 13%. For the Reynolds and pulsating numbers, their maximum uncertainties are estimated as 5% and 3% respectively.

3. Results and Discussions

Due to the rareness of the geometrical feature tested by present study, a series of non-reciprocating heat transfer tests were carried out to provide the heat transfer reference datum to which the reciprocating results could be compared. Since the range of coolant temperatures covered by the present study did not cause a significant change in the Prandtl number, the focuses for the non-reciprocating tests were the influences of Reynolds numbers on local Nusselt number distributions. For the reciprocating tests, as we expected both spatial and temporal Nusselt number variations, the influences of Reynolds and pulsating numbers on the local instantaneous and time-aver-

Fig. 4 Effect of Reynolds number and separation distance on stagnation Nusselt number for different nozzle and confinement ratios

aged Nusselt numbers were examined.

3.1 Non-reciprocating results

The heat transfer results at the stagnation point are presented in terms of dimensionless Reynolds and Nusselt numbers as depicted in Fig. 4. Also indicated in this figure are the results of Besserman et al.⁽⁴⁾ for which their heat transfer test conditions are geometrically and parametrically similar to the present study, except the impinging surface tested by Besserman et al.⁽⁴⁾ is flat rather than spherically concave. For the purposes of consistency and comparison in order to assess the effect of nozzle-to-heater separation distance on heat transfer, the characteristic length, D_{eff} , for the Nusselt number, $Nu_{D_{eff}}$, is chosen to be the length used by Besserman et al.⁽⁴⁾ as 15.3 mm. As shown in Fig. 4, although the heat transfer levels decrease with the increase of separation distance, S/d , when the data from present study and Besserman et al.⁽⁴⁾ are compared, the tendency for Nusselt number variation with Reynolds number shares the similar increasing slope. For $S/d=3.0$, impingement may still occur within the potential core, thereby preserving the centerline velocity, while enhancing heat transfer due to the fluid mixing as a result of entrainment from the spent reversing fluid after impingement⁽⁴⁾. For $S/d=5.0$ ⁽⁴⁾ and $S/d=22.5$ (present study), the attendant decay in the centerline velocity reduces heat transfer even if the enhanced mixing associated with the entrainment and counterflow between spent reversing-flow and jet-stream prevails over the impinging zone. After this stagnation point where the flow momentum is significantly reduced, the boundary layers re-develop in the downstream direction and the

Fig. 5 Non-reciprocating Nusselt number distributions along heating surface at Reynolds numbers of 18 000, 26 000, 30 000, 35 000 and 40 000

flow following the spherically concave heating surface is considerably stabilized due to the loss of flow momentum. This modification in the flow field affects the downstream heat transfer and its influences can be illustrated by the Nusselt number distributions along the heating surface as shown in Fig. 5.

Figure 5 shows the typical Nusselt number distributions along the heating surface at five different flow rates with corresponding Reynolds numbers of 18 000, 26 000, 30 000, 35 000 and 40 000. As shown for each tested Reynolds number, the spatial maximum of heat transfer occurs at the stagnation point when the separation distance, S/d , keeps at 22.5. After this stagnation point, the downstream local Nusselt number values in the spherically concave region considerably decay from the stagnation level. However, among this area where $0 \leq z/L < 16\%$, the Nusselt number values increase with the increase of Reynolds number, but the bands of Nusselt number spread due to the variation of Reynolds number are gradually "shortened" when increasing downstream distance as depicted in Fig. 5. In other words the increasing rate of local Nusselt number with Reynolds number systematically reduces as flow developed downstream, indicating the phenomenon of flow stabilization. At location of $z/L = 16\%$ where the spherically concave surface joints with the cylindrical annulus, the Nusselt number values for all the Reynolds number tested fall into a level agreed well with a Nusselt number value of laminar annular pipe flow⁽¹¹⁾. Because the Nusselt number value at location of 16% z/L remains at 4.95 for all the different Reynolds numbers tested, the heat transfer at this particular spot becomes Reynolds number independent and is stabilized into laminar flow. After this particular point, the Nusselt number value for each the Reynolds number examined sudden-

ly increases and then slightly reduces into a constant level when axial locations, z/L , exceeds 30% . Also indicated by the results at locations beyond the laminarized spot ($z/L = 16\%$) is the increased data band driven by Reynolds number variation.

Above all, among the concave heating surface, the heat transfer rates reduce rapidly from the stagnation level into a value corresponding to the laminar annular flow at the location where the spherically concave surface joints with cylindrical annulus. Thus we expect that the flow in this region is strongly laminarized after the impingement onto the spherically concave heating surface. In this particular region, although the spatially characteristic distributions of heat transfer for impinging jet flow are observed, the variation pattern of heat transfer for present study differs from the typical "bell-function distribution" usually resulted from the impinging jet onto the flat surfaces⁽⁷⁾. Such difference in the spatial heat transfer distribution can be resulted from the concave heating surface. Nevertheless, if the influences of separation distance, confinement, heating geometry, flow entrainment and Prandtl number are resolved into a location depend coefficient, A , the local non-reciprocating Nusselt number correlation for the stagnation point, the area of laminarized concave surface and the annular flow region may be the form of

$$Nu_0(Z) = A(Z) \times Re^{n(Z)} \quad (6)$$

Coefficients A and n in Eq. (6) are functions of downstream location and the examination of the variation of the power index, n , against the downstream location, Z , may reveal the evolution of flow structure along the heating surface. In Sheriff and Zumbrennen⁽⁷⁾, and Besserman et al.⁽⁴⁾, the coefficients, n , in their Nusselt number correlations for stagnation spot were derived respectively as 0.589 and 0.5. As shown in Fig. 6, the power index of Reynolds number, n , at the stagnation point for the present study is found as 0.5, indicating the predominant heat transfer physic to be impingement. After this stagnation spot, the coefficient, n , gradually reduces towards 0 when wall jet flow develops further downstream. At the laminarized spot where $z/L = 16\%$, the coefficient n becomes 0 so that the flow structure is confirmed as laminar flow. In the spherically concave region, another coefficient, A , in Eq. (6), that physically represents the enhancing heat transfer effects due to increased fluid mixing, continuously increases from the stagnation point when the coefficient, n , gradually reduces. Therefore, the flow structure in the spherically concave region is gradually laminarized when the wall-jet flow develops downstream. In the meanwhile, jet entrainment and the counter-flows between spent reversing fluid and jet stream continuously

Fig. 6 Variations of coefficients A and n for non-reciprocating Nusselt number correlation along heating surface

enhance fluid mixing during the laminarized process. After the wall-jet flow passes the laminarized spot, the coefficient, n , suddenly increases toward the turbulent annular flow value from 0 to 0.8. At station of 19% z/L , the wall-jet flow starts transiting into turbulent annular flow. Further downstream, the Nusselt number value fixes at a level corresponding to turbulent annular flow so that the coefficients A and n after 38% z/L remain unchanged.

3.2 Reciprocating results

It is clear, due to the particular geometrical feature constructed by the heat transfer test module, several different flow structures are resulted when flow travels along the heating surface. Because the flow structures are location dependent as demonstrated in previous section, we expect the reciprocating motion may have different heat transfer effects in these different flow regions.

Figure 7 shows the instantaneous Nusselt number distribution along heating surface when test module reciprocates at frequency of 2.08 Hz. The equivalent nonreciprocating results obtained at the same Reynolds number of 32 000 are also depicted as the dot line shown in Fig. 7 for the purpose of comparison. For the reciprocating results, we observe the temporal Nusselt number variation at most of the spots where thermocouples are installed. The range of such temporal Nusselt number variation at each measurement location is typified by a data span as shown in Fig. 7 and in the middle of each data span shows a mark

Fig. 7 Typical reciprocating Nusselt number distribution along heating surface at Reciprocating frequency of 2.08 Hz

point that corresponds to the time-averaged Nusselt number value. However, such time varied heat transfer results due to the unsteadiness provided by the reciprocating force are typical for all the reciprocating tests. As shown in this figure, the significant instantaneous heat transfer variation appears at the stagnation point. At the position of 19% z/L where the flow transits into turbulent flow, relatively large data span indicates considerable temporal Nusselt number variation. Among the spherically concave surface, the data span is gradually shortened from the stagnation point toward the laminarized location with $z/L=16\%$ at which the temporal Nusselt number variation almost disappears. It is worth of noticing, for the present study; the flow bulk temperatures at locations corresponding to the wall temperature measurement spots are directly measured by thermocouples inserted into the flow passage. When the test module reciprocates, the slight temporal variation of flow bulk temperature at each measurement spot is found. As the on-line data acquisition system simultaneously catches the instantaneous wall and flow temperatures, the resultant temporal Nusselt number variations shown in Fig. 7 are the influences of reciprocating force on the flow structure and turbulence. At and near the stagnation point, because the eddy motion due to the separation takes place, the influence of reciprocation on heat transfer is pronounced. If the last term of Eq. (1) is referred, we expect large reciprocating effect in the stagnation region, where considerable velocity gradient in a same direction of reciprocating acceleration appears when the high momentum jet impinges onto the heated surface. Nevertheless, once the flow develops toward the laminarized point, the influence of reciprocating force on turbulence reduces due to the reduction of turbu-

lence as illustrated in previous section which therefore leads to a tendency of reducing data span in the downstream of spherically concave surface. Additionally, as the velocity gradient of flow in this region is relatively moderate compared to the stagnation region, the reciprocating effect on heat transfer in this flow region is further weakened. Further downstream from the laminarized spot, a considerable Nusselt number span re-appears at the transition point where the flow structure starts yielding into turbulent. At this particular location where instability waves may occur⁽¹²⁾, the introduction of reciprocating force tends to further enhance the "chaos" of flow so that the amplitude of temporal heat transfer variation increases. Once the flow develops into turbulent annual flow, the ranges of temporal Nusselt number variations at locations of 30%, 38% and 49% z/L are almost identical. Nevertheless, when the time-averaged reciprocating and non-reciprocating Nusselt number distributions along the heating surface are compared, significant heat transfer enhancement due to reciprocating effect is resolved at stagnation point while the heat transfer impediments appear after the spherically concave area. Therefore, it is important to consider the effects of reciprocating motion on the heat transfer performances of cooling passages arranged within a piston.

To further investigate the effect of reciprocation on heat transfer, the reciprocating time-averaged Nusselt number at each measurement location is divided by the equivalent non-reciprocating Nusselt number value, Nu_0 , so that this Nusselt number ratio quantifies the influence of reciprocating force on heat transfer. The distributions of such Nusselt number ratios, Nu/Nu_0 , generated with the same Reynolds number but made up with three different pulsating numbers; are collected and compared in each individual graph of Fig. 8. As a first glance shown Fig. 8, the considerable heat transfer enhancement with a consistent tendency to be further increased when the pulsating number increases is observed at the stagnation point for each individual graph. When Reynolds number gradually increases from 27 000 to 40 000, the effect of pulsating number variation on heat transfer at the stagnation point increases with the Reynolds number. At Reynolds and pulsating numbers of 40 000 and $8.41E-5$, the reciprocating Nusselt number value at the stagnation point could be increased up to a level about 3.2 times of the non-reciprocating level. Among the spherically concave heating area, the Nusselt number ratios are quite close to unity, indicating less reciprocating effect. After this flow region, the time-averaged reciprocating Nusselt number values are reduced to the levels about 80%~95% of non-reci-

Fig. 8 Effect of reciprocation on heat transfer distributions along heating surface

procating heat transfer values and this heat transfer impediment shows quite independent to the variation of Reynolds number if the results shown in each individual graph are cross examined. Thus, we presume there are coupling effects of Reynolds and pulsating numbers on the reciprocating heat transfer at the stagnation point. For the remaining flow regions, we find that the variations of Reynolds number from one graph to another do not noticeably enlarge or reduce the data spans driven by the variations of pulsating number at each measurement spot. Therefore, within the parametric conditions tested, the power index, n , involving in Eq. (6) for these remaining flow regions may not be yielded by system reciprocation if the relationship between Reynolds and reciprocating Nusselt numbers is correlated by the format of Eq. (6). Nevertheless, the spatially averaged Nusselt number ratios, Nu/Nu_0 , along the heating surface for all the data shown in Fig. 8 are in the range of 1.03~3.3, which suggests an overall heat transfer enhancement on the heat transfer of such confined impinging jet due to pulsation effect. Thus, from an industrial point of view, the experimental results verifies the

overall beneficial effect resulted from system reciprocation if the measure of impinging “bore” cooling scheme applies to the cooling of piston in the marine diesel engine. Further reduction of separation distance from 22.5 to 20 jet diameter enhances the reciprocating heat transfer level and the effects of system reciprocation for flow regions near the stagnation and transition points, while the increase of separation distance from 22.5 to 25 jet diameter shows the inverse reciprocating effects⁽¹³⁾. In the annular flow region, the variation of separation distance from 20 to 25 jet-diameter does not show considerably different reciprocating effects. Therefore, with the separation distance varied from 20~25 jet-diameter, the applicability of these experimental results remains.

Figure 8 suggests the inertial and reciprocating forces, which are characterized by Reynolds and pulsating numbers respectively, generate the coupling effects on stagnation heat transfer. In other words the variations of reciprocating Nusselt number driven by the changes of pulsating number depend on the value of Reynolds number tested. To illustrate and investigate this particular heat transfer phenomenon further, the reciprocating stagnation heat transfer in terms of Nusselt number ratio, Nu/Nu_0 , are plotted against Reynolds number with the test module reciprocates at frequencies of 0.83, 1.67 and 2.08 Hz in Fig. 9. The increase of Reynolds number at a fixed reciprocating frequency actually reduces the value of pulsating number. However, if the reciprocating frequency is fixed at 0.83, 1.67 or 2.08 Hz, the Nusselt number ratio, which represents the effects of reciprocation on heat transfer, increases with the increase of Reynolds number and such heat transfer increasing rate becomes higher at high Reynolds numbers. When the Reynolds number is fixed, the increase of pulsating number by increasing the reciprocating frequency systematically increases the Nusselt number ratio for all the Reynolds numbers tested. Thus, within the parametric range tested, the increases of Reynolds and pulsating numbers result in further heat transfer enhancement at the stagnation spot.

4. Conclusions

We experimentally examine the non-reciprocating and reciprocating heat transfer of confined impinging jet onto spherically concave surface, for application to the design of cooled piston of marine heavy diesel engine. In conclusion the following observations result from this investigation.

1. Due to the geometrical feature tested, the non-reciprocating heat transfer distribution along the heating surface shows a local maximum at the stagnation spot from which the Nusselt number values decay

Fig. 9 Effect of reciprocation on stagnation heat transfer

toward a level corresponding to the laminar annular flow. Examination of the spatial variation of power index, n , ensures that the flow structures are evolved from the separation flow at stagnation point toward laminar annular flow in the region of spherically concave surface while the flow then transits into turbulent in the further downstream area.

2. When the test module reciprocates, the temporal Nusselt number variations at most of the spots along the heating surface are observed due to the unsteadiness provided by the reciprocating force. Because the flow structures vary with the downstream location, the influence of reciprocation on heat transfer is location dependent and therefore the ranges of temporal heat transfer variation are spatially varied with the largest temporal span resolved at the stagnation point. In the spherically concave heating area where the flow is stabilized, the spans for temporal Nusselt number variations are gradually shortened from the stagnation point toward the laminarized spot at which the temporal Nusselt number variation almost disappears. Following this laminarized spot beyond which the flow transits into turbulent flow, relatively large span of temporal Nusselt number variation occurs and the ranges of temporal Nusselt number variation in the turbulent annular flow region become almost identical.

3. At the stagnation point, the most obvious reciprocating effects on heat transfer are found with considerable enhancements relative to the non-reciprocating situations. At Reynolds and pulsating numbers of 40 000 and $8.41E-5$ respectively, the heat transfer level at this particular spot could be increased up to about 3.2 times of the non-reciprocating value. In the spherically concave heating area, the time-averaged reciprocating Nusselt number ratios, Nu/Nu_0 , are quite close to unity, indicating less reciprocating effect. After this flow region, reciprocation reduces

the time-averaged heat transfer to the levels about 80%~95% of non-reciprocating heat transfer values. In terms of the spatially averaged Nusselt number ratios, Nu/Nu_0 , along the heating surface, an overall heat transfer enhancement on the heat transfer of the confined impinging jet due to pulsation effect is concluded. From an industrial point of view, it is experimentally confirmed the overall beneficial effect due to system reciprocation when the measure of impinging "bore" cooling scheme applies to the cooling of piston in the marine diesel engine.

4. The inertial and reciprocating forces, which are characterized by Reynolds and pulsating numbers respectively, generate the coupling effects on stagnation heat transfer. Therefore, with the same variation span of pulsating number, the varying bands of Nusselt number ratio, Nu/Nu_0 , are different when tests are performed with different Reynolds numbers. At this stagnation point, the Nusselt number ratio, Nu/Nu_0 , increases with the increase of Reynolds number at fixed reciprocating frequency. When the Reynolds number is fixed, the increase of pulsating number increases the stagnation Nusselt number ratio for all the Reynolds numbers tested. Therefore, within the parametric range tested, the increase of Reynolds or pulsating numbers while keeps one of these two flow parameters unchanged provides further heat transfer enhancement at the stagnation spot.

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