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TURBULENT HEAT TRANSFER ON THE SURFACE OF A STEAM-CHUGGING BUBBLE

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ABSTRACT

The length and velocity scales of the turbulent eddies at the surface of a type of steam-chugging bubble (the detached bubble) are derived from a physical model. The corresponding turbulent heat transfer result, which contains no free parameters, is $Nu = 0.04\ Re^{7/8}\ Pr^{1/2}$. A comparison with the available experimental data shows excellent agreement.

Introduction

A phenomenon known as "steam-chugging" occurs in the suppression pool of a boiling-water reactor in the late stages of a postulated loss-of-coolant accident. The physical system includes the steam-filled reactor containment, the suppression pool, and a number of pipes venting the steam from the containment vessel into the pool. In the late stages of the accident, when the steam flow rate is so low that it fails to sustain a continuous steam region at the pipe exit, steam chugging occurs. The characteristics of this phenomenon are the periodic growth and collapse of steam bubbles at the pipe exit and, in between two successive bubbles, the water "chugging" into the vent.

The pressure spikes associated with the bubble collapse are particularly annoying because of their potential for damaging the structures in the pool.

Numerous small-scale experiments have been performed by various groups to study the interface motion and the pressure loads [1, 2, 3, 4]. Of these, Lee and Chan [1, 4] observed and recorded the most detailed data on both the interface motion and the pressure loads. Three distinct chugging modes are observed in the low-flow region (<75 kg/m^2-s): (1) the internal chug where the interface oscillates inside the injection pipe, (2) the detached bubble chug where the steam bubble detaches from the pipe exit as it collapses immediately after its formation, and (3) the encapsulating bubble chug where the bubble grows to envelop the pipe exit.

Of the three chugging modes observed, the encapsulating bubble chug produces the highest dynamic loads. Recognizing its industrial importance, Lee and Chan [4] made a detailed study of the bubble interface. They observed that the bubble surface is smooth and glassy during the growth phase; and as the bubble reaches its maximum size, a water jet begins to penetrate the bubble from below. Then the lower portion of the bubble suddenly roughens causing violent collapse.

Lee & Chan [1, 4] asserted that the sudden roughening of the interface is a turbulence phenomenon; and from their experimental data they derived the time-averaged heat transfer coefficients for twelve encapsulating bubbles. The large heat transfer coefficients obtained ($\sim$100 kw/m^2-°C) substantiated their postulate that the interface is indeed turbulent. However, they were not the first to suggest such a phenomenon. Numerous authors [5, 6, 7] in fact have proposed turbulence models for the interface. These models are generally order-of-magnitude type predictions based on some turbulence parameters that must be derived from experimental data. None of them offered a physical explanation for the sudden onset of turbulence, nor did they give theoretical estimates of the turbulence length and velocity scales that govern the heat transfer.

This paper presents a physical model for the turbulence characteristics at the surface of the detached bubble that characterize the detached bubble chug. The predicted heat transfer coefficient is compared with the data given in [1]. The agreement is within $\pm 40\%$. In view of the state-of-ignorance of these turbulent interface problems, the agreement is excellent.

The Eddy Penetration Model

In order to facilitate the present discussion, the movie data of a typical detached-bubble steam chugging event [1] (Run No. FM3, Bubble 3) are chosen as reference. Consider the water discharge phase of the chugging process (Picture (1), FIG. 1): the discharge velocity is 4 m/s, which gives a Reynolds number of $Re_D = 6.8 \times 10^5$. The flow is definitely turbulent although not fully developed ($L/d \sim 5$). Nevertheless, two types of eddies are identified, the large inertial eddies in the turbulent core and the small dissipative eddies in the buffer layer. For the large eddies, the length scale is on the order of the pipe diameter and the velocity is that of the bulk flow. For the small eddies, the length scale is about the same as the buffer layer thickness, whereas the velocity scale is given by the friction velocity. Since the buffer layer thickness is at $y^+ = 30$, the length scale of the small eddies is given by

$$\lambda = \frac{30}{V_*} \nu \qquad (1)$$

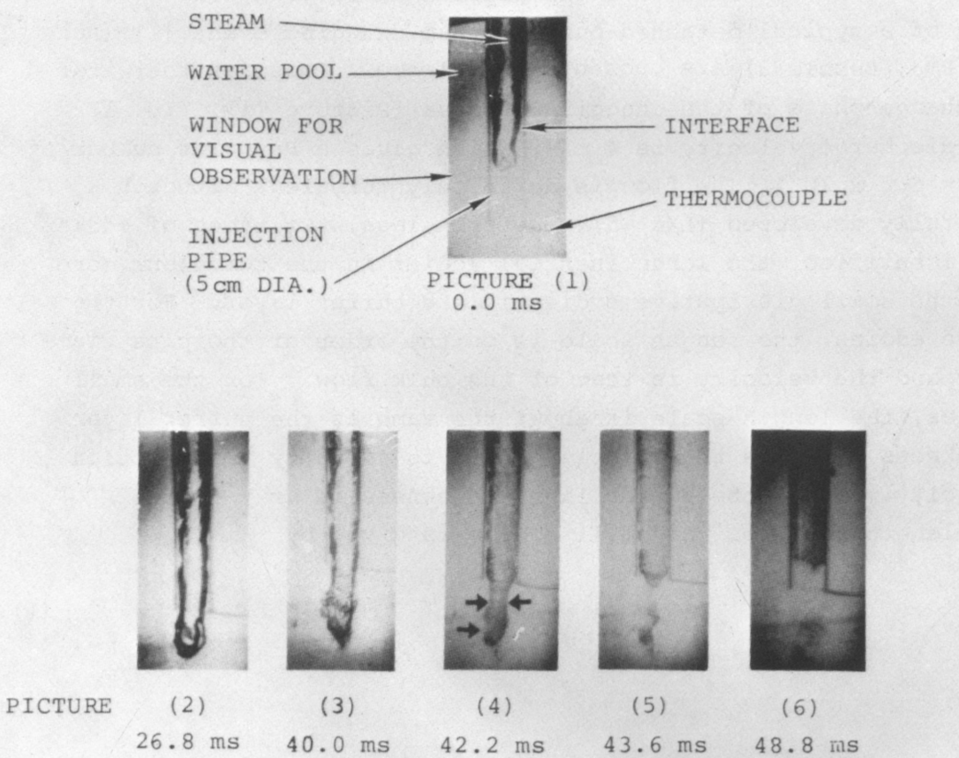
where ν is the kinematic viscosity of water, and

$$V_* = 0.2\ V_D Re_D^{1/8} \tag{2}$$

is the friction velocity, a function of the velocity V_D and Re_D at the discharge. For the present case, $\lambda = 5.8 \times 10^{-5}$ m and $V_* =$ 0.15 m/s. The time scale ($\tau = \lambda/V_*$) is therefore on the order of a millisecond. Since the time scale of the water discharge is ∿50 ms, there is plenty of time for the flow to produce these eddies, i.e., the buffer layer is fairly well established.

FIG. 1. Photos of Interfacial Motion: Detached Bubble Chug

Run No. FM3, Pool Subcooling = 37.2°C, Submergence = 50 cm
Note: Arrows Indicate Direction of Liquid Jet



As the water is discharged, the inertial eddies travel to distant ranges while the small scale eddies remain near the pipe exit for lack of inertia. The extremely small eddies, which are on the order of the viscous sublayer, will generally be left on the pipe wall; and as the interface sweeps by, surface tension would completely eliminate them. As a result, when the bubble begins to form at the end of the discharge, there is an abundance of eddies around the pipe exit, the length and velocity scales of which are very unique: the length scale is about the same as the shear layer thickness of the discharge flow, and the velocity scale, the corresponding friction velocity.

The major characteristic of this type of chugging is that immediately after its formation, a number of water jets will rush toward the exit "cutting off" the bubble from the pipe (hence, the detached bubble) while the bubble collapses axisymmetrically below the exit. The immediate rush of the surrounding water toward the bubble indicate that the initial steam pressure is below that of the surrounding water. The bubble formation process is by and large a steam penetration process where the steam region progresses a couple of pipe diameters below the pipe exit before it stops, and the collapse follows immediately. In this case, little time is given for the surrounding eddies to decay. The time of decay is expected to be a few times the time scale of the eddies [8], i.e., a few milliseconds in this case; and, from Pictures 3 and 4, the collapse is well on its way in ~ 2 ms. Therefore, the decay of these eddies can be neglected. During the penetration process the eddies are convected away from the interface because of the diverging flow; however, upon collapse, the eddies are convected toward the interface. Evidently, from Picture (3), at the incipience of the collapse, the bubble surface is bombarded by a number of inertial (large) eddies. These eddies, however, could not have reached the interface without penetrating the layer of small eddies around the pipe exit. This is the initiation of the rapid condensation. Note that the interface need not be very rough for rapid heat transfer to take place. As long as small eddies exist at the immediate vicinity of the interface rapid heat transfer is expected.

The interface roughening occurs one frame later ($\sim$.4 ms). The roughness is similar to that shown on Picture (4). Since the rapid condensation is initiated by the small eddies convected to the interface by the rather random process of large eddy penetration, it may be conjectured that the rapid condensation occurs initially at some localized spot on the interface, but not on the entire bubble surface. The underpressure created by this local jump in condensation will propagate as two rarefactions, one in the water and the other in the steam. However, the one in the water travels at four to five times the speed of the other. Thus for a very brief moment, the surrounding water at the interface has a lower pressure than the steam. A weak shock is formed at the interface that propagates into the water. The shock strength however is not uniform because of the spherically decaying rarefaction in the water. Except for very special situations, the passage of a shock will cause even a potential flow to to become rotational. For a shock, with spatially non-uniform strength, generated at the curved interface, some rotation would undoubtedly be introduced. If the eddies are considered to be small vortices, an increase in rotation increases their turbulent intensity. Another way of looking at this is that an impulse is delivered to the water due to this mismatch in the speeds of the rarefaction waves. For the present case, the time period of this overpressure is a fraction of a millisecond. For eddies having time scales large compared to a millisecond, the impulse has a negligible effect. However, for eddies having comparable or smaller time scales, this impulse can give them an acceleration that results in a velocity sufficient to penetrate the interface. From Levich [9], this critical velocity is given roughly by

$$\rho V^2_{crit} = \frac{\sigma}{\lambda} \tag{3}$$

where σ is the surface tension. In this case, V_{crit} is $\sim$1 m/s, not much larger than the turbulent velocity V_* for these eddies.

However, as soon as the eddies reach the interface, surface tension and viscous effects will reduce their intensity. Assuming that the velocity gain from the impulse is cancelled by these effects, a surface renewal model [10] would give the heat transfer as

$$h = 1.12\ \rho c \left(\frac{\alpha V_*}{\lambda}\right)^{1/2} \tag{4}$$

where ρ, c, and α are the density, specific heat and thermal diffusivity of water respectively. Substituting Eqns. 1 and 2, and after some manipulation, Eqn. 4 becomes

$$Nu_B = 0.04\ Re_D^{7/8}\ Pr^{1/2} \tag{5}$$

where both the Nusselt number and the Reynolds number are based on the diameter of the injection pipe.

The bubble pressure as well as that in the injection pipe is instantly reduced. Simultaneously, a rarefaction wave propagates from the bubble and up the injection pipe. There is an increase in the steam flow at the pipe exit due to the rarefaction. However, it can be shown that the amount injected into the bubble within a millisecond or so is small compared to the initial mass of steam in the bubble before the onset of turbulence. In the meantime, a water slug enters the injection pipe. The underpressure caused by the abrupt penetration of the eddies is the pressure difference initiating the more rapid collapse. Of course, the heat transfer coefficient given by Eqn. 4 is applicable only at the onset of turbulence. As the collapse progresses, the interface shape and velocity will augment the intensity of the eddies. If high shear develops, eddies of other sizes will also be created.

Data Reduction And Comparison With Theory

The validity of the above model must be checked by comparing its predictions with the available experimental data. However, before the comparison is done it is important to recognize that the movie data reported provides only the following quantities: the water discharge velocity (V_D), bubble volume (V_{BUB}), surface area (A_{BUB}), and the time-averaged heat flow ($\overline{hA}$). On the other hand, the theory gives only a heat transfer coefficient at the onset of turbulence. To bridge the gap, the following physical model is used.

First, the assumption is made that once a small eddy is convected to the interface, it will stay with the interface until the final stages of the collapse. Second, the eddies are assumed to be spherical. As the bubble collapses, the eddies are squeezed to become ellipsoids. The behavior of the eddies in a flow contraction has been studied by Taylor [11]. It is found that for a homogeneous isotropic turbulence, the streamwise component of the turbulence velocity decreases as ℓ^{-1} (=the contraction ratio) while the lateral components increases as $\ell^{1/2}$. Geometrically the streamwise turbulence length scale will increase as ℓ while the lateral scales will decrease as $\ell^{-1/2}$ to conserve volume. The time scale for the streamwise component therefore increases as ℓ^2 while the time scales for the lateral components decreases as ℓ^{-1}. Using the surface renewal model, the heat transfer (summed over all components) will increase as $\ell^{1/2}$. This, of course, assumes that the eddies protrude into the steam as half-ellipsoids. The observed rough interface lends strong support to this assumption (Picture (4)).

For an axisymmetric collapse of a cylindrical bubble, the heat transfer surface will decrease with the radius. The movie data suggest that the collapse is not only axisymmetric but also rather uniform, i.e., the radius decreases linearly with time. These observations lead to the following equations for the heat transfer coefficient and the collapse radius:

$$h = h_o\sqrt{\frac{R_o}{R}} \tag{6}$$

$$R = R_o\left(1 - \frac{t}{\Delta t}\right) \tag{7}$$

where R_o is the initial bubble radius, h_o is the initial heat transfer coefficient, Δt is the time of collapse, and it is recognized that $\ell = R_o/R$. The time-averaged heat flow $\overline{hA}$ is

$$\overline{hA} = \frac{A_{BUB}h_o}{\Delta t}\int_o^{\Delta t}\sqrt{1 - \frac{t}{\Delta t}}\,dt = \frac{2}{3}\,A_{BUB}h_o \tag{8}$$

where A_{BUB} is the initial bubble surface area. This relates the heat transfer coefficient at onset of turbulence to the measured $\overline{hA}$.

The movie data in [1] are scanned to obtain the detached bubble chug events. However, only Runs FM2, 3, 4 and 5 are taken with a fast enough frame-speed (2500 frames/s) to allow accurate prediction of the desired quantities. The reduced data are tabulated in Table I along with the predicted heat transfer coefficient from Eqn. 5. The agreement between the experimentally derived h_o and the predicted is within $\pm$40%.

Because of the cylindrical nature of these bubbles, the error incurred in the data reduction process is small. An error estimate on the calculated bubble volume indicates an error of ∿10%. The error incurred in the measurement of bubble collapse time is ∿10%. However, in the computation of $\overline{hA}$ these errors tend to

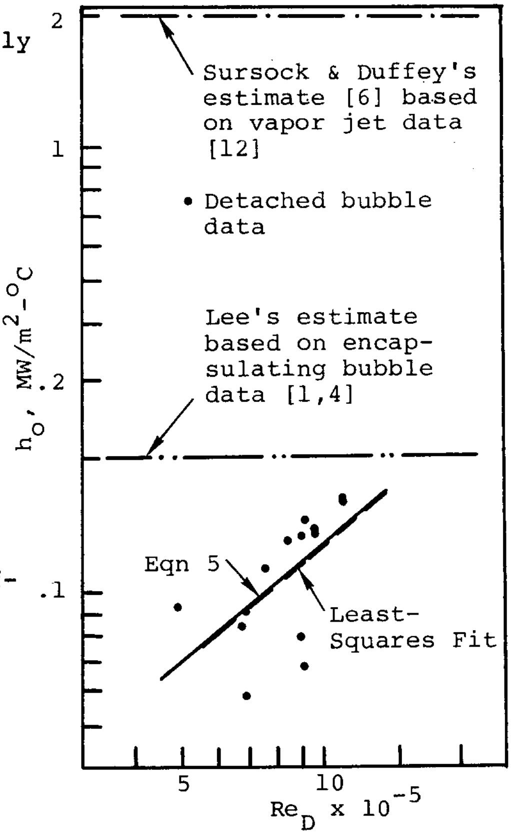
TABLE I. COMPARISON WITH EXPERIMENTAL DATA

Run No. (Subcooling)	Re_D $x10^{-5}$	V_{BUB} $10^{-6}m^3$	A_{BUB} $10^{-4}m^2$	$\overline{hA}$ kW/°C	h_o, EXPT. $kW/m^2 °C$	h_o, THEORY $kW/m^2 °C$
FM1 (46.1°C)						
Det.Bub. 1	8.2	79	71	.59	126	106
" " 2	8.9	108	114	.58	77	114
" " 3	8.9	40	51	.43	128	114
FM2 (37.2°C)						
Det.Bub. 1	6.8	103	41	.15	57	90
" " 2	9.6	62	65	.57	132	121
" " 3	9.6	58	61	.54	133	121
" " 4	10.9	79	71	.74	156	136
FM3 (62.7°C)						
Det.Bub. 1	10.9	43	46	.48	158	136
" " 2	9.1	124	104	.46	66	116
" " 3	6.8	72	76	.45	88	90
FM4 (54°C)						
Det.Bub. 1	9.1	181	120	1.16	145	116
" " 2	4.9	79	84	.51	91	68
FM5 (37.2°C)						
Det.Bub. 1	7.6	114	80	.59	111	99
" " 2	6.8	146	103	.55	81	90

cancel. The resulting error in $\overline{hA}$ is no more than a few percent. This is the case because if the timing on the onset of collapse is one frame (.4 ms) later than the actual onset, the bubble volume would have decreased. The effect on $V_{BUB}/\Delta t$ is therefore small. The major error is in the estimation of the heat transfer surface because of the irregular nature of the surface. By comparing the areas obtained by assuming the bulges on the interface to be surfaces of cones to that of a flat surface, the error is estimated to be at best 40% since the aspect ratio (height/base) of these indentations is generally less than a half. The overall error in the computation of h_o, assuming that Eqn. 8 is correct, is no more than 50%.

The results in Table I are shown in FIG 2. As expected, the data show a considerable amount of scatter (like a factor of two) typical of data of this sort [13]. Because of this scatter any curve fit or theory would suffer basically the same order of inaccuracy. In this case, the least-squares fit and the theory actually coincide. Considering the amount of data scatter, and the state-of-ignorance of the present problem, the agreement between the theory and the data is exceptionally good. The estimated order of magnitude of the interface heat transfer coefficient of a steam jet condensing in subcooled water [12] is also plotted. As pointed out before [4], the heat transfer coefficients for condensing jets are generally an order of magnitude higher than those measured in steam chugging. The reason that this line (constant) is on this plot is because the same order of magnitude has been used by some [6] in order to predict the dynamics of steam chugging, in lack of anything better to use. A more reasonable estimate is given by Lee [1] who derived the heat transfer coefficient for encapsulating bubbles by matching theoretical bubble collapse times with those measured. His result agrees well with those derived from the movie data [4]. Finally it should be mentioned that a theoretical prediction of $\overline{hA}$ can be obtained from Eqns 5 and 8; and, a direct comparison with the measured $\overline{hA}$ can be done. However, since the heat transfer coefficient is the physical quantity of interest, the present comparison is adopted.

FIG. 2. Plot of h_o vs Re_D



Conclusion

The above physical observations show that the turbulent heat transfer at the surface of the steam bubble in the detached bubble chug is controlled by the small eddies generated in the buffer layer of the turbulent water flow in the pipe during the discharge phase of the chugging process. It may further be concluded that the transient heat transfer on the surface of a vapor bubble is very sensitive to the method by which the bubble is formed. If the formation process causes the production of small turbulent eddies, the heat and mass transfer can be greatly enhanced. For the steam-chugging bubble, in a detached bubble chug, the following theoretical result is offered for the turbulent heat transfer:

$$Nu_B = 0.04\ Re_D^{7/8}\ Pr^{1/2}$$

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References

1. C.K.B. Lee, Ph.D. dissertation, University of California Los Angeles (June 1979).

2. T. Kukita et al., NUREG/TR-0036, Japan Atomic Energy Research Institute (June 1977).

3. G.B. Andeen and J.S. Marks, EPRI-NP-908, Electric Power Research Institute (October 1978).

4. C.K.B. Lee and C.K. Chan, Basic Mechanisms in Two-Phase Flow and Heat Transfer, Proc., ASME-WAM, Chicago (November 1980).

5. W. Kowalchuk and A.A. Sonin, NUREG/CR-0221, U.S. Nuclear Regulatory Commission (1978).

6. J.P. Sursock and R.B. Duffey, Basic Mechanisms in Two-Phase Flow and Heat Transfer, Proc., ASME-WAM, San Francisco (December 1978).

7. G. von Class, KFK-2487, Nuclear Research Centre Karlsruhe (1978).

8. G.K. Batchelor, Homogeneous Turbulence, Cambridge University Press, London (1967).

9. V.G. Levich, Physicochemical Hydrodynamics, Prentice Hall, Englewood Cliffs, New Jersey (1962).

10. G.E. Fortescue and J.R.A. Pearson, Chemical Engineering Science, 22: pp. 1163-76 (1967).

11. G.I. Taylor, Zeitschrift fur angewandte Mathematik und Mechanik, Vol., XV, pp. 91-6 (1935).

12. C.P. Greef, RD/B/N3262, Central Electricity Generating Board, Berkeley Nuclear Laboratories, United Kingdom (1975).

13. S.G. Bankoff, Paper presented at EPRI Workshop on Basic Two-Phase Flow Modeling In Reactor Safety and Performance, Tampa, Florida (February 1979).