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Temporal and spatial distribution of heat flux in oscillating flow subjected to an axial temperature gradient

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Abstract—The temporal and spatial distribution of heat flux within counter-oscillating slugs of fluid, along which is maintained a constant axial temperature gradient, is examined. It is found that the resultant axial heat flux pulsates at twice the base oscillation frequency and that the time-averaged axial heat flow under tuned conditions is orders of magnitude larger than that present in the absence of oscillations. Such thermal pumping is produced by the time-dependent interaction of a transverse conduction flux, produced by large transverse temperature gradients, with the periodic axial fluid motion.

1. INTRODUCTION

IN SEVERAL recent articles [1–3] we have examined the characteristics of a novel heat transfer mode in which heat is transported from a hot to a cold fluid reservoir by means of sinusoidal oscillations of a viscous fluid contained within open-ended capillary tubes connecting the reservoirs. In this transfer process (related to the process of gas exchange in high frequency pulmonary ventilation [4]) there is a periodic conduction heat transfer between the fluid core and the boundary-layer region of the oscillating flow coupled to a periodic axial convective heat transport. However, unlike existing convective heat transfer methods, this thermal pumping technique involves no net convective mass transfer [5] and hence should find considerable application in areas where one wishes to remove heat at high rates but does not want the accompanying exchange of mass (i.e. cooling of radioactive liquids). Also, since devices using this transfer process are driven by external oscillations, one can consider devices based on the technique as thermal valves; these might find applications in the field of cryogenics. Heat transfer rates in excess of those achievable with standard heat pipes [6] are readily obtainable by this thermal pumping process. The latest experimental results using water in capillary tubes have yielded effective axial heat conduction rates 1.97×10^5 times those due to axial molecular conduction at the same axial temperature gradient [7].

It is the purpose of this paper to examine the temporal and spatial distribution of heat flux in the thermal pumping process during the various stages of the fluid oscillation cycle. This work supplements our earlier analytical and experimental studies [1–3] where attention was confined to only time-averaged conditions, so that details of the combined conduction and convection heat transfer process were not evident. To simplify the calculations we will confine our attention to counter-oscillating slug flows bounded by outer non-conducting walls. The resultant geometry avoids the need to deal with the rather cumbersome expressions involving multiple derivatives of Kelvin functions, which arise when dealing with problems of this kind in cylindrical geometries [2, 8]. As will be shown, the heat transfer process consists of time-dependent transverse conduction coupled to a periodic axial convective transport. The net effect of this cyclic interactive process is to transport large quantities of heat from the hot to the cold ends of the oscillating fluid slugs. This heat flow increases with both oscillation frequency and amplitude. The heat transfer process is shown to be tunable and under optimum conditions can yield transverse temperature gradients in excess of 10^6 K m^{-1} during part of the cycle. Time averaged axial heat flows in excess of 10^8 W m^{-2} are readily achievable with liquids such as water at a frequency of 10 Hz, an axial displacement of 0.2 m and an axial temperature gradient of 200 K m^{-1}.

2. VELOCITY AND TEMPERATURE DISTRIBUTION

We consider the oscillating flow shown schematically in Fig. 1. It consists of two fluid slugs each of width a undergoing 180° out-of-phase axial sinusoidal oscillations. The maximum difference in parallel displacement of the two slugs is Δx (termed the tidal displacement) and the corresponding anti-symmetric velocity profile is

$$U(y,t) = U_0\cos\omega t = U_0[e^{i\omega t}]_R \qquad (1)$$

in the range $0 < y < a$ and the negative of this for $-a < y < 0$. Here ω is the angular frequency, y the transverse coordinate, t the time and the subscript R indicates the real part of the function shown. A con-

NOMENCLATURE

a	fluid slug width
c	specific heat
$\mathscr{F}(\beta, \eta)$	temperature function shown in Fig. 2
g	transverse dependence of the temperature distribution
G, H, I	functions of β occurring in $\dot{Q}$
k	thermal conductivity
L	fluid slug length ($L \gg a$)
Pe	Péclet number, $U_0 a/\kappa$
$\dot{Q}$	transverse conduction heat flux at $y = 0$
t	time
T	temperature field
T_H, T_C	temperature of hot and cold fluid reservoir
ΔT	temperature difference between $\eta = 1$ and $\eta = 0$
U	axial velocity
U_0	maximum axial velocity
x	axial coordinate
Δx	tidal displacement
y	transverse coordinate.

Greek symbols

β	frequency parameter, $a\sqrt{\omega/\kappa}$
γ	axial temperature gradient, $-(T_H - T_C)/L = \partial T/\partial x$
η	non-dimensional transverse coordinate, y/a
κ	thermal diffusivity
κ_{eff}	effective axial thermal diffusivity
λ	non-dimensional effective thermal diffusivity
ρ	fluid density
τ	non-dimensional time, $t\kappa/a^2$
ω	angular frequency of oscillating fluid.

Subscripts

I	imaginary part of the function shown
R	real part of the function shown.

stant axial temperature gradient $\gamma = -(T_H - T_C)/L$ is superimposed on the fluid, where T_H and T_C are hot and cold reservoir temperatures maintained at the large axial distances of $x = -L/2$ and $x = +L/2$, respectively, with L being the channel length. The fluid exiting the channel at $\pm L/2$ is assumed to undergo rapid turbulent mixing in the fluid reservoirs found there. The tidal displacement Δx is always assumed smaller than $L/2$ so that no direct convective mass transfer is possible between the reservoirs under the laminar flow conditions assumed to exist in the narrow channel. The thermal conditions imposed on the problem are that the normal derivative of temperature at $y = \pm a$ is zero. This can be interpreted to mean either that the flow is bounded by insulating walls or that one is dealing with a periodic array of counter-oscillating slug flows of width $2a$ each [3]. Note, that since we are concerned here only with

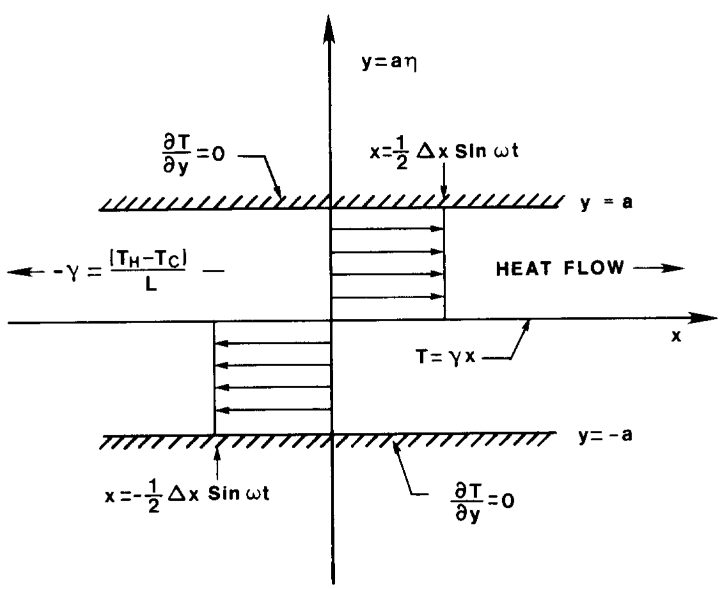

FIG. 1. Counter-oscillating slug flow geometry.

slug type flows, the Womersley number dependence involving the ratio of oscillatory inertia to viscous forces does not enter into the problem. This simplifies the calculations considerably, while at the same time retaining the major characteristics of the thermal pumping process. It will also be recognized that the present flow geometry is unstable to Kelvin–Helmholtz type of disturbances. In experiments this difficulty could be overcome by replacing the fluid by counter-oscillating solid conducting bars insulated at $y = \pm a$, or by placing a thin, rigid, conducting sheet at $y = 0$ to prevent fluid mixing.

The temperature distribution within the range $0 < y < +a$ is given by the temperature equation

$$\frac{\partial T}{\partial \tau} + a\,Pe\,e^{i\beta^2\tau}\frac{\partial T}{\partial x} = \left(a^2\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial \eta^2}\right) \qquad (2)$$

where $\eta = y/a$ is the non-dimensional transverse distance; $Pe = U_0 a/\kappa = \omega a \Delta x/2\kappa$ the Péclet number, κ the thermal diffusivity, $\tau = t\kappa/a^2$ the non-dimensional time based on the transverse thermal diffusion time, and $\beta = a\sqrt{\omega/\kappa}$ a measure of the magnitude of the thermal diffusion time to the oscillation period. The parameter β is also equal to the product of the Womersley number and the square root of the fluid Prandtl number and, in this form, appears in a multiple time scale analysis used in one of our earlier studies [2]. The boundary conditions in the η direction appropriate for the geometry considered are

$$\frac{\partial T(1,x,t)}{\partial \eta} = 0, \quad T(0,x,t) = 0. \qquad (3)$$

An analytic solution of equation (2), subject to boundary conditions (3), can be found when using the approximation

$$T(\eta,x,t) = \gamma[x + a g(\eta)\,e^{i\omega t}]_R \qquad (4)$$

first proposed by Chatwin [9] and used in the fluid dispersion study of Watson [8] and in two of our own earlier investigations [1, 3]. The basis for approximation (4) is already contained in the physical assumptions made by Taylor [10] in a study of dispersion in steady, laminar pipe flow, and is based on the fact that the axial gradient $\partial T/\partial x$ is small compared to the much larger time-dependent transverse temperature gradient existing during most of the oscillation cycle. This means that $\partial^2 T/\partial^2 y$ is much larger than $\partial^2 T/\partial x^2$ while the value of $\partial T/\partial x$ is taken to equal the time-averaged value γ. With this approximation, the temperature equation (2) assumes the simplified form

$$\frac{d^2 g}{d\eta^2} - i\beta g = Pe \qquad (5)$$

which, in view of boundary conditions (3), has the exact solution

$$g(\eta) = \frac{iPe}{\beta^2}[1 - \mathscr{F}(\beta,\eta)] \qquad (6)$$

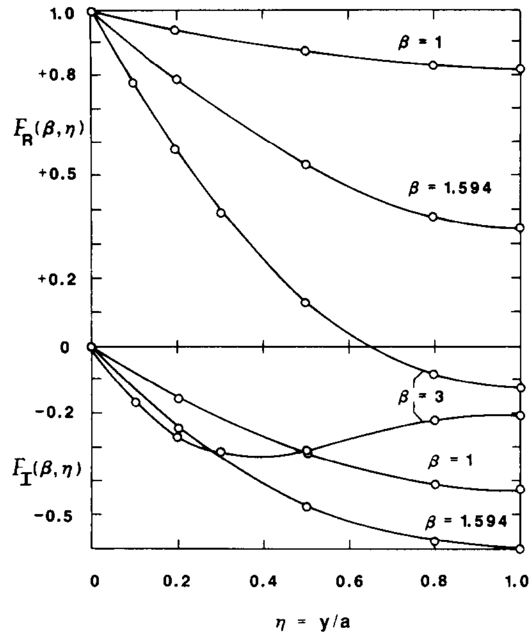

FIG. 2. Temperature function $\mathscr{F}(\beta,\eta)$ for three values of the frequency parameter β.

where

$$\mathscr{F}(\beta,\eta) = \mathscr{F}_R(\beta,\eta) + i\mathscr{F}_I(\beta,\eta) = \frac{\cosh\sqrt{i\beta}(1-\eta)}{\cosh\sqrt{i\beta}}$$

$$= \left[\cosh\frac{\beta}{\sqrt{2}}(1-\eta)\cos\frac{\beta}{\sqrt{2}}(1-\eta)\right] + i\left[\sinh\frac{\beta}{\sqrt{2}}(1-\eta)\sin\frac{\beta}{\sqrt{2}}(1-\eta)\right] \Big/ \left[\cosh\frac{\beta}{\sqrt{2}}\cos\frac{\beta}{\sqrt{2}} + i\sinh\frac{\beta}{\sqrt{2}}\sin\frac{\beta}{\sqrt{2}}\right]. \qquad (7)$$

Plots of the real part and imaginary part of the function $\mathscr{F}(\beta,\eta)$ for three different values of β are given in Fig. 2. The value $\beta = 1.594$ corresponds to the optimum tuned condition for the flow under consideration as will be made clear below, while $\beta = 1$ and $\beta = 3$ correspond to low and high frequency conditions, respectively.

3. EFFECTIVE AXIAL THERMAL CONDUCTIVITY

To determine the axial flow of heat in the present configuration, we equate the total axial conduction plus convection heat flux to an effective thermal conductivity $\rho c \kappa_{eff}$ multiplied by the axial temperature gradient γ. Here ρ is the fluid density and c the specific heat. Mathematically one has

$$\kappa_{eff}\gamma = \kappa\gamma - \frac{1}{a}\int_0^a [U]_R[T]_R\,dy \qquad (8)$$

which can, with the aid of (1), (4) and (6), be rewritten in the convenient form

$$\lambda = \frac{(\kappa_{\text{eff}} - \kappa)}{\omega \Delta x^2} = -\frac{1}{4}\int_0^1 [(2x/\Delta x)\cos\omega t + (1/2)(\mathscr{F}_R - 1)\sin 2\omega t + \mathscr{F}_I \cos^2\omega t]\,d\eta. \quad (9)$$

In most instances the product of the integral in this last expression times $\omega\Delta x^2$ is much larger than the molecular thermal diffusivity κ, so that the term κ in the definition of λ can in most instances be neglected. Note, as first observed by Chatwin [9] in a related study, that the non-dimensional effective thermal diffusivity λ is a time-dependent function containing terms oscillating at the fundamental driving frequency ω plus one harmonic term at twice the fundamental frequency. If one time averages over one period of the fluid oscillation the $\cos\omega t$ term and the $\sin 2\omega t$ term average out to zero, but $\cos^2\omega t$ assumes the value of 1/2. The resultant time-averaged value of λ thus assumes the finite value

$$\lambda = \frac{i}{16}\int_0^1 [\mathscr{F} - \bar{\mathscr{F}}]\,d\eta = \frac{1}{8\beta}[\sqrt{i}\tanh\sqrt{i}\beta]_R \quad (10)$$

where the bar indicates the complex conjugate of the function shown. An evaluation of this last result yields

$$\lambda = [\sinh\sqrt{2}\beta - \sin\sqrt{2}\beta]/8\sqrt{2}\beta \times [\cosh\sqrt{2}\beta + \cos\sqrt{2}\beta]. \quad (11)$$

A graph of this function is shown in Fig. 3. We see there that the curve of λ vs β has a single maximum. This maximum occurs at $\beta = 1.59394$ with a corresponding value of $\lambda_{max} = 0.052153$. On either side of this maximum there is a steep drop-off in the value of λ. The asymptotic values of λ as derived from equation (11) are

$$\lambda = \frac{\beta^2}{24}\left[1 - \frac{19}{120}\beta^4\right], \quad \beta \ll 1 \quad (12)$$

and

$$\lambda = \frac{1}{8\sqrt{2}\beta}\left[1 - 2\sqrt{2}\sin\left(\sqrt{2}\beta + \frac{\pi}{4}\right)e^{-\sqrt{2}\beta}\right], \quad \beta \gg 1. \quad (13)$$

These formulas indicate that $(\kappa_{\text{eff}} - \kappa)$ is proportional to ω^2 at low β and proportional to $\omega^{1/2}$ at large β. This way of representing the time-averaged axial dispersion coefficient clearly shows that, if the oscillation frequency is fixed, there is a unique value for the width a at which a maximum in axial heat flow can be expected. This width equals

$$a = 1.594\sqrt{\kappa/\omega}. \quad (14)$$

For the present configuration with a fluid thermal diffusivity equal to that of copper, namely, $\kappa = 1.12 \times 10^{-4}\ \text{m}^2\ \text{s}^{-1}$, and a frequency of 10 Hz, this would imply a width of $a = 0.697 \times 10^{-2}$ m. This value at optimum tuning represents conditions where the time of transverse heat diffusion and the oscillation half period of the flow are approximately equal. Also

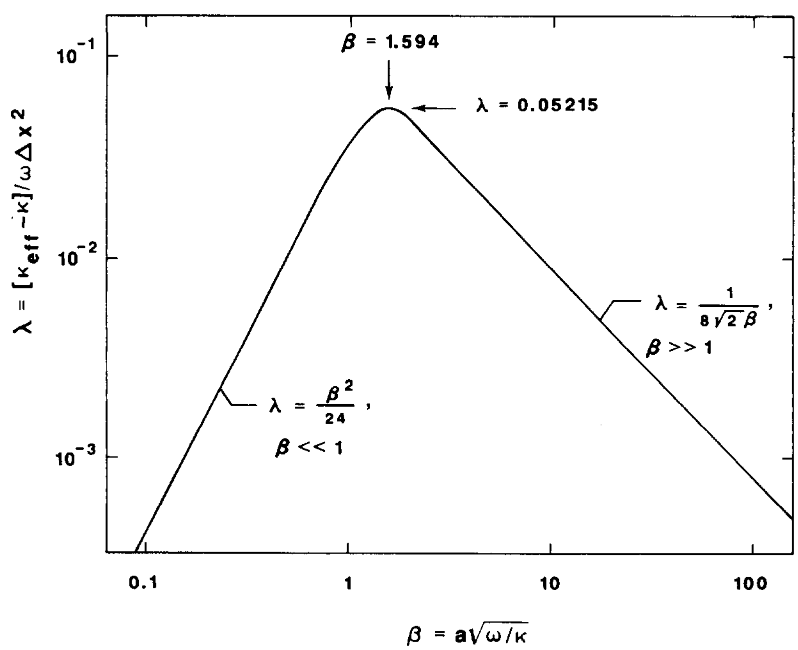

FIG. 3. Effective thermal diffusivity as a function of β.

one notes that at the optimum point, the effective thermal diffusivity is proportional to the product of angular frequency ω and the square of the tidal displacement Δx, indicating that one must not only choose the right channel width in order to maximize the axial heat transfer but that one must also make the frequency and tidal displacement as large as possible.

4. TRANSVERSE AND AXIAL THERMAL FLUX

To understand better what is happening in this thermal pumping process, we next examine the temporal dependence of both the transverse and axial heat flux existing in the fluid. For this purpose we first consider the value of the instantaneous temperature difference between the interface at $\eta = 1$ and $\eta = 0$. This value is obtained by substituting equation (6) into equation (4) and recognizing that γx represents the time-independent temperature at $\eta = 0$. Explicitly one finds

$$\Delta T = 1/2\gamma\Delta x\{\mathscr{F}_{\mathrm{I}}(\beta,1)\cos\omega t + [\mathscr{F}_{\mathrm{R}}(\beta,1)-1]\sin\omega t\} \tag{15}$$

with the real and imaginary values of $\mathscr{F}$ given by equation (7) or Fig. 2. A plot of this result for the three representative values $\beta = 1$, 1.594 and 3 are shown in Fig. 4. Our choice for these particular values of β was dictated by our time-averaged result which indicates that these values correspond, respectively, to a low, a tuned, and a high fluid oscillation frequency for a fixed value of a. Note that, since $\gamma < 0$ in the present problem, the fluid at $\eta = 0$ is generally colder than that at $\eta = 1$ during the first half of the cycle while it is generally hotter during the second half of the cycle. Since transverse conduction heat flow moves in a direction opposite to the temperature gradient, it is then clear (see Fig. 1) that transverse conduction heat flow will be in the positive transverse direction during most of the period for which the lower fluid slug $(-1 < \eta < 0)$ is positioned to the right of the upper fluid slug. It will be in the negative transverse direction when the reverse holds true. The temperature difference in the transverse direction is observed to increase with increasing β but will not go much above the value of $-2\Delta T/\gamma\Delta x = 1$ which would be expected as the oscillating fluid becomes a non-conductor (i.e. as $\beta \to \infty$). The reason that the transverse temperature difference becomes small as β decreases toward zero is that the oscillation period is then very slow compared to transverse conduction time, so that a large temperature difference cannot maintain itself because of conduction losses.

The corresponding conduction heat flux in the transverse direction during the various phases of the oscillation, can be readily calculated using the standard Fourier law. At the interface $\eta = 0$ and at the axial position $x = 0$, it is found, after some manipulations, to be

$$\begin{aligned}\dot{Q} &= -k\left.\frac{\partial T}{\partial y}\right|_{y=0} \\ &= -\gamma\frac{k\Delta x}{2a}\left\{\frac{\mathrm{d}\mathscr{F}_{\mathrm{I}}(\beta,0)}{\mathrm{d}\eta}\cos\omega t + \frac{\mathrm{d}\mathscr{F}_{\mathrm{R}}(\beta,0)}{\mathrm{d}\eta}\sin\omega t\right\} \\ &= -\frac{\gamma k Pe}{\beta}\left\{\frac{G(\beta)\cos[\omega t+(3\pi/4)]+H(\beta)\sin[\omega t+(3\pi/4)]}{I(\beta)}\right\}\end{aligned} \tag{16}$$

where

$$G(\beta) = 1-\mathrm{e}^{-2\sqrt{2}\beta} \tag{17}$$

$$H(\beta) = -2\sin\sqrt{2}\beta\,\mathrm{e}^{-\sqrt{2}\beta} \tag{18}$$

and

$$I(\beta) = 1+2\mathrm{e}^{-\sqrt{2}\beta}\cos\sqrt{2}\beta+\mathrm{e}^{-2\sqrt{2}\beta}. \tag{19}$$

A plot of this transverse conduction heat flux is given in Fig. 5 for our three representative values of β. The non-dimensional value of $2a\dot{Q}/k\gamma\Delta x$ is seen to be positive during the early part of the cycle and negative during the later stage, with a general increase in value of the heat flux noted as β increases. These results are consistent with the instantaneous transverse temperature gradient existing in the flow. An indication of the magnitude of this transverse heat flow and corresponding temperature gradient can be obtained by considering the special case for water where $k = 1.4\times10^{-7}$ m^2 s^{-1} with $\gamma = -200$ K m^{-1}, $a = 2\times10^{-4}$ m, $\Delta x = 10$ cm, and at an oscillation frequency of 5 Hz. This condition corresponds approximately to $\beta = 3$ and, according to Fig. 5, gives a maximum transverse conduction heat flow of $\dot{Q} = -3.2\times10^5$ W m^{-2} at $\omega t = 5\pi/16$. The transverse temperature gradient at $\eta = 0$ at this instant is 1.9×10^5 K m^{-1}, representing a 950-fold increase over that existing axially. As long as the flow remains laminar, there will be no convective heat flux in the transverse direction. Such laminar conditions are expected to hold for most fluids near tuned conditions in view of the relatively small values of a required there. The very large transverse gradients existing in these flows is what makes the large axial heat transfer possible.

In addition to the transverse conduction heat flow there will be an axial convective transport which can be expected to be a function of β, $\gamma\omega\Delta x^2$, ωt and also be dependent on the transverse coordinate η. Its spatially integrated value is given by equation (9) and

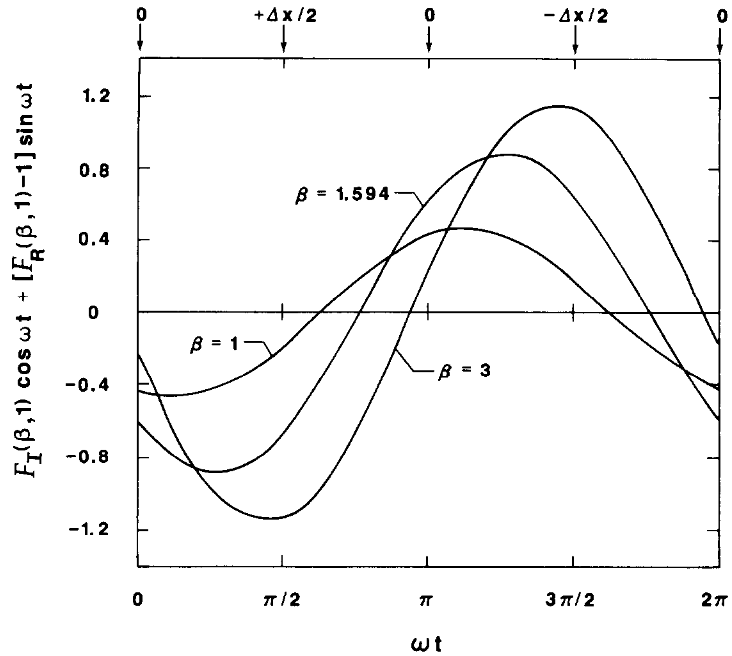

FIG. 4. Transverse temperature difference.

its incremental value at $\eta = 1$ is

$$-U(1,t)[T(1,x,t)-T(0,x,t)] = -\frac{1}{4}\gamma\omega\Delta x^2 \times\{\mathscr{F}_I(\beta,1)\cos^2\omega t + \frac{1}{2}[\mathscr{F}_R(\beta,1)-1]\sin 2\omega t\}. \quad (20)$$

A plot of this equation is given in Fig. 6 for the same three values of β used earlier. Note that this axial thermal flux is composed of the sum of a periodic function of angular frequency ω and one of 2ω. This leads to curves which repeat themselves twice during each oscillation period as clearly indicated in the figure. Each of the three curves show that the axial convective heat transfer is pulsating at twice the base frequency with more heat transported on average in the positive x direction than in the negative x direction. Note that, at the tuned condition of $\beta = 1.594$, the time integrated heat flux going to the cold side of the fluid slugs minus that moving to the hot side is larger than in either the $\beta = 1$ or $\beta = 3$ cases. Also it is observed that as β gets small, the heat being con-

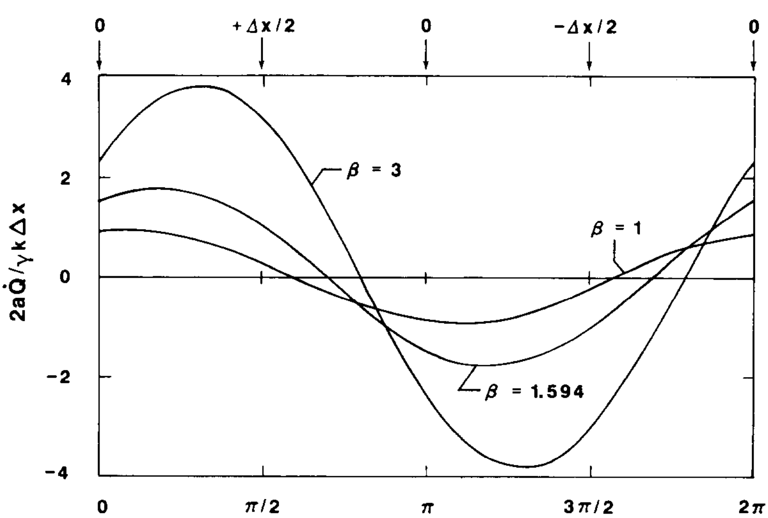

FIG. 5. Conduction heat transfer across the interface at $\eta = 0$.

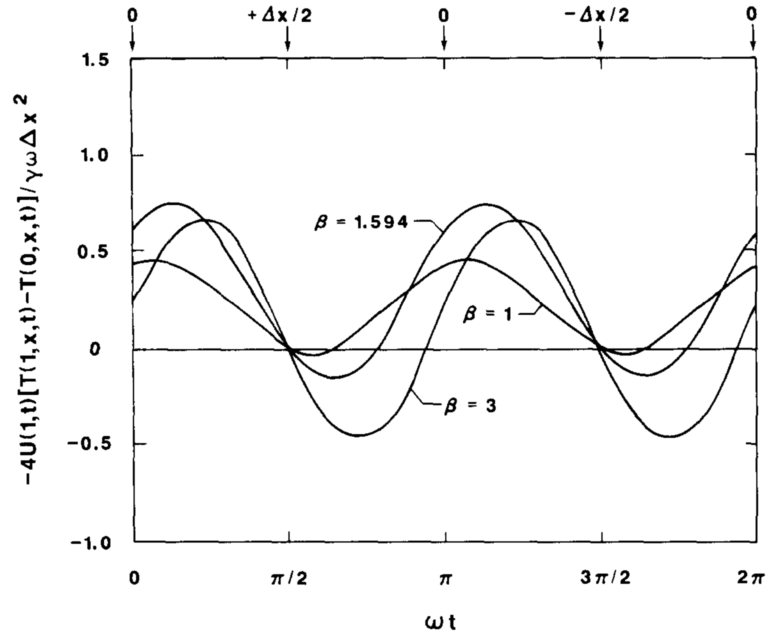

FIG. 6. Axial convective heat flux at $\eta = 1$. Note the double frequency structure of the axial heat flow pulsations.

vected in the positive x direction is much larger than that going in the reverse direction, however, the overall magnitude of the net positive axial thermal flux remains less than that under tuned conditions.

As a final calculation, we have evaluated the transverse spatial dependence of the axial convective flux at time intervals of $\omega t = \pi/4$ in the range $0 \leqslant \eta \leqslant 1$. The calculations involve the evaluation of equation (20) after replacing the values of $\mathscr{F}(\beta, 1)$ found there by the values $\mathscr{F}(\beta, \eta)$ as given in Fig. 2. Results of such an evaluation are shown in Fig. 7 for the tuned condition at $\beta = 1.594$. Also included there is the negative of the spatial transverse temperature variation obtainable from equation (15), by again replacing $\mathscr{F}(\beta, 1)$ by $\mathscr{F}(\beta, \eta)$. One sees from this last figure that the positive axial flux reaches a maximum near

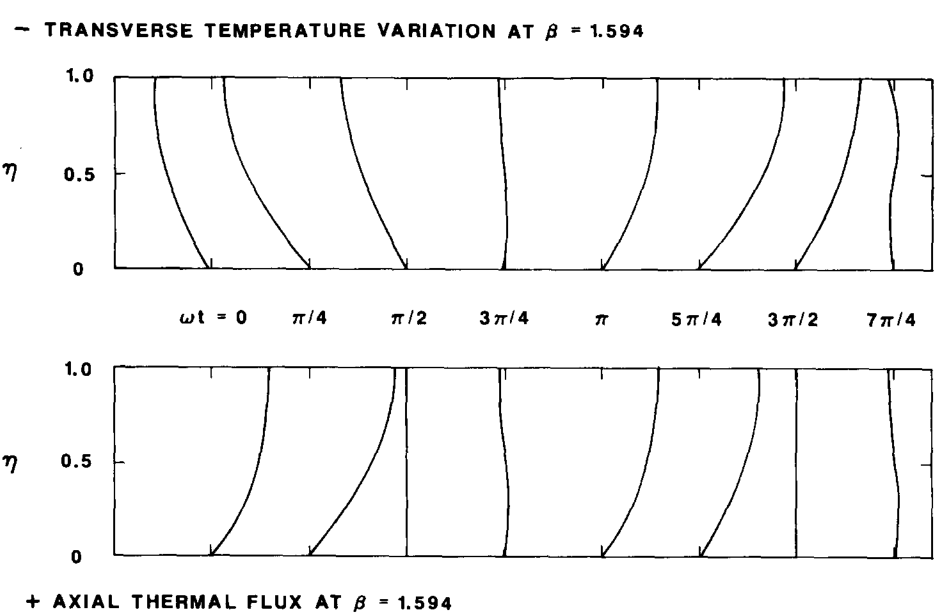

FIG. 7. Spatial variation of the axial convection flux and the negative of the transverse temperature variation at $\pi/4$ intervals of the oscillation cycle.

the times $\omega t = \pi/4$ and $5\pi/4$ and disappears at $\omega t = \pi/2$ and $3\pi/2$. There is only very little negative axial flux present and this occurs near $\omega t = 3\pi/4$ and $7\pi/4$.

5. DISCUSSION AND CONCLUDING REMARKS

We have examined the temporal and spatial distribution of thermal flux within a simplified version of the thermal pump [5] and have shown that under tuned conditions large time-averaged axial heat flows are possible in such a device. The mechanism responsible for the axial heat transport without a net convective mass transfer is a cyclical process in which transverse heat conduction interacts with axial convective transport to produce a large pulsating axial heat flow which, when time-averaged, is in a direction opposite to the imposed axial temperature gradient γ. The transverse conduction heat flows are large in such a device since the transverse temperature gradient can become very large and the surface area across which heat is conducted is considerably larger than in the absence of a tidal displacement. Experimentally one can measure the predicted time integrated axial heat flux resulting from such an oscillatory heat transfer process [1], but would probably have some difficulty in trying to time resolve the predicted double periodic axial heat flux oscillation given by equation (20). To detect such a variation will require a thermocouple capable of responding to frequencies in the 10–50 Hz range, and in a way which does not disturb the oscillatory flow.

Extensions of the present study would involve replacing the simple slug flow profile given by equation (1) by the more complicated hyperbolic cosine form as used for example in ref. [3]. This will produce some small changes from those presented here, but will not alter the basic results found here that thermal pumping is produced by the time-dependent interaction of transverse conduction flux with axial convection heat transport. The process involves no net convective axial mass flux although some mass diffusion is possible, becoming larger for lower Lewis number fluids. Finally, it would be interesting to see if the sinusoidal displacement function used here maximizes the heat transfer process or if a square wave or some other time variation would still further improve the already high axial heat transfer rates.

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DISTRIBUTION SPATIALE ET TEMPORELLE DU FLUX THERMIQUE DANS UN ECOULEMENT OSCILLANT SOUMIS A UN GRADIENT DE TEMPERATURE AXIAL

Résumé—On examine la distribution spatiale et temporelle du flux de chaleur dans des mouvements oscillants de fluide avec un gradient axial, constant de température. On trouve que le flux thermique axial résultant est pulsé à une fréquence double de celle des oscillations de base et que le flux de chaleur axial moyenné sur le temps est supérieur de plusieurs ordres de grandeur à celui trouvé en l'absence d'oscillations. Un tel pompage thermique est produit par l'interaction, dépendant du temps, d'un flux transversal de conduction, produit par des gradients de température transverses, avec le mouvement périodique axial du fluide.

ZEITLICHE UND RÄUMLICHE VERTEILUNG DER WÄRMESTROMDICHTE IN OSZILLIERENDER STRÖMUNG BEI EINEM AXIALEN TEMPERATURGRADIENTEN

Zusammenfassung—Untersucht wird die zeitliche und räumliche Verteilung der Wärmestromdichte innerhalb gegenschwingender Flüssigkeitspfropfen, entlang derer ein konstanter axialer Temperaturgradient aufrechterhalten wird. Die sich ergebende axiale Wärmestromdichte pulsiert mit der doppelten Frequenz wie die Schwingungen. Der zeitliche Mittelwert des axialen Wärmestroms ist unter bestimmten Bedingungen mehrere Größenordnungen höher als beim Fehlen der Schwingungen. Solch ein thermisches Pulsieren wird durch eine zeitabhängige Wechselwirkung zwischen querverlaufendem Wärmestrom durch Leitung, verursacht durch große querverlaufende Temperaturgradienten, und der periodischen axialen Fluidströmung erzeugt.

ПРОСТРАНСТВЕННО-ВРЕМЕННОЕ РАСПРЕДЕЛЕНИЕ ТЕПЛОВОГО ПОТОКА С УЧЕТОМ ОСЕВОГО ТЕМПЕРАТУРНОГО ГРАДИЕНТА ПРИ ОСЦИЛЛИРУЮЩЕМ ТЕЧЕНИИ

Аннотация—Исследуется пространственно-временное распределение теплового потока внутри встречных движущихся и осциллирующих снарядах жидкости, вдоль которых создан постоянный осевой температурный градиент. Найдено, что результирующий осевой тепловой поток пульсирует с частотой, равной удвоенной основной частоте, а осредненный по времени осевой тепловой поток при таких условиях на несколько порядков больше потока при отсутствии колебаний. Подобная тепловая накачка обеспечивается временным взаимодействием поперечного кондуктивного потока, созданного большими поперечными температурными градиентами, с периодическим осевым движением жидкости.