Heat Transfer in Reciprocating Spiral Tube with Piston Cooling Application*

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This paper describes a detailed experimental investigation of heat transfer in a reciprocating spiral tube with particular reference to the piston cooling application. The flow studied is turbulent upon entering the coils but transits into laminar in the further downstream of the coils. A selection of heat transfer measurements with which the physics of pulsating and buoyancy forces interactively affect the heat transfer along the inner and outer edges of the reciprocating coiled tube is illustrated. The pulsating force with buoyancy interaction causes the considerable heat transfer modifications from the static results. Although enhancing the buoyancy level improves heat transfer, the local Nusselt number in the reciprocating coils is initially impaired from the static value with weak reciprocation; but recovered at the higher level of pulsating force. This study focuses on the development of the experimental procedure that could lead to a physically consistent empirical correlation, which assists to evaluate the local heat transfer in the reciprocating coils by permitting the individual and interactive effects of centrifugal force, torsional force, pulsating force and reciprocating buoyancy on the forced convection to be quantified.

Key Words: Reciprocating Heat Transfer, Spiral Cooling Passage

1. Introduction

For commercial ship propulsions, the developments of marine diesel engine have been directed toward the low fuel consumption, slow engine speed and high power-to-weight ratio. The typical values of engine speed, maximum cycle pressure and temperature for this class of engine are, respectively, in the range of 90-115 rev/min and about 125 bar and 1500°C. The hot gas trapped in the combustion chamber, which usually involves swirls, provides considerable thermal loads on the piston. The swirling and blow-by-exhausted gases are highly erosive, especially when the material of piston is softened at high temperature. In order to minimize the material loss from the surface of piston crown due to the erosive damage, the heat transfer augmentation for the cooling system in the piston becomes a design focus. It is also important to ensure that the material tempera-

* Received 26th February, 2001
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Fig. 1 Conceptual design of cooling networks in a piston
to facilitate effective cooling. After impingement, the coolant is forced to channel in the spiral cooling passage, which flow cools the upper edge of piston crown and the piston skirt. When the coolant flows in the spiral passage, the centrifugal force arises to modify the flow configuration that improves the heat convection. The exit flow is then directed toward the concentric passage in the piston rod, which converges the residual heat out of the piston for the further usage. This recovered energy has been used as the heating sources for the distillation plant and the marine boiler in order to improve the overall thermal efficiency of a ship. Note that, the piston motion affects the heat transfers of impinging jets array and the convective flow in the spiral passage. It is the combined effect of the forces induced in the spiral passage and the reciprocating motion that the present investigation is concerned.

When the coolant flows through the curved tube, the centrifugal force is induced to generate the cross-plane secondary flow cells, among which the fluid is driven from the central core toward the outer edge. The cross-sectional distribution of streamwise velocity is distorted with its peak-value shifting toward the outer surface of the duct. When these Dean-type vortices are present, the spatially averaged heat transfer is enhanced, but the local heat transfer varies peripherally. On the outer edge of the curved tube, the degree of heat transfer augmentation increases with the increase of Dean number; while the inner-edge heat transfer could be reduced from the straight tube level by increasing the Dean number when the through flow momentum becomes relatively weak. Also the critical Reynolds number for the laminar-to-turbulent transition in the curved tube is modified. In general, the transition to turbulent flow in the curved tube requires higher Reynolds numbers than those in a straight duct. It therefore could be a practical situation that the flow is turbulent upon entering the coils but becomes laminar in the further downstream of the coils. In such a range of Reynolds numbers, Kalb and Seader experimentally demonstrated a rapid transition to laminar flow in the coils. As depicted in Fig. 1, the turbulent-to-laminar transition could take place in the spiral passage. When such transition occurs, the cooling performance is degraded and the variation of heat transfer in the developing flow region becomes important.

In addition to the centrifugal force effect illustrated above, the torsional force induced in a spiral tube has been demonstrated to affect the flow and heat transfer, which effect could cause the asymmetric distributions of secondary flow and isotherm on the cross plane of tube when the pitch of coils is substantial. A loose-coiling analysis that embedded a helical coordinate system to describe the Navier-Stoke concept led to two dominant parameters, namely Dean and Germano numbers, which respectively quantified the relative strength of centrifugal and torsional forces in the coils. The interaction of buoyancy with the vortices induced by the torsional and centrifugal forces in the coils further complicates the heat convection. By enhancing the buoyancy level, the spots with maximum and minimum heat transfer values, which locate respectively on the outer and inner edges of the coil when the buoyancy interaction is absent, could rotate. This buoyancy interaction therefore causes a reduction of heat transfer difference between the inner and outer surfaces. When the spiral cooling passage in the piston reciprocates, the unsteady pulsating and reciprocating buoyancy forces could interact with the secondary flows induced by the torsional and centrifugal forces in a complex manner. Neither the flow configurations nor the characteristics of heat transfer in such a reciprocating spiral passage have been previously reported in the open literature. However, a few studies have look into the effects of imposition of the oscillating and pulsatile pressures waves on the flow and heat transfer in a static coil. With the presence of pulsatile pressure waves, a variety of unsteady cross-plane secondary flows develop. In such a coil, the cross-pipe motion consists of three re-circulating regions, which rotate around the center of the pipe, changing their size and orientation in time. When the pulsatile amplitude increases considerably, the richer cross-pipe secondary flow structures develop. Due to the development of unsteady secondary flow cells in a spiral tube, the instant and circumferential time-averaged heat transfer could be considerably increased. In Ref.(14), the temporally varied flow field showed a reverse phase angle relative to the instant heat transfer so that the instant with maximum flow velocity in a pulsatile cycle corresponded to a minimum local heat transfer rate. Although there is still the lack of studies investigating the heat transfer of pulsating and oscillating flows in the static coils, which involves the streamwise turbulent to laminar transition, there are fundamental differences between the pulsating or oscillatory flows in a static duct with the reciprocating duct flows. Therefore it is felt that the heat transfer in the reciprocating spiral tube differs from the pulsatile flow in a static likewise spiral tube.

The present paper describes the results of a series of experiments aimed at studying the effect of reciprocating motion on heat transfer in a smooth-walled spiral tube. This study forms part of the on-going
research program into the heat transfer augmentation of reciprocating flow with application to the piston cooling. It has two phases. Initially the static tube 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. The detailed heat transfer measurements along the inner and outer helix diameters of the coils with and without system reciprocation are initially illustrated. It is followed by a parametric study of the coupled and individual effects of pulsating and reciprocating buoyancy forces on heat transfer with the attempt to reveal the heat transfer physics in the reciprocating spiral tube. Finally, a methodology is proposed to devise the physically consistent heat transfer correlation of the reciprocating spiral tube for the design application.

Nomenclature

English symbols

$A, B$: constant coefficients
$Bu$: buoyancy number $=\beta(T_w-T_s)\rho^2Rd/W_n^2$
$C_p$: specific heat of coolant [Kg$^{-1}$K$^{-1}$]
$d$: hydraulic diameter of test tube [m]
$D$: diameter of coil [m]
$Dn$: Dean number $=Re\sqrt{d/D}$
$f_{ax, ax}$: axial location dependent coefficients
$Gn$: Germano number $=Re$
$H$: pitch ratio of coil $=P/0.5d$
$k$: thermal conductivity of coolant [Wm$^{-1}$K$^{-1}$]
$Nu$: reciprocating Nusselt number $=q_d/(T_w-T_s)k_d$
$Nu_n$: non-reciprocating Nusselt number
$Nu_w$: Dittus-Boelter Nusselt number value
$Pu$: Pulsating number $=\omega R/W_n$
$P$: pitch of coil [m]
$q$: convective heat flux [Wm$^{-2}$]
$Re$: reciprocating amplitude [m]
$Re$: Reynolds number $=\rho W_d/\mu$
$T_s$: flow bulk temperature [°C]
$T_w$: wall temperature [°C]
$W_n$: mean through flow velocity [ms$^{-1}$]
$z$: axial coordinate [m]
$Z$: dimensionless axial location $=z/d$

Greek symbols

$\beta$: thermal expansion coefficient of coolant [K$^{-1}$]
$\eta$: dimensionless torsion parameter $=(H/2\pi)/(D/d)^2+(H/2\pi)^2$
$\rho$: coolant density [Kgm$^{-3}$]
$\mu$: dynamic viscosity of coolant [Kgs$^{-1}$m$^{-1}$]
$\omega$: angular velocity of rotating disc creating reciprocating motion [rads$^{-1}$]
$\phi$: turning angle of helix in respect to the central axis of coil [rad]
$\xi, \zeta$: unknown functions

Superscripts

$I$: inner edge of coiled tube
$O$: outer edge of coiled tube

Subscripts

$0$: static state

2. Experimental Details

2.1 Strategy

The primary tasks of the present study are to generate the heat transfer data in a reciprocating smooth-walled spiral tube and to examine the combined effect of the induced centrifugal/torsional forces and reciprocating motion on heat transfer. The translating velocity of test duct, motivated by the rigid slider-crank mechanism of a piston or the present reciprocating facility, is not a constant value but a skewed temporal sinusoidal function. With large length ratio of the connecting rod to the crank arm, the translating velocity of test duct can be approximated by $Re\sin\omega t[19]$. Because the temporal velocity of the reciprocating test duct is not constant but periodically varies, a modified version of Navier-Stokes equations that considers the effect of 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 equations reveal the governing non-dimensional flow parameters those control the convection of fluid momentum and energy. A study of the momentum conservation equations, with the fluid motion referred to a coordinate frame which reciprocates with the flow boundary itself, identifies two reciprocation related flow parameters; namely the pulsating ($Pu$) and reciprocating buoyancy ($Bu$) numbers[19]. The pulsating number, $Pu$, and buoyancy number, $Bu$, respectively, quantify the ratio of pulsating to inertial forces and the relative strength of the reciprocating buoyancy effect[19]. 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 ($Nu$) in a reciprocating spiral tube could be parametrically described by the following equation

$Nu = \phi(Re, Dn, Gn, Bu, Boundary \ conditions)$

(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 correlation is
the strategic aim. For a predefined coiling geometry of tested spiral passage, the geometrical coefficients of \( d/D \) and \( \eta \) are accordingly fixed by the experimental test rig. Therefore the relative strengths of centrifugal and torsional forces, which are respectively quantified by \( Dn \) and \( Gn \) in a coiling passage, are proportional to the Reynolds number. The local Nusselt number in a static spiral passage could be reduced as the function of Reynolds number. Due to the three-dimensionality of the flow field produced in the coiled tube, the unknown function \( \phi \) in Eq. (1) is dependent on the circumferential and axial locations on the tube surface. In the following, the heat transfer measurements along the inner and outer edges of the test tube are referred by the \( I \) and \( O \) superscripts, respectively. However the experimental data with zero buoyancy interaction (\( Bu=0, \omega=0 \)) are not feasible to be directly generated by the heat transfer experiments because a real fluid has non-zero beta value. In such a limiting case having a vanishing small wall-to-fluid temperature difference but the test duct reciprocates, the heat transfer function \( \phi \) in Eq. (1) shall reflect the combined \( Re/Dn \) and \( Pr \) effects 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/Dean/Germano or pulsating number with different wall-to-fluid temperature differences into the zero buoyancy asymptote. For a set of boundary conditions simulated by the heat transfer test module, such as the geometrical features of the reciprocating surface 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 further analysis.

2.2 Facilities

The smooth-walled spiral test tube was fitted on a reciprocating test facility, which has been previously described in Ref. (15). A brief description of the reciprocating test facility and an illustration of the test section now follow. Figure 2 shows the layout of the test facilities and the airflow loop featuring the main components and instrumentation. The pressurized air in tank (1) fed continuously by a compressor (2) is channeled into the spiral test tube (3) through a dryer (4), a pressure regulator and filter (5), a pressure transducer and Tokyo Keisyo TF-1120 mass flow meter (6) and a needle valve (7) for the control of mass flow rate. The nominal through flow Reynolds and Dean numbers were 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 spiral heating passage (3), the coolant flows through a heated straight flow calming section (8) with an equivalent
length of 20 tube diameter, in which the flow could be well developed. System reciprocation is created by a crank-wheel mechanism driven by a 2,500 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 test section (3) was made of a 5260 mm long, 0.5 mm thick stainless steel tube of 20 mm inner tube diameter, with a coil diameter of 264 mm and a pitch of 20 mm. The coil-to-tube diameter ratio was 13.2. This configuration requires Reynolds number above 8,758 to ensure the turbulent flow in the smooth-walled coils(9). A straight flow calming section (8) was made in front of the coils. Also shown in Fig. 2 is the helical coordinate system. The origin of the coordinate system, O, is defined at the bottom of the first coil where corresponds to the juncture of the straight flow calming section (8) and the coiled section. For the convenience of engineering application, the turning angle of helix in respect to the central axis of coil, φ, which relates with the curve length along the center of helical pipe, z, is obtained as $\phi = \frac{z}{132.038}$ for the present coiling configuration.

Two electrical terminals were respectively connected at the starting point of the straight section (8) and the exit of the coils for feeding the heating power into the test module (3). Because the test tube was heated directly by passing an electrical current through the spiral tube, a reasonably uniform heat flux boundary condition was generated. An adjustable DC power supply unit that allowed control of the heating power provided the required heating current. Adjusting the heating power supply varies the relative strength of the overall buoyancy level at any fixed flow condition. The complete test module (3) was wrapped and insulated with fiberglass as shown in Fig. 2 to minimize the external heat loss. Seventeen K-type thermocouples were mounted along each of the two opposite inner and outer helix diameters with a regular interval. The measured curve length covered the first 1.286 coils. For the present coiled test section, the coolant-flows in the streamwise regions of $2.267 < Z < 18.681$ ($19.48^\circ < \phi < 162^\circ$), $21.676 < Z < 41.156$ ($188^\circ < \phi < 357^\circ$) and $44.367 < Z < 53.8$ ($385^\circ < \phi < 463^\circ$) correspond respectively to the upward, downward and upward flow regions. Two additional thermocouples penetrated into the core of the flow path to measure the fluid temperatures at entry and exit planes of the coil. The temperature measurement at the flow entrance was based to evaluate the fluid temperature in the coils using an enthalpy balance method. All the temperature measurements were monitored and stored on an IBM 33 PC through a Trend-Link Fluke Hydra 2620 A-100 data logger for subsequent data processing. Note that, for this test geometry, the flow with Reynolds number in the range of $2.300 - 8.758$ became the developed turbulent flow at the end of the straight flow calming section (8) upon entering the spiral section; but gradually transited into laminar flow in the coils. As the turbulent-to-laminar transition in the coils reduces the cooling performance, the heat transfer within this particular flow regime was investigated.

2.3 Data reduction

The definition of local Nusselt number was based on the wall to fluid bulk temperature difference ($T_w - T_f$). Justified by the very thin and uniform thickness of the tube wall (0.5 mm), the wall temperature, $T_w$, at the fluid/wall interface was corrected from the measurement using the one dimensional conduction law. However, it is not practical to measure the streamwise fluid temperature variation without disturbing the flow field. Therefore the fluid temperature measured at the flow entrance was used as the starting reference for a sequential integration of the local enthalpy to calculate the intermediate values of fluid bulk mean temperatures. The local convective heat required for the accountancy of local enthalpy balance was obtained by subtracting the net conductive heat and the external heat loss from the total heating flux. To estimate the net wall conductive heat at the measurement spot, the measured wall temperature distribution was numerically interpolated and a simple finite difference representation for the Fourier conduction law was used. The characteristics of external heat loss were evaluated based on the results of a series of static and reciprocating heat loss calibration runs. It is worth noting that, even in the time-dependent system when the test duct reciprocated, the local fluid bulk mean temperatures were defined by the same method of sequential integration of the local entherialy. To verify the above calculating process, each calculated fluid bulk temperature at the exit plane of the coils for any flow condition tested 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 bulk temperatures at the exit plane of the coils was less than ±8%. With the local fluid bulk temperature to be defined, the local coolant properties such as $\rho$, $C_p$, $k$, and $\mu$ were then evaluated by means of standard polynomial functions using the calculated fluid bulk temperature as the determined
variable. The mass flow rate measurements, reciprocating frequency, temperature data, along with the local fluid properties were incorporated with other measurements such as the convective heat flux to calculate the local non-dimensional groups in Eq. (1).

2.4 Program
Initially the instrumentation and data reduction were checked out with a series of baseline heat transfer experiments conducted at the static conditions. This was followed by a series of reciprocating experiments, which data were compared with the baseline results to examine the individual and combined effects of $Re$, $Dn$, $Pu$ and $Bu$ on heat transfer. Both the static and reciprocating experiments were conducted in the Reynolds number range of $4500 - 7000$ to allow for the turbulent-to-laminar transition in the developing flow region of the coils. The reciprocating data were generated at fixed Reynolds numbers. At each fixed Reynolds number, five sets of tests at the reciprocating frequencies of 0, 0.833, 1.25, 1.67 and 2 Hz were performed. The data generated in this manner revealed the effect of pulsating number on heat transfer. The ranges of corresponding Dean, Germaino and pulsating numbers over which the experiments have been performed were $1.050 - 1.600$, $8.217 - 12.782$ and $0.132 - 0.428$, respectively. By varying the tube wall heat flux systematically, a detailed examination of the buoyancy number effect could be performed. For each selected Reynolds number, Dean number and pulsating number pair tested, five different levels of heating power raised the tube-wall temperature values at the axial location of $54Z$ to $60^\circ C$, $80^\circ C$, $100^\circ C$, $110^\circ C$ and $130^\circ C$. The coolant entry temperature to the test tube was typically in the range of $30 - 45^\circ C$, and this in conjunction with the other specified experimental conditions, permitted tests to be undertaken with buoyancy numbers in the range of $0.000335 - 0.000938$. For each individual test, the flow and heating level were fixed for about 30 min in order to assure the quasi-steady flow condition. Once such a flow state was reached, the on-line data acquisition system collected and stored the instantaneous data for a period of 10 seconds. These data were subsequently time-averaged and processed into the dimensionless groups defined in Eq. (1).

An uncertainty approximation of the data reduction was conducted. Because the quasi-steady state of the reciprocating flow system was approximated when variations of the local time-averaged wall temperatures were in the range of $\pm 0.5^\circ C$, the maximum uncertainty in temperature measurement was estimated to be $\pm 0.5^\circ C$. These temperature measurements were the major sources to attribute uncertainty. With the temperature difference between the wall and fluid varied from 28 to 79°C, the maximum uncertainty for the Nusselt, Reynolds, Dean, Garmano, pulsating and buoyancy numbers were about 15%, 4.6%, 5%, 5.2%, 1.2% and 5.3%, respectively.

3. Results and Discussion
3.1 Non-reciprocating results
The typical streamwise Nusselt number distributions along each of the two opposite inner and outer helix diameters of the static coils are illustrated in Fig. 3 using the results obtained with nominal Reynolds number of 7,000 as an illustrative example. The data trends revealed in Fig. 3 are representative for all the static results. In Fig. 3, the present data is also compared with the equivalent Dittus-Boelter Nusselt number value, $Ntu$, in order to reveal the combined effect of centrifugal force and turbulent-to-laminar transition on the heat transfer in the developing flow region. At the axial location of 2.267, the Nusselt number values on the inner and outer surfaces are almost identical. There is no noticeable circumferential heat transfer variation near the entrance of the coiled section since the Dean vortices at the immediate flow entrance are not well established. Note that the present Nusselt number values at the axial location of 2.267 agree well with $Ntu$. This agreement confirms the developed turbulent flow condition in the straight calming tube before the coolant entering the coiled section. In the axial range of $Z > 2.267$, the heat transfer differences between the inner and outer edges are obvious. After flow traveling 2.267Z, the effect of Dean vortices on heat transfer is already evident. Except at the first axial location on the outer surface, among which the cross-plane secondary flows are not well established, the streamwise heat transfer variations along the inner and outer edges follow the trend of exponential decay. The asymptotic Nusselt,

![Fig. 3 Typical axial distributions of heat transfer along inner and outer edges of static coils](image-url)

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number value for the present data resolves near the axial location of 54 tube-diameter. This asymptotic heat transfer value is in good agreement with the correlation proposed by Xin and Ebadian for the developed coil flow. The pattern of axial heat transfer variation in the developing flow region \( Z < 54 \) is a net result of the turbulent to laminar transition and the developments of Dean vortices and boundary layers. Along the outer edge of the coils, the heat transfer levels are consistently higher than the Dittus–Boelter value, \( Nu_{\infty} \), although the process of turbulent to laminar transition proceeds in the coils that gradually stabilizes the flow and undermines the heat transfer performance. The heat transfer improvement on the outer surface form the straight pipe level is mainly attributed from the cross-plane flow circulations. The centrifugal force induced in the coiled tube drives the cooler fluid toward the outer surface from the central core. The circumferential secondary flows wash the heated wall and joint at the spot near the inner surface under the process of turbulent to laminar transition. Therefore, the considerable heat transfer reduction from the Dittus–Boelter value, \( Nu_{\infty} \), is observed along the inner edge of the coiled tube. It is also noticed that a minor degree of gradual streamwise increase of \( Nu_{\infty} \) at the outer edge takes place in the flow region of \( 44.367 \leq Z \leq 53.4 \) where the coolant is driven upward. As the gravitation field respectively accelerates and decelerates the flow in the downward and upward flow regions, the streamwise heat transfer increase in the region of \( 44.367 \leq Z \leq 53.4 \) after the considerable upstream turbulent-to-laminar transition could be a result of the decelerating effect provided by gravity. In this respect, a similar experimental observation has been reported, for which the decelerating gravity effect in the upward flow region that moderated the radial velocity distribution was considered as the main cause for such heat transfer increment. In the static coiled tube, the variation of heating flux resulted in a very thin Nusselt number spread at each axial location. Thus the gravitation-driven buoyancy effect in the present parametric range did not cause a considerable influence on heat transfer in the static coils. Because there was neither noticeable change in the coolant Prandtl number nor the considerable buoyancy interaction for the range of temperatures covered by the experimental program in the static coils, the local Nusselt number along the inner and outer edges of the coiled tube were correlated in the form of

\[
Nu_{\infty}^{0.9} = A(Z) \times Re^{B(Z)} \tag{2}
\]

Coefficients \( A \) and \( B \) are functions of axial location, \( Z \). The streamwise variations of coefficients \( A \) and \( B \) reflect the developing nature of the flow in the coils with regards to the developments of boundary layer, Dean vortices, and the turbulent to laminar transition. It is worth noting that, the dimensionless parameters \( Dn \) and \( Gn \) are respectively quantified as \( Re \sqrt{d/D} \) and \( Re \eta \) for which the values of \( \sqrt{d/D} \) and \( \eta \) remain constant for a given test geometry. Therefore the effects of centrifugal and torsional forces are absorbed in both coefficients \( A \) and \( B \) of Eq. (2). In this particular flow regime, it is interesting to note that the static Nusselt number, \( Nu_{\infty}^{0.9} \), can be well correlated by \( Re^{0.50} \) for the entire range of Reynolds, Dean and Germaino numbers and the heat flux levels tested. A physical implication of the constant \( B \) value is the unmodified convective inertia force effect on heat transfer, even if the flow undergoes the turbulent to laminar transition in the developing flow regime. However, the effect of flow development on the heat convection is reflected in the streamwise variation of coefficient \( A \). Except at the first axial location on the outer surface of the coiled tube, the coefficient \( A \) is well correlated by the following equations

\[
A^{0}(Z) = 0.1795 + 0.071074e^{-0.009172Z} 2.267 \leq Z \leq 54 \tag{3}
\]

\[
A^{0}(Z) = 0.2101 + 0.3213e^{-0.12533Z} 5.838 \leq Z \leq 54 \tag{4}
\]

Over the entire range of Reynolds, Dean and Germaino numbers and the heat flux levels studied, 95% of the present experimental Nusselt numbers were correlated by Eqs. (3) and (4) within ±9%. The streamwise variations of coefficient \( A \) that reflect the developing nature of the flow could also be well described by the exponential decay function (see Eqs. 3 and 4). It is therefore felt that the streamwise developments of Dean vortices and turbulent to laminar transition in the coiled tube follow the same asymptotic nature with the boundary layer development.

A comparison of the correlated coefficients in Eqs. (3) and (4) could reveal the different heat transfer features on the inner and outer surfaces in the static coils. In the developed flow region, the coefficient \( A^{0}(0.2101) \) is always greater than \( A^{0}(0.1795) \) that indicates the higher developed Nusselt number values along the outer edge. Also noticed that the coefficient \( A^{0} \) decays in a faster rate than \( A^{1} \) due to the larger value of power index for the exponential function in Eq. (4), which is correlated from the data collected from the outer surface. This finding implicitly demonstrates that the flow along the outer edge of the coils undergoes a faster rate of turbulent to laminar transition than that on the inner surface. Nevertheless, since the Prandtl number effects were absorbed into the coefficients of correlations and were not included as a parameter in the correlations that were developed, the results of this study are essentially limited to dry air. Having estab-
lished the non-reciprocating heat transfer datum and the correlation equations, the comparative difference of heat transfer in the static and reciprocating coiled tube can be examined as follows.

3.2 Reciprocating effects in general

When the coiled tube reciprocates, the possibility of bulk-flow pulsation and the modified flow structure and vorticity could similarly trigger a dynamic process for the time-wise variations of the secondary flow cells developed in the coiled tube as that found in the artery of human heart\(^{(13)}\). The streamwise developments of Dean's and Lyne's\(^{(21)}\) vortices and the turbulent to laminar transition in the reciprocating coiled tube are thus expected to be affected. However, this could produce the attendant temporal Nusselt number variations along the reciprocating surface. 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 inner and outer helix diameters were obtained. In this regard, the streamwise Nusselt number variations obtained with a nominal Reynolds number of 4 500 at reciprocating frequencies of 0, 0.833, 1.25, 1.67, and 2 Hz are compared in Fig. 4 to illustrate the general reciprocating effects on heat transfer. Note that, because the variation of pulsating number is made by adjusting the reciprocating frequency at a fixed Reynolds number rather than by varying the Reynolds number at a fixed frequency, the differences in Nusselt number values between the static and reciprocating results shown in Fig. 4 are caused by the combined pulsating and buoyancy effect. As shown in Figs. (4-a), (4-b) and (4-c) when the reciprocating frequency increases from 0.833 to 1.67 Hz, the overall heat transfer levels along the inner and outer surfaces gradually decrease with the increase of pulsating number. It is interesting to note that the reciprocating Nusselt number levels at the first axial spot of inner and outer surfaces shown in Fig. (4-a) are higher than the Dittus–Boelter value at the reciprocating frequency of 0.833 Hz, but to be gradually reduced when the reciprocating frequency increases from 0.88 to 1.67 Hz. Because the straight flow calming section also reciprocates with the coiled section, the reduction of Nusselt number value from the Dittus–Boelter level when the reciprocating frequency increases from 0.833 to 1.67 Hz at the immediate flow entrance suggests the local flow stabilization when the pulsating number increases from 0.16 to 0.32. The further increase of the reciprocating frequency to 2 Hz, which increases the pulsating number to 0.38 as showed in Fig. (4-d), causes a subsequent heat transfer recovery along the entire inner and outer surfaces from the results depicted in Figs. (4-c) and (4-b). Thus, when the pulsating number gradually increases, the overall heat transfer in the reciprocating coils 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.

Relative to the static heat transfer references, the pattern of streamwise heat transfer distribution is considerably modified. The heat transfers in the static coils shown by the dotted lines follow the typical exponential decay described in Eq. (2.1). As a result of Dean vortices effects, the considerable heat transfer differences between the inner and outer surfaces in the static coils are soon established after \(Z>2.267\). However, the apparent heat transfer differences between the inner and outer surfaces in the reciprocating coils initiate after coolant travels about 28Z. This result has led to the consideration of delayed development of Dean vortices in the reciprocating coils. Justified by the streamwise increasing trends of heat transfer in the flow region of \(Z>28\) shown in Fig. 4, the flow could be developing in the region of \(Z\approx53.4\) when the coils reciprocate. Thus the heat transfer correlation for reciprocating flow generated by the

Fig. 4 Axial distributions of reciprocating Nusselt number along inner and outer Edges at Reynolds number of 4 700 (\(Re=1\times10^4, Gr=8.58\))
present study, which will be illustrated in the section of parametrical presentation, is applicable for the developing flow. Relative to the static results, the heat transfer reductions from the static heat transfer levels in the axial range of \( Z \leq 28 \) are observed in Fig. 4, especially for the data collected from the reciprocating outer surface. As a result of reciprocating effects in general, the local heat transfer is improved from the static heat transfer level after the flow travels about 28 tube diameters. Therefore the typical streamwise heat transfer variation in the reciprocating coils is different from the exponential decay pattern found in the static coils. Although the above described heat transfer deviations from the static reference share the same feature along the inner and outer edges, but the larger degree of heat transfer modification occur on the outer surface.

Also worth noting that the data shown in Fig. 4 initially increase from the first point and then decrease until \( Z = 28 \), which resembles the behavior observed in the static coils. In the static coils, such streamwise reduction of heat transfer has been previously reported as a result of turbulent-to-laminar transition\(^{(6)} \), which phenomenon is similarly observed for the present reciprocating results in the flow region of \( 5.83 \leq Z \leq 28 \). As the Reynolds number of 4700 provides turbulent flow in the straight smooth-walled pipe but is less than the Reynolds number required for turbulent flow in the static coils\(^{(6)} \), it is felt that there is the possibility of turbulent-to-laminar transition in the reciprocating coils which causes the streamwise heat transfer variations in the region of \( Z \leq 28 \) showed in Fig. 4. As described previously that the streamwise development of outer-to-inner Nusselt number difference has been delayed in the reciprocating coils, the considerable heat transfer reductions observed along the outer helix in the flow region of \( Z \leq 28 \) could be the combined effects of the reciprocating forces, the delayed development of Dean vortices and the transition of turbulent to laminar flow.

Also demonstrated in Fig. 4 is the importance of the buoyancy effect on heat transfer in the reciprocating coiled tube. The increase of heat flux causes the increase of buoyancy number at the fixed Reynolds, Dean, Germano, and pulsating numbers, which produces the various degrees of data spreads as shown in each plot of Fig. 4. This manner of data presentation shows the reciprocating buoyancy effect in isolation. In spite of the results depicted in Fig. (4-a) where the reciprocating buoyancy level is too weak to provide the appreciable heat transfer effect; the local heat transfer is consistently increased when the buoyancy number increases at the fixed Reynolds and pulsating numbers. This is indicated by the upward data spread driven by the increase of buoyancy number as illustrated in Figs. (4-b), (4-c) and (4-d). A relative large data spread, driven by the buoyancy variation is consistently found at the outer edge, reflecting large buoyancy effect on this surface.

### 3.3 Parametrical presentation

Figure 4 has demonstrated the different heat transfer modifications when the pulsating and buoyancy numbers vary at the fixed value of Reynolds number, thereby confirming the interactive effect between the pulsating and buoyancy forces in the reciprocating coils. In order to quantify the reciprocating effect on heat transfer, the relative Nusselt number, \( \frac{N_u}{N_{u0}} \), is defined by normalizing the reciprocating Nusselt number, \( N_u \), with the equivalent static Nusselt number, \( N_{u0} \), obtained at the same Reynolds number. Because the reciprocating buoyancy number, \( Bu \), involves the pulsating term and the buoyancy parameter, \( \beta (T_w - T_e) \), in its mathematic structure, it is felt that there may be a possibility of collapsing all the experimental data obtained at a fixed axial location onto a tight data trend by plotting the relative Nusselt number against the buoyancy number.

The data plot constructing in this manner is useful for providing an overall parametrical review of the relative heat transfer changes from the static conditions. Figure 5 shows the variation of relative

![Fig. 5 Reciprocating effects on heat transfer](image-url)
Nusselt number, \( \frac{Nu}{Nu_0} \), with buoyancy number, \( Bu \), for the entire data generated at several axial locations. As seen, there is a tendency for all the data to collapse into a tight band in each plot. The relative Nusselt numbers on the inner and outer surfaces could be, respectively, reduced to the values about 80% and 65% of the static tube levels at the axial location of 5.83%Z, even with the buoyancy interaction present that enhances the heat transfer in the reciprocating coils. At the axial location of 54%Z, the subsequent heat transfer improvement after a range of heat transfer impediment when \( Bu \) increases leads the relative Nusselt numbers to the values about 1.42 and 1.6 at the buoyancy number of 0.07 on the inner and outer edges, respectively. The data trends revealed in Fig. 5 are the results of the combined pulsating and buoyancy force effects, which effects are the axial-location dependent. The streamwise variation of the reciprocating effects, which is revealed by the systematic variation of the data trends showed from Figs. (5-a) to (5-e) and from (5-f) to (5-j), is in consistent with the streamwise flow developments illustrated by Fig. 4.

In Fig. 5, it is noticed that all the relative Nusselt numbers fall into unity when the buoyancy number, \( Bu \), becomes zero. The results with \( \frac{Nu}{Nu_0} = 1 \) actually reflect the static conditions because all the heat transfer data with \( Bu = 0 \) were created by the zero reciprocation (\( \omega = 0 \)) with a finite wall-to-fluid temperature difference. Therefore the parametrical presentation of the reciprocating data in this manner suffered from a physical confusion as regard to the uncoupling of the individual effects of reciprocating and buoyancy forces. In the limiting case of a condition having a vanishing small wall-to-fluid temperature difference, although the coiled tube still reciprocates, Fig.5 suggests that \( \frac{Nu}{Nu_0} = 1 \) when \( Bu = 0 \), which represents no change in the relative forced convection from the static level. This is not physically true. Therefore it is not suitable to correlate the local relative Nusselt number, \( \frac{Nu}{Nu_0} \), with the single variable, \( Bu \). As the real fluids have non-zero \( \beta \) values, it is not feasible to conduct the heat transfer tests at the zero buoyancy number with various combinations of pulsating, Reynolds, Dean and Germano numbers when the coiled tube reciprocates. Alternatively, the zero-buoyancy reciprocating effect may be inferred by extrapolating a family of heat transfer data taken at a specific pulsating number with different wall-to-fluid temperature differences. In practice, the relative Nusselt number data obtained with five different heating levels at a constant pulsating number could be extrapolated into the zero buoyancy asymptote. This is demonstrated in Fig. 6 where the experimental relative Nusselt number data are plotted against the buoyancy number at the axial location of 21.67%Z. The evidence of this extrapolating process along with the extrapolated zero buoyancy data at four different pulsating numbers are also illustrated and compared in Fig. 6. As shown, the asymptotic relative Nusselt numbers at the zero buoyancy condition are initially reduced but to be recovered when the pulsating number increases from 0.17 to 0.23. As this heat transfer characteristic typifies all the reciprocating results, the initial impediment in the relative heat transfer, which is followed by the subsequent heat transfer recovery, is caused by the individual pulsating force effect. Justified by the data trend at any specific pulsating number in the parametric range tested, the relative Nusselt number is reasonably well correlated by the equation with the linear version of

\[
\frac{Nu}{Nu_0} = \xi[Pu, Z] + \xi[Pu, Z] \times Bu
\]

The function \( \xi \) in Eq. (5) shall describe the relative heat transfer changes caused by the pulsating force effect alone. With the condition of \( Pu \neq 0 \) and \( Bu = 0 \),
the variation of relative Nusselt number is defined by \( \xi \) function. When the pulsating number becomes zero, which automatically sets the buoyancy number to zero, a physical constraint that requires the value of \( \xi \) function to be unity can force the relative Nusselt number into unity, which removes the static forced convection solution in Eq. (5). The lines shown in Fig. 6 are the linear regressions of the experimental data. As seen, the slopes of these correlating lines, which are the values of \( \zeta \) function in Eq. (5), consistently reduce with the increase of pulsating number. Physically, the reduced \( \xi \) function at the higher value of pulsating number suggests the weakened buoyancy effect when the pulsating number increases. Using a series of cross plots based on Fig. 6 but applied to all the axial locations examined, it interpolated a series of curves that correlated the values of \( \xi \) and \( \zeta \) functions at all the pulsating numbers tested. Figures 7 and 8, respectively, show the typical example of the manner in which the functional values of \( \xi \) and \( \zeta \) vary with the pulsating number. The \( \xi \) value showed in Fig. 7 illustrates the variation of relative Nusselt number at the zero buoyancy condition with the pulsating number. Each curve of \( \xi \) functions on the inner or outer surfaces could be treated as the reference datum from which the buoyancy interaction initiates. Also combined in Fig. 7 with the \( \xi \) functional values are the measured reciprocating heat transfer data that reflect the combined pulsating and buoyancy force effects on heat transfer. As the buoyancy interaction within the tested parametric range improves heat transfer, the data spread from the zero-buoyancy reference datum (\( \xi \) function curve) is driven upward with the increase of buoyancy number as shown in Fig. 7. The correlating curve of \( \xi \) value shown in Fig. 8 follows the exponential decay that agrees with the weakened buoyancy effect at the higher value of pulsating number. However, a detailed examination of the data trends for all the axial location versions of Figs. 7 and 8 suggest that the values of \( \xi \) and \( \zeta \) functions could be reasonably correlated by the equations of

\[
\xi = f_s(Z) \times Pu + f_o(Z) \times Pu + 1 \quad (6)
\]

\[
\zeta = g_s(Z) \times e^{u(Z) - u_o(Z)} \times Pu \quad (7)
\]

In Eq. (6), \( \xi \) value is virtually unity when \( Pu \) becomes zero, which satisfies the zero reciprocating forced convection limiting case. The numerically determined curve fits for the \( f_s \) and \( g_s \) coefficients involved in Eqs. (6) and (7) are summarized in Table 1 and 2 respectively.

The attempts to reveal the heat transfer physics and uncouple the pulsating and buoyancy force effects in the reciprocating coiled tube through the experimental program with a detailed data analysis have been illustrated. A net result is the empirical correlation of Eq. (5). As the centrifugal and torsional force effects have influenced the heat transfer data generated in the static and reciprocating phases of experiments, the influences of centrifugal and torsional forces on heat transfer in the present spiral coil tested are implicitly involved in the numerical coefficients.
Table 1   Functional values of $f_1(Z)$ and $g_1(Z)$

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Table 2   Functional values of $g_2(Z)$ and $g_3(Z)$

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depicted in Eq. (2) and summarized in Tables 1 and 2. The overall success of this proposal is indicated in Fig. 9 where the entire experimental data are compared with the correlative predictions evaluated by Eq. (5). The maximum discrepancy of ±20% between the experimental and correlative results is achieved for 93% and 91% of the entire data generated from the inner and outer surfaces, respectively. With the similar test ranges of pulsating and buoyancy numbers, the ranges of variation for the relative Nusselt number along the inner and outer edges are 0.7 ± 1.33 and 0.51 ± 1.52 as shown in Figs. (9-a) and (9-b) respectively. The greater influences of reciprocation on the outer edge heat transfer than its inner counterpart are demonstrated. Owing to the complex interactions between the Dean's and Lyne's vortices, the pulsating and buoyancy forces and the transition of turbulent to laminar flow in the developing region of the reciprocating coiled tube, it becomes a formidable task to predict the local heat transfer. However, justified by the achieved accuracy, Eq. (5) could provide the reasonable heat transfer evaluation to assist the design initiative of the cooling networks featured in Fig. 1.

4. Conclusions

This experimental program investigates the heat transfer in a reciprocating coiled tube along the inner and outer edges. 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 pulsating and buoyancy forces on heat transfer. In conclusion the following salient points emerge from this study.

1. In the static coiled tube, the considerable centrifugal effects generate cross-stream Dean vortices and initiates a streamwise turbulent-to-laminar transition in the Reynolds number range of 4 500 - 7 000. These effects result in the circumferential heat transfer variation with relative high heat transfer on the outer edge compared with that on the inner edge, which level is reduced from the Dittus-Boelter value. A set of empirical correlations is derived for the local static Nusselt number along the inner and outer edges.

2. The extrapolating zero-buoyancy results reveal the individual pulsating 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 improves heat transfer. These two interactive and combined force effects in the reciprocating coiled tube could cause the relative Nusselt numbers on the
inner and outer surfaces to be, respectively, reduced to the values about 80% and 65% of the static tube levels. Such heat transfer impediment has to be particularly considered when the spiral cooling passage 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 forced convection, pulsating force and reciprocating buoyancy in the coils to be taken in to account.

Acknowledgements

This work was financially supported by National Science Council, Taiwan, Republic of China, towards National Kaohsiung Institute of Marine Technology under grant number, NSC-89-2212-E-022-002.

References