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International Journal of Heat and Mass Transfer 43 (2000) 3197–3207

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Transient measurements of the heat transfer coefficient in unsteady, turbulent pipe flow

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Received 9 October 1998; received in revised form 28 September 1999

Abstract

The 'transient' response of the heat transfer coefficient is investigated in turbulent and relaminarising, unsteady flow. We find that the response to sinusoidal perturbations collapse on a single parameter, ω^+, and we use this scaling to develop an empirical model for predicting the response to non-sinusoidal changes. The model works well under fully turbulent conditions but fails when the flow relaminarises. We find no evidence of any mechanism that would lead to heat transfer enhancement over and beyond a quasi-steady effect. On the contrary, relaminarisation appears to decrease the average heat transfer in some unsteady flows. © 2000 Published by Elsevier Science Ltd. All rights reserved.

Keywords: Unsteady flow; Convective heat transfer; Relaminarisation; Unsteady heat transfer

1. Introduction

Heat transfer in unsteady pipe flow is a feature of a variety of engineering devices. Better understanding of the mechanisms could assist with the design of such practically important devices as automotive engines and pulse combustors [1,2]. Furthermore, it has been suggested that flow unsteadiness could increase the effectiveness of heat exchangers [3,4]. Better understanding would help to evaluate the usefulness of unsteadiness as a performance enhancement mechanism.

Previous studies have sought to determine whether the average heat transfer is enhanced, reduced or entirely unaffected by flow unsteadiness. It seems that there is little effect when the amplitude of the unsteady component is small compared to the average flow rate [5–7]. However, when the amplitude is sufficiently large for flow reversal to occur, the enhancement has been reported to be both less [6,8] and more [2,8] than that predicted from a simple quasi-steady model. By making time resolved measurements of the transient heat transfer in unsteady flow we gain some insight into the mechanisms that cause the change. That is the basis of the work reported here.

Transient heat transfer is interesting not only for the understanding it gives us of the mean heat transfer problem but also in its own right, where it might find application, for example, in the design of inlet manifolds in automotive engines [9,10]. The quantity of interest is the unsteady heat flux at the wall, which depends on the variation of the heat transfer coefficient and of the bulk temperature. These two variables have very different characteristics; the heat transfer coefficient is essentially a local parameter that depends mainly on the flow conditions while the bulk temperature depends on the entire thermal history of the flow

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PII: S0017-9310(99)00305-1

Nomenclature

a	constant in Eq. (3)
h	heat transfer coefficient
$j(t^+)$	scaled time domain model for the response of the heat transfer coefficient
$J(\omega^+)$	scaled frequency domain model for the response of the heat transfer coefficient
Nu	Nusselt number
Pr	Prandtl number
q	heat flux
R	radius
Re	Reynolds number
So	Stokes number
t	time
t^+	scaled time, tu_*^2/ν
$t_{\tilde{h}}^+$	scaled time constant
T	temperature
u_*	friction velocity, $\sqrt{\tau/\rho}$
U	bulk velocity

Greek symbols

ρ	parameter in Eq. (5)
ν	kinematic viscosity
ρ	density
τ	shear stress
ϕ	phase
ω	angular frequency
ω^+	scaled frequency parameter, $\omega\nu/u_*^2$

Subscripts

b	bulk quantity
c	centreline
w	wall

Character modifiers

–	time mean part
˜	deterministic unsteady part
+	non-dimensional quantity
⟨ ⟩	ensemble average
| |	absolute value or complex modulus

[11,12]. We will concentrate on the heat transfer coefficient.

Initially, we consider the effect of small sinusoidal oscillations superimposed on an otherwise steady, fully turbulent flow. Arguably this flow is too simplistic for our results to find direct application in real devices but it is a useful starting point to investigate scales and behaviour. We present experimental measurements of the variation of the heat transfer coefficient and discuss the processes responsible. Many unsteady flows of practical interest (such as those in automotive systems or the human body) do not vary sinusoidally. In an attempt to analyse such flows, we develop a linear model that predicts the variation of the heat transfer coefficient, for an arbitrary wave shape. To test the model we investigate the response of the heat transfer coefficient to small step changes of the flow rate and to brief pulses of slightly increased flow. In these experiments, we see evidence of a secondary transient heat transfer phenomenon, directly correlated to changes of the pressure gradient. This has the counter-intuitive effect of causing the heat flux to fall in response to a velocity increase. Finally, we examine the process of relaminarisation, which affects both the predictions of the transient model and the mean heat flux.

2. Experimental rig

The flow we investigated was of air through a 38 mm diameter, 1.5 m long steel pipe with 6.35 mm wall thickness. The pipe was heated with resistance wire wrapped in a spiral fashion around it, with sufficiently fine pitch for the heat to be well diffused at the inner wall. A centrifugal fan generated a steady airflow, which was measured using a laminar flow meter at the fan inlet. The steady flow Reynolds number could be varied up to approximately 40,000. A mechanical filter and a trip ring were located just before the heated length to clean the air of impurities that could have affected the transducers and to ensure that the flow was fully turbulent at low Reynolds numbers.

A number of transducers were mounted in short removable length of pipe located at the pipe exit. This test section is illustrated in Fig. 1. It was constructed in two halves so it could be removed easily to modify or repair the transducers. Care was taken to ensure that the joint between the test section and the main pipe was smooth, so as not to disrupt the flow. This section was heated separately from the main length, with short lengths of heater wire soldered to the outside of the pipe. The two heaters were controlled independently, using proportional integral differential controllers. The wall temperatures at the test section and near the exit of the main heated length, were both set to approximately 80°C.

The velocity and temperature were measured using a hot-wire anemometer and a cold-wire resistance thermometer, respectively. The hot-wire signal was corrected to account for the air temperature variation at the post-processing stage. Both probes could be traversed over the pipe radius but for the experiments we

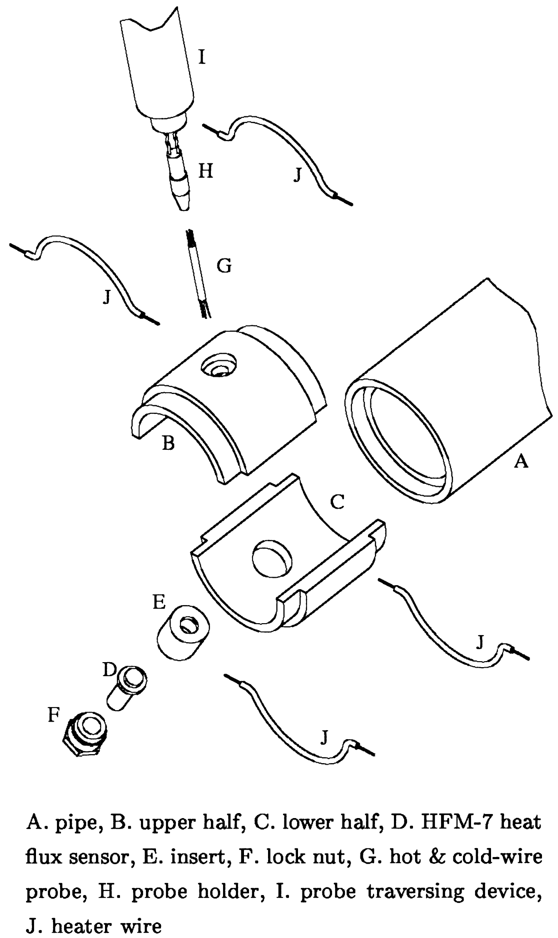

Fig. 1. Exploded view of the test section, showing the mounting of the heat flux sensor and the hot- and cold-wire probe.

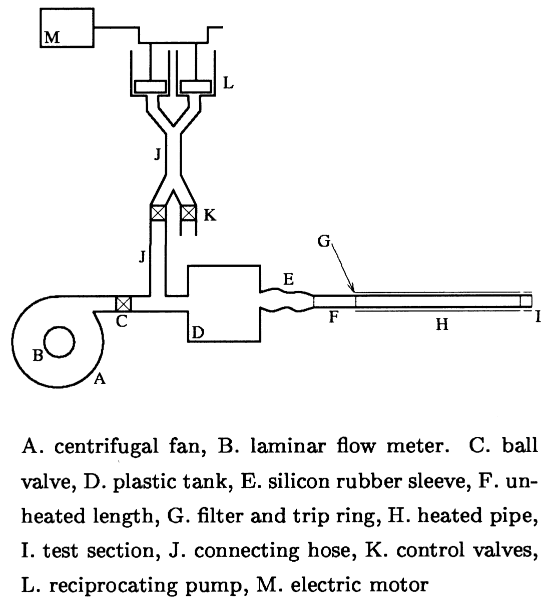

Fig. 2. Configuration of the rig used to generate sinusoidal flow rate variation.

describe here, only centreline measurements were made. The cold-wire was calibrated in position using a thermocouple. The absolute accuracy of the calibration was estimated to be correct to within a degree. The hot-wire was also calibrated in situ using the laminar flow meter. This was in turn calibrated using an orifice plate and a domestic gas flow meter. The difficulties of making measurements in heated flow meant that the uncertainty in the flow rate measurements is appreciable; perhaps as high as $\pm15\%$. However, the relative variation of the velocity and temperature are more important to us than their absolute values so the uncertainty in both measurements is considered acceptable. The frequency response of the hot- and cold-wire transducers were found to be approximately 2.5 and 2 kHz, respectively. While this response rate was no doubt too low to fully capture the spectrum of the turbulent fluctuations it was considerably greater than the frequency of the cyclic variation being measured (30 Hz with up to three harmonics in some cases). Consequently, we are confident of the accuracy of the phase-averaged results and additionally were able to get a fair impression of low frequency, turbulent fluctuations.

The heat flux at the wall was measured with a sensor based on the well-known principle of finding the temperature difference over a known thermal resistance. Our transducer (an HFM-7 micro-sensor manufactured by Vatell corporation [13]) is usual in that the response is much faster than most other commercial sensors. The sensor is manufactured using a thin film sputtering technique and is less than 2 microns thick. The manufacturer quotes a response time of 6 μs (167 kHz) and a sensitivity of approximately 170 $\mu V/(W/cm^2)$. The frequency response of the amplifier was considerably lower than this (approximately 2 kHz) so the overall response of this system was comparable to the hot and cold-wire sensors. The manufacturers calibration of the heat flux sensor was checked by ensuring that the predictions of well-known design correlations [14,15] could be reproduced in steady flow. The sensor also incorporated a resistance thermometer that we used to measure the wall temperature. A specially made insert was used to mount the sensor in the pipe wall and ensure that the surface around the sensor blended smoothly with the pipe wall.

All data signals were converted using a 12-bit analogue to digital card and logged using a personal com-

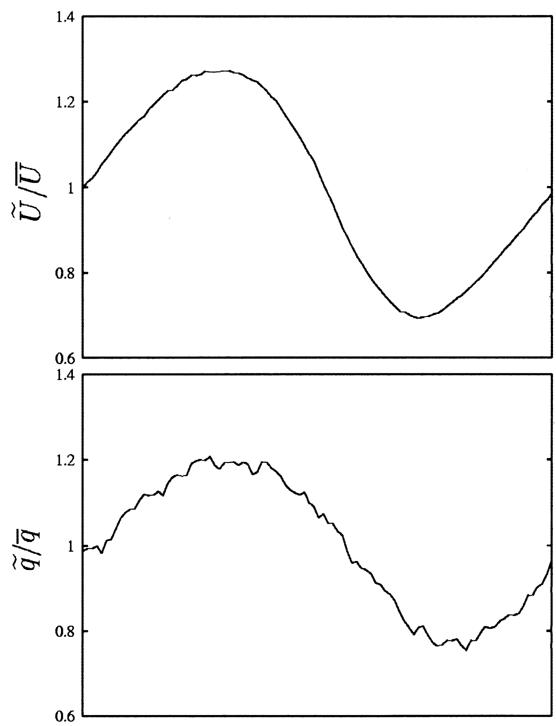

Fig. 3. Typical waveforms of the phase averaged centreline velocity and the heat flux at the wall.

puter. Further details about all aspects of the rig can be found in Ref. [12].

2.1. Experiments with sinusoidal oscillations

The flow circuit between the centrifugal fan and the heated pipe varied, depending on the unsteady flow considered. The configuration used to create sinusoidal variation is sketched in Fig. 2. The flow rate of the steady component was controlled with a ball valve located at the exit of the centrifugal fan and oscillations were generated using a reciprocating pump joined to the main pipe by a connecting hose. The pump was constructed using two cylinders of a 1.6 l automobile engine, driven with an electric motor. The frequency ranged from 0.5 to 30 Hz. The amplitude was controlled using two valves, one in the connecting hose and one discharging to the atmosphere. A large plastic tank located in the main line smoothed out higher harmonics of the waveform ensuring that the variation of the flow rate was close to sinusoidal. Fig. 3 shows a typical phase averaged waveform for both the velocity and heat flux. The tank was connected to the steel pipe with a silicon rubber sleeve to ensure that vibrations from the pump were not transmitted to the test section.

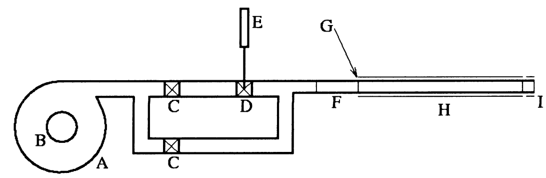

Fig. 4. Configuration of the rig used to generate step changes and brief increases of the flow rate.

2.2. Experiments with sudden changes

The configuration used to generate step and pulsed changes of the flow rate is illustrated in Fig. 4. The flow from the fan was split into two parallel branches. A pneumatic valve in one of the branches could be opened, closed or opened and immediately closed again to increase, decrease or briefly increase the flow rate, respectively. Two further valves, one in each branch, gave control over the initial flow rate and the magnitude of the velocity change.

3. Response to sinusoidal perturbations

In our initial experiments the time-mean Reynolds number ranged from 8,000 to 30,000, the frequency from 0.5 to 30 Hz and the amplitude of the oscillatory component of the bulk velocity from 8% to 20% of the steady component. In this parameter range the flow stayed fully turbulent throughout the unsteady cycle. The sampling time was dictated by the resolution required when the results were transformed to the frequency domain. The length of the samples was limited primarily by the storage and processing capabilities of our computer. The sampling frequency, 120 Hz, was too low to resolve the turbulent fluctuations but fast enough to resolve the regular unsteady component. At a minimum the fundamental and the second harmonic were resolved. As the signals were very close to sinusoidal in these experiments the signal content aliased from higher harmonics was negligible. Between three and ten repetitions were made at each flow condition.

3.1. Effect on the mean heat transfer

The effect on the mean heat transfer was determined by comparing the heat flux in steady flow to that in

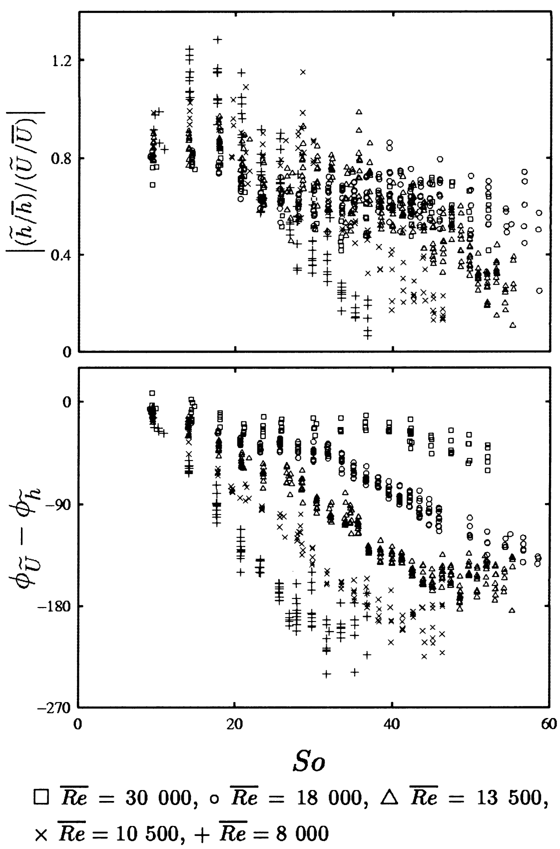

Fig. 5. The relationship between velocity and heat transfer coefficient variation in sinusoidally perturbed flow, scaled with the Stokes number.

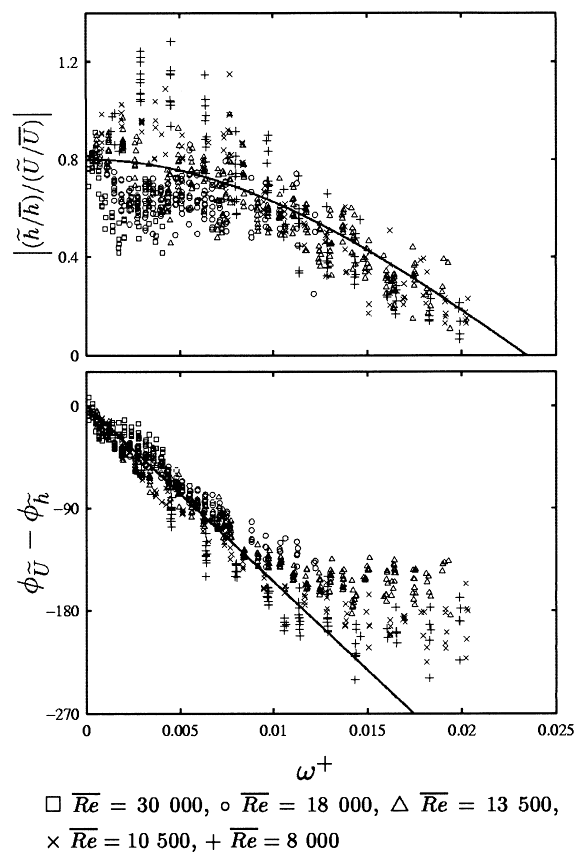

Fig. 6. The relationship between velocity and heat transfer coefficient variation in sinusoidally perturbed flow, scaled with ω^+.

runs with oscillations. In the parameter range of these experiments, the oscillations had no measurable effect on the average heat transfer, in line with the findings of other studies of heat transfer in similar flows [5–7]. Quantities such as the time-mean Reynolds stresses, velocity profile and wall shear stress have also been found unaffected by sinusoidal perturbations of a fully turbulent flow [16–18], so it would have been a surprise if the average heat flux was modified. However, this conclusion only holds for a limited parameter range; as we will see, the mean heat transfer is altered when the oscillation amplitude is increased.

3.2. Measurements of the transient response

The transient response is described by presenting the results in the form of a linear transfer function between the relative variation of the bulk velocity, $\tilde{U}/\bar{U}$, and the heat transfer coefficient, $\tilde{h}/\bar{h}$. The ratio of these two quantities has both a magnitude and a phase, which we wish to determine as functions of frequency and mean flow rate. Our assumption of linearity, which requires that an increased oscillation amplitude causes a proportional increase of the variation of the heat transfer coefficient, was found to be a good one in the limited amplitude range of these experiments.

As the heat transfer coefficient, defined as $h = q(T_w - T_b)$, cannot be measured directly, determining $\tilde{h}/\bar{h}$ requires that the variation of both the bulk temperature and the heat flux is known (the thermal inertia of the wall is large, so the wall temperature does not change significantly during the cycle). Unlike the velocity, however, the phase and magnitude of the temperature oscillations vary considerably with radius [12] so it is not straightforward to estimate the variation of the bulk temperature from the centreline measurement alone. It might have been possible to determine the bulk temperature variation by making measurements across the entire radius of the pipe, however, this would have involved a prohibitive increase in the experiments required for each flow condition. Fortunately, the relative temperature variation was found to be no more than a quarter of the relative variation of the heat flux, and usually much less [12], so as a first approximation, we neglected the variation of the bulk temperature and assumed $\tilde{h}/\bar{h} \approx \tilde{q}/\bar{q}$.

One frequency parameter commonly used to characterise unsteady flow is the Stokes number, $So = R\sqrt{\omega/\nu}$. We plot our data against the Stokes number in Fig. 5. From the phase plot, it is evident that the heat transfer coefficient lags the velocity by an increasing amount as the frequency increases and as the flow rate falls. In some cases the phase lag is as great as 180° implying that the heat transfer coefficient is at a maximum when the velocity is at it lowest, and vice versa. It is difficult to distinguish clear trends from the magnitude plot, as the collapse of data on this frequency parameter is poor.

Recently, a number of investigators have favoured the use of the scaled frequency, $\omega^+ = \omega\nu/u_*^2$ to characterise sinusoidally perturbed, turbulent flow [17,18]. They argue that this parameter has particular physical meaning as it relates the turbulent nature of the steady component, characterised by the thickness of the viscous sub-layer, to the thickness of an oscillatory shear wave penetrating into the flow. It can be used to consolidate data from studies of pipe, channel and boundary layer flows, and also to collapse data gathered at different flow rates [17,18].

The collapse of our own data on this frequency parameter is demonstrated in Fig. 6. The collapse of the phase curve is good, leaving only a slight effect of the flow rate. At very low ω^+ the magnitude appears to collapse around a value of 0.8. There is also reasonable collapse at high frequency where the magnitude of the variation is very small. We discuss below reasons for the scatter in the magnitude plot at intermediate frequencies, and the curve fit through the data.

3.3. The response mechanism

It is convenient to think of the velocity profile as being composed from the superposition of the profile of a steady flow at the mean Reynolds number and the profile of an oscillating component. For this model to be useful, we require that the oscillating component does not interact with the steady component but this assumption seems to be well supported by evidence that the mean velocity profile, shear stress and turbulence intensity are unaffected by perturbations in the parameter range of our experiments [16,17].

In the presence of an oscillating pressure gradient the slow moving fluid near the wall reacts relatively more than the more energetic core region. Consequently, the variation of the bulk flow rate is felt at the wall first and at the centreline last, giving the effect of a wave propagating from the wall into the flow. The resulting modification to the turbulence and velocity profiles gives rise to the modulation of the heat transfer coefficient. There appears to be reasonable agreement as to the shape of the unsteady velocity profile set up by this shear wave [16,17]. With $\omega^+ \lesssim 0.002$ the combined velocity profile is approximately quasi-steady. As the frequency increases, the shear wave does not propagate as far into the flow and when $\omega^+ \gtrsim 0.025$ the unsteady component of the velocity causes a 'quasi-laminar' shear layer to be set up within the viscous sub-layer near the wall [16,18]. Although we propose to use this illustrative model to explain the behaviour, we observe in our experiments, we do not suppose that there will always be well defined modulation of the turbulence, velocity profile or other flow characteristics during the course of a single period. In any single cycle, the turbulent fluctuations can bury the underlying order implied by the shear wave model. The assumed linearity of our model, therefore, does not apply within a cycle but rather to ensemble averaged quantities. We will return to this point, and illustrate it, when we consider the response to step changes.

At very low frequencies the magnitude of

$$\frac{\tilde{h}/\bar{h}}{\tilde{U}/\bar{U}}$$

appears to be concentrated around a value of 0.8 and the velocity and the heat transfer coefficient are in phase. In the quasi-steady case, we expect the Dittus–Boelter correlation,

$$Nu = 0.023 Re^{0.8} Pr^{0.4}, \tag{1}$$

will be approximately true at any particular instant so that $h(t) \alpha U(t)^{0.8}$. For small perturbations about a mean value this relationship can be linearised to get $\tilde{h}/\bar{h} = 0.8\tilde{U}/\bar{U}$, so the magnitude in the low frequency limit seems to be approximately that predicted by simple quasi-steady theory.

At high frequencies the oscillatory shear is contained in a thin viscous layer near the wall, so the unsteady velocity does not modulate the turbulence production. This limit is approached as ω^+ nears 0.025, with the result that the magnitude of the heat transfer coefficient variation falls. The scatter in the phase plot increases as it becomes harder to make a coherent measurement.

At intermediate frequencies, the behaviour is characterised by the lag of the heat transfer coefficient behind the velocity. As the lag results from the time taken for the shear wave to propagate from the wall to the turbulence producing regions, we expect it to depend on the thickness of the viscous sub-layer. This, in turn, depends on the mean flow rate, explaining the different slopes of the phase curves in Fig. 5. When the data are scaled with ω^+ the flow rate dependence is incorporated into the frequency parameter, presumably accounting for the collapse of the phase curves in Fig. 6. At low frequencies, however, the shear wave set up by the flow oscillations extends over the radius of the pipe. In this respect, the radius, rather than a near wall length scale, might be the most appropriate length scale with which to collapse the data. We expect, therefore, that the collapse of data on ω^+, which contains a near wall length scale, becomes worse as ω^+ gets smaller. This might explain the scatter in the magnitude plot at intermediate ω^+. Brereton and Mankbadi [17] observed a similar increase in scatter at low ω^+ for shear stress variation.

4. Response to non-sinusoidal velocity changes

4.1. Time domain model

To make predictions of transient heat transfer in unsteady flow with non-sinusoidal waveforms, we would like to develop a simple empirical transfer function that describes the experimental results. The model we choose must capture the effect of varying the mean flow rate and the frequency. Of course a linear model incorporates the effect of amplitude variation directly. In Fig. 5, where the data is scaled on the Stokes number, the shape of the transfer function clearly depends on the flow rate. When the data are scaled on ω^+, however, the phase relationship is almost independent of the flow rate. Furthermore, the magnitude of the transfer function appears to be almost independent of the flow rate in the high, and very low, frequency limit. Consequently, despite our observation that the magnitude does depend slightly on the flow rate at intermediate frequencies, we will try to express the relationship between the velocity and heat transfer coefficient purely as a function of ω^+.

For $\omega^+ \lesssim 0.015$, the phase relationship in Fig. 6 is approximately linear. Beyond this value the magnitude of the response is small, so we believe that it is acceptable to model the phase relationship as linear at all frequencies. The slope of the line is taken to be

$$\frac{\mathrm{d}\phi}{\mathrm{d}\omega^+} = t_{\tilde{h}}^+ = 270, \tag{2}$$

with ϕ measured in radians. The fit chosen for the magnitude response should fulfil a number of requirements.

- The curve fit must provide a reasonable approximation to the data in Fig. 6, tending to 0.8 at low frequencies and falling to zero at around $\omega^+ = 0.025$.

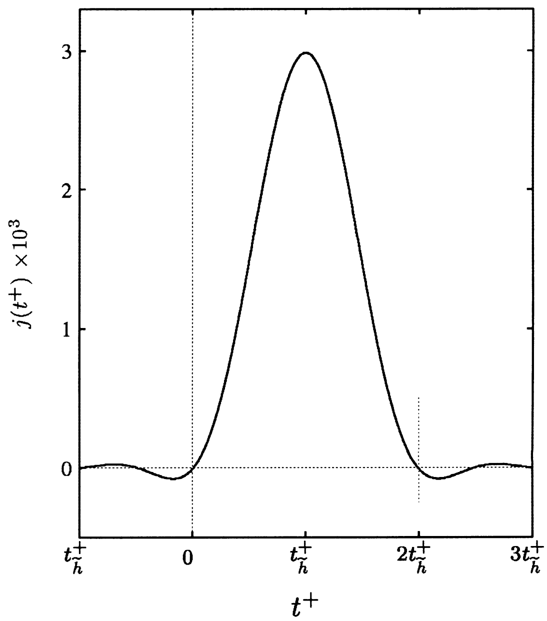

Fig. 7. The impulse response of the empirical model for the response of the heat transfer coefficient to velocity changes.

- An increase in velocity leads to an increase in heat flux, while a velocity decrease will cause a heat flux reduction. Consequently, the impulse response must be causal and positive.
- The frequency and the time domain models should be smooth and continuous.
- It would be desirable for the frequency domain model, and its transform in the time domain, to have simple analytical forms.

One function that comes close to meeting all of these requirements is

$$J(\omega^+) = 0.8\Pi\left(\frac{a\omega^+}{\pi}\right)\cos a\omega^+ \mathrm{e}^{-i\omega^+ t_{\tilde{h}}^+} \tag{3}$$

where $\Pi(x)$ is defined

$$\Pi(x) = \begin{cases} 1 & \text{for } -\dfrac{1}{2} < x < \dfrac{1}{2} \\ 0 & \text{otherwise} \end{cases} \tag{4}$$

We have no theoretical reason for selecting this function. It is proposed only as a convenient empirical fit to the experimental data. Other than the time delay, $t_{\tilde{h}}^+$ the only empirical constant required is a, for which we find a suitable value is 67. This empirical model, Eq. (3), is the solid line plotted in Fig. 6. The impulse response of the model is

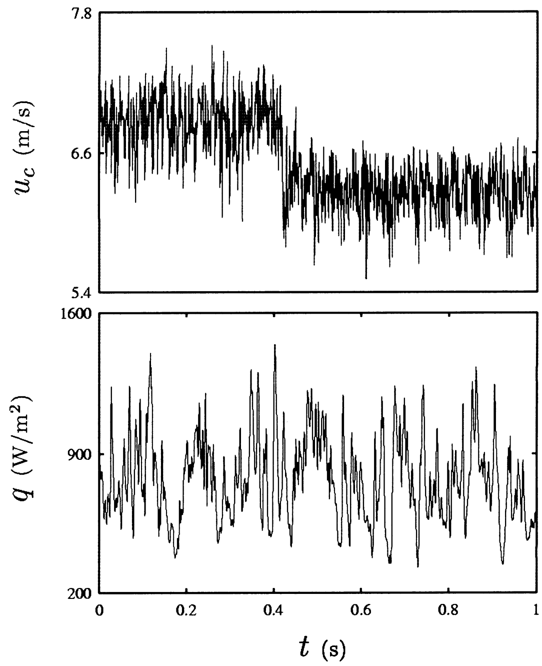

Fig. 8. The turbulent fluctuations of the heat flux disguise the underlying response to the step change in the flow rate.

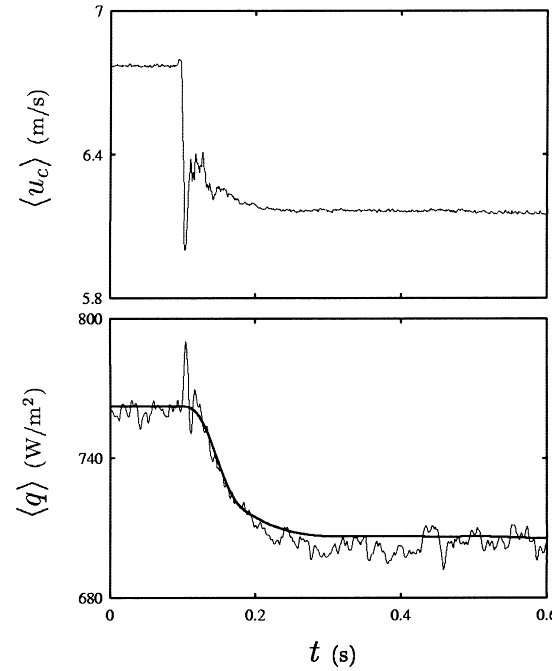

Fig. 9. The deterministic response of the heat flux to a step change in the flow rate.

$$j(t^+) = \frac{\mathrm{sinc}(\beta + 1) + 2\,\mathrm{sinc}\,\beta + \mathrm{sinc}(\beta - 1)}{10a} \tag{5}$$

where

$$\mathrm{sinc}\,x = \frac{\sin \pi x}{\pi x}$$

$$\beta = \frac{t^+ - t_h^+}{2a}$$

$$t^+ = \frac{tu_*^2}{\nu}$$

The shape of the impulse response is plotted in Fig. 7. The model is not strictly causal, nor is it always positive. However, it comes close to meeting both of these requirements.

4.2. Signal to noise characteristics of the heat flux

Before discussing the next series of experiments, we touch on the relative strengths of the turbulent component and the deterministic change of both the velocity and the heat flux. We have proposed the model of a shear wave travelling from the wall, modifying the turbulence and hence the heat transfer coefficient. The results presented shortly suggest that the underlying order implied by this model does indeed exist. However, this organised behaviour can easily be disguised by turbulent variation and it is only after the averaging process that it becomes evident. Experiments with step changes in the flow rate illustrate this point well, so we take this opportunity to present an example.

Fig. 8 shows typical traces of the centreline velocity and heat flux signals from one of our tests. The bulk flow rate undergoes a downward step approximately halfway through the time sequence shown. The velocity change is visible despite the turbulent variation, however, it is not clear from just one ensemble that the heat flux is at all affected. We found that at least 800, and in some cases up to 5000 ensembles were required to observe a systematic response of the heat flux to the velocity change.

4.3. Response to step and pulse-like changes

Fig. 9 shows the trend after approximately 3000 instances of the type of time series shown in Fig. 8 are averaged. Clearly, the velocity drop results in a systematic decrease in heat flux, but comparison of the vertical scales on the heat flux plots in Figs. 8 and 9 shows the fall to be almost an order of magnitude smaller than the size of the turbulent fluctuations. To model the transient response, the velocity is normalised by dividing through by its mean value, convolved with the scaled time domain representation of our modelled

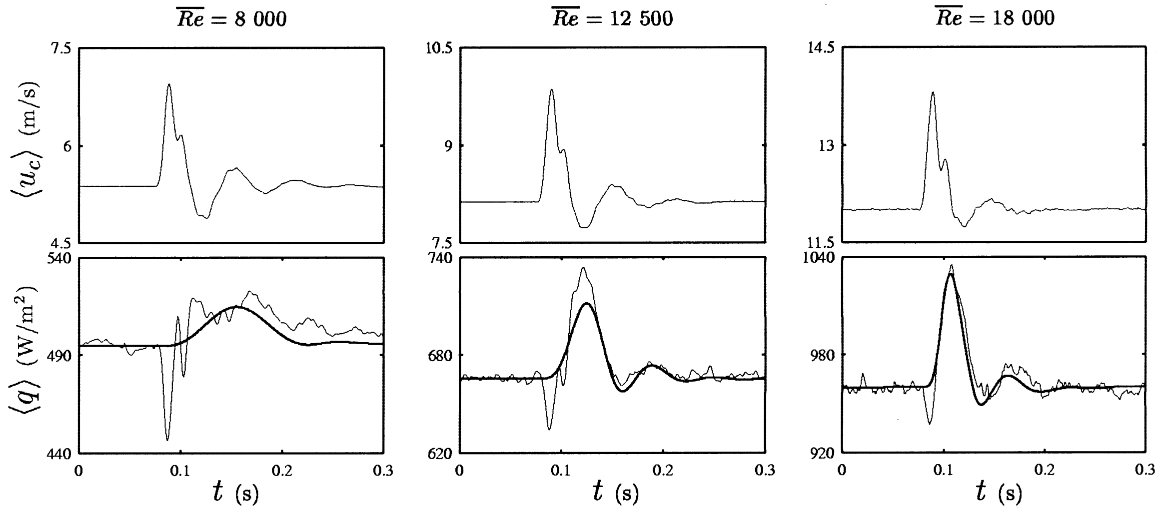

Fig. 10. The deterministic response of the heat flux to brief increases in the flow rate.

transfer function (Eq. (5)) and then multiplied by the average heat flux. The heavier line in Fig. 9 shows the resulting prediction. In this case, the model proved to be a fair predictor of the transient response and the quality of the model fit is typical of our other experiments with step changes.

The velocity change also appears to be associated with a second transient effect. At the time of the valve closure a pressure wave travelled along the pipe causing a sharp downward velocity 'spike'. This seems to correspond to an equally sharp, upward change in the heat flux. This counter-intuitive result could be reversed; an upward velocity 'spike' caused a downward change of the heat flux. Further investigation of this effect showed it to be highly correlated with the pressure variation and that it occurred in laminar, as well as turbulent, flow [12]. It appears that the pressure wave associated with the flow changes interacts with the boundary layer in some way to cause a radial component of velocity. This temporarily thickens or thins the thermal boundary layer at the pipe wall, decreasing or increasing the temperature gradient at the wall. Further discussion of this effect can be found in Ref. [12].

The second test of our linear model was a series of experiments to find the response of the heat transfer to brief increases of an otherwise steady flow rate. The impulse response of a linear system can be found by exciting the system with a unit impulse and observing the resulting output. Our tests were intended to resemble this procedure but obviously, the pulse of flow has a finite duration in these experiments.

The analysis was similar to that described for tests with steps and once again, a large number of instances were required to extract the systematic response of the heat flux. Fig. 10 shows results from three runs, demonstrating a progression as the Reynolds number increases. In each case, there is a delay between the peak of the velocity and the peak of the heat flux but this delay is reduced, and the response sharpened, as the flow rate increases. At high flow rates the model appears to perform a reasonable job of predicting the delay, shape and magnitude of the response but it does not do so well at lower Reynolds numbers. The transient response associated with the pressure wave appears to be relatively more significant and the response of the heat transfer coefficient is less well defined at the lower flow rates.

5. Effects of relaminarisation

In the experiments described so far the flow remained fully turbulent throughout the unsteady cycle or event. However, it has been shown that unsteady flow can relaminarise even when the instantaneous Reynolds number is well above the critical value for steady flow [19–21], so it is useful to consider whether relaminarisation has any impact on the behaviour we have described.

5.1. Transient behaviour

The effects of larger oscillations are shown in a progression displayed in Fig. 11. In this series of experiments the frequency of a sinusoidal oscillation is fixed at approximately 3 Hz ($\omega^+ \approx 0.003$), the mean Reynolds number is 10,500 and the relative amplitude is

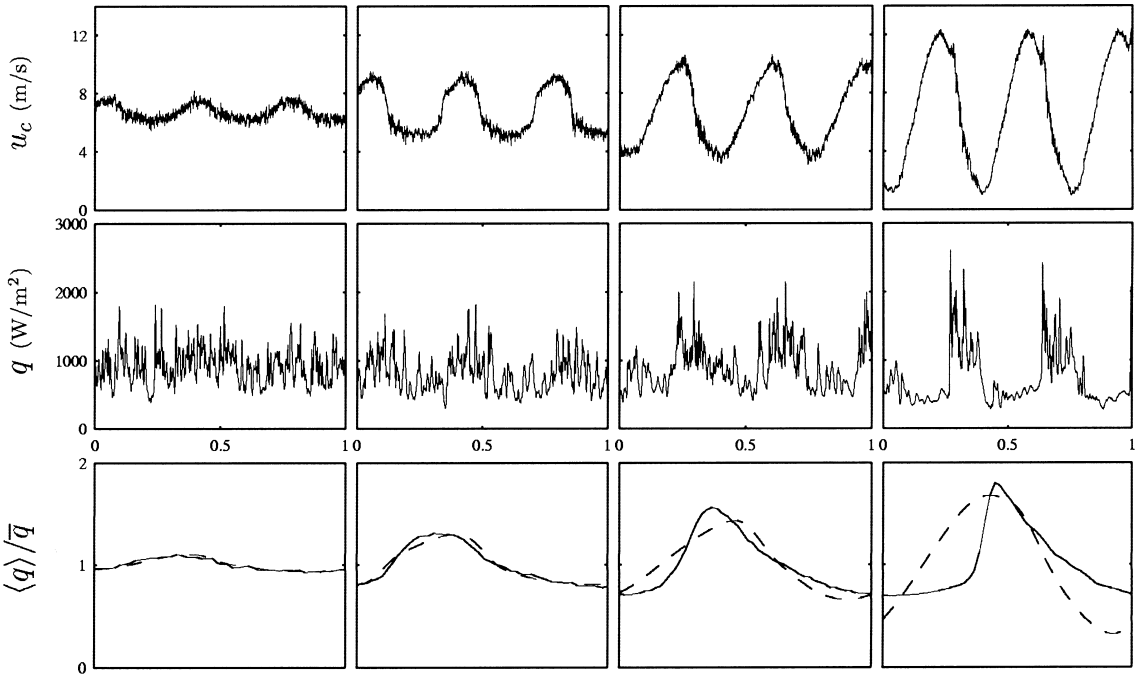

Fig. 11. The effects of the progression towards relaminarisation as the amplitude increases.

increased in steps from around 10% to approximately 80%. We were concerned that the larger oscillations might have led to flow reversal in an annular region at the pipe walls, such behaviour having been reported with relative amplitudes as low as 64% [18]. Fortunately, flow reversal was very easy to detect in these experiments. Cold air drawn backward into the tube at times of flow reversal caused an increase in heat transfer at the pipe wall, clearly distinguishable from any increase caused by a change in flow conditions by both the suddenness and magnitude of the change. If the amplitude increased beyond 80% flow reversal occurred limiting the range of these experiments.

The two upper subfigures for each amplitude illustrate typical one second sequences of the velocity and heat flux variation. The lower plot shows the ensemble averaged, cyclic variation of the heat flux (solid line) and the modelled prediction (dashed line).

At small amplitudes, the character appears similar to that of a steady turbulent flow. The turbulent fluctuations have approximately the same magnitude throughout the cycle and the underlying order to the heat flux behaviour only becomes clear after ensemble averaging. With increased amplitude the turbulent fluctuations become noticeably stronger during deceleration and weaker during acceleration. The modulation of the heat flux also becomes more apparent. When the amplitude is increased still further, the heat flux undergoes a dramatic increase and decrease during the cycle. This behaviour appears to be consistent with the patterns of relaminarisation and retransition observed in a number of previous studies of similar unsteady flows [19–21] and so we believe this to be the cause of the variation in heat flux. However, two alternative explanations of the observed behaviour were also considered. Flow reversal has been discussed above and was discounted for the reasons given. Variation of the bulk temperature, caused by variation of the residence time in the pipe, was also observed but the heat flux changes appear to reflect neither the magnitude nor the phase of the air temperature changes. Assuming that relaminarisation is indeed the cause of the behavior seen in these experiments, as we believe, it is not surprising that our empirical model fails to capture this non-linear behaviour.

5.2. Effect on the mean heat transfer

It is interesting to consider the effect of relaminarisation on the mean heat flux. In the first three cases shown in Fig. 11 the mean heat flux varies by only a few percent; an amount that could easily be attributed to experimental error. However, in the final case, where the relaminarisation effect is strong, the average heat flux dropped by about 20%. This is significant enough drop for us to conclude that relaminarisation can cause an appreciable reduction of the average heat transfer. In flow with strong reversals it is possible to maintain fully laminar flow throughout the unsteady cycle [19,21] so these results indicate a mechanism by which heat transfer could be reduced below the quasi-steady level.

6. Conclusions

In a limited parameter range, we have shown that the transient response of the heat transfer coefficient can be modelled with a linear relation to unsteady velocity changes. When the frequency is scaled into the Stokes number, the shape of the transfer function depends on the flow rate. In turbulent flow an alternative frequency parameter, ω^+ relates the thickness of the unsteady shear layer to the viscous sub-layer characteristic of a steady turbulent flow. The shape of the transfer function is almost independent of the flow rate when plotted against this parameter.

The assumption of linearity enables us to treat non-sinusoidal waveforms by utilising an empirical curve fit to the transfer function for sinusoidal variation. We find the time domain transform for this empirical model and predict the response of the heat transfer coefficient by convolving this impulse response with an arbitrary velocity signal. The model makes fair predictions for cases where the velocity undergoes a step change or briefly increases. However, the systematic response is much less than the random variation caused by turbulent motions, so the underlying order implied by our model is only evident after averaging.

Our linear model fails when the flow begins to relaminarise during the course of the unsteady cycle. Relaminarisation appears to be responsible for a reduction in the time-mean heat transfer in at least one of our tests. It seems likely that relaminarisation will decrease convective heat transfer in some cases of unsteady flow pipe flow.

References

[1] D.W. Wendland, Automotive exhaust-system steady-state heat transfer, in: 1993 Vehicle Thermal Management Systems — Conference Proceedings, Society of Automotive Engineers, SAE 931085, 1993, pp. 125–139.

[2] J.E. Dec, J.O. Keller, Pulse combustor tail-pipe heat-transfer dependence on frequency, amplitude, and mean flow rate, Combustion and Flame 77 (1989) 359–374.

[3] R. Lemlich, C.K. Hwu, The effects of acoustic vibration on forced convective heat transfer, American Institute of Chemical Engineering Journal 7 (1) (1961) 102–106.

[4] R.C. Herndon, P.E. Hubble, J.L. Gainer, Two pulsators for increasing heat transfer, Industrial and Engineering Chemistry, Process Design and Development 19 (1980) 405–410.

[5] J.S. Park, M.F. Taylor, D.M. McEligot, Heat transfer to pulsating turbulent gas flow, in: Proceedings of the 7th International Heat Transfer Conference, vol. 3, 1982, pp. 105–110.

[6] N.S. Liao, C.C. Wang, On the convective heat transfer in pulsating turbulent pipe flow, in: 1st World Conference — Experimental Heat Transfer, Fluid Mechanics, and Thermodynamics, 1988, pp. 536–542.

[7] L.G. Genin, A.P. Koval, S.P. Manchkha, V.G. Sviridov, Hydrodynamics and heat transfer with pulsating fluid flow in tubes, Thermal Engineering 39 (5) (1992) 251–255.

[8] R.H. Keil, M.H.I. Baird, Enhancement of heat transfer by flow pulsation, Industrial and Engineering Chemistry, Process Design and Development 10 (4) (1971) 473–478.

[9] P.J. Shayler, M.J.F. Colechin, A. Scarisbrick, Heat transfer measurements in the intake port of a spark ignition engine, SAE Transactions — Journal of Materials and Manufacturing 5 (1996) 257–267.

[10] W. Bauer, Heat transfer and mixture vaporization in intake systems of spark ignition engines. PhD thesis, Massachusetts Institute of Technology, 1997.

[11] W. Yang, Transient heat transfer in a vapor-heated heat exchanger with arbitrary timewise-varient flow perturbation, Transactions of the ASME — Journal of Heat Transfer 86 (1964) 133–142.

[12] A.R. Barker, Heat transfer in unsteady pipe flow. PhD thesis, University of Cambridge, 1998.

[13] J.M. Hager, S. Simmons, D. Smith, S. Onishi, L.W. Langley, T.E. Diller, Experimental performance of a heat flux microsensor, Transactions of the ASME — Journal of Engineering for Gas Turbines and Power 113 (1991) 246–250.

[14] F.W. Dittus, L.M.K. Boelter, Heat transfer in automobile radiators of the tubular type. University of California, Berkeley, Publications in Engineering 2 (13) (1930) 443–461. Reprinted in International Communications in Heat and Mass Transfer 12 (1985) 3–22.

[15] V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering 2 (16) (1976) 359–368.

[16] Z.X. Mao, T.J. Hanratty, Studies of the wall shear stress in a turbulent pulsating pipe flow, Journal of Fluid Mechanics 170 (1986) 545–564.

[17] G.J. Brereton, R.R. Mankbadi, Review of recent advances in the study of unsteady turbulent internal flows, Applied Mechanics Reviews 48 (4) (1995) 189–212.

[18] S.F. Tardu, G. Binder, R.F. Blackwelder, Turbulent channel flow with large-amplitude velocity oscillations, Journal of Fluid Mechanics 267 (1994) 109–151.

[19] M. Hino, M. Sawamoto, S. Takasu, Experiments on transition to turbulent in an oscillatory pipe flow, Journal of Fluid Mechanics 75 (1976) 193–207.

[20] M. Iguchi, M. Ohmi, Transition to turbulence in a pulsatile pipe flow. 3rd report, flow regimes and the conditions describing the generation and decay of turbulence, Bulletin of the JSME 27 (1984) 1873–1880.

[21] D.M. Eckmann, J.B. Grotberg, Experiments on transition to turbulence in oscillatory pipe flow, Journal of Fluid Mechanics 222 (1991) 329–350.