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Direct Contact Heat Transfer Between Immiscible Liquids in Turbulent Pipe Flow

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This paper reports on an experimental study of the direct contact heat transfer between oil and water in turbulent pipe flow under nonboiling conditions. Data were taken by a new technique, namely, monitoring on a very fast response recorder the output of a small thermocouple placed in the two-phase flow. The variables studied were the liquid velocity, the pipe diameter, the water volume fraction, and, to a lesser degree, the interfacial tension and the oil viscosity. A successful semiempirical method of correlating the data is also presented.

A new saline water conversion process, the cardinal feature of which is heating brine by dispersing it in a hot immiscible liquid in highly turbulent pipe flow, was proposed by Wilke (*13*) in 1958. Since then, various aspects of this process have been under investigation at the Sea Water Conversion Laboratory of the University of California at Berkeley, and a survey of these investigations has been given in a recent paper by Wilke et al. (*14*). That paper suggested possible advantages for a direct contact heat transfer process for seawater conversion over conventional distillation processes using metallic heat transfer surfaces, and went on to outline how the basic idea might be employed in developing a multistage seawater conversion plant. The potential advantages include the abatement of scaling and corrosion problems, the use of simple and inexpensive equipment, and the adaptability of the process to very large throughputs.

The present paper reports on an experimental study of the *direct contact* heat transfer between oil and water in turbulent pipe flow under nonboiling conditions. Data were taken by a new technique, namely, monitoring on a very fast response recorder the output of a small thermocouple placed in the two-phase flow. Evidence is presented that the (consistent) method adopted for interpreting the thermocouple traces gives reliable liquid temperatures. The variables studied were the liquid velocity, the pipe diameter, the water volume fraction, and, to a lesser degree, the interfacial tension and the oil viscosity. A successful semiempirical method of correlating the data is also presented. Some measurements of the heat transfer during boiling, the pressure drop in two-phase liquid-liquid flow, and the identity of the continuous phase were also made, but these are only touched upon here. The interested reader may consult the thesis by Porter (*7*) for the details.

The most closely related published work is that of Grover and Knudsen (*1*), who measured the rates of heat transfer between a petroleum solvent and water flowing cocurrently in a 1.50-in. diameter horizontal pipe at total flow rates ranging from 2,500 to 15,000 lb./hr. These flow rates are sufficiently low that stratification occurred within the pipe downstream from the mixing section. Heat transfer coefficients were computed from the liquid temperatures measured in the stratified region. Grover and Knudsen found that for their range of variables the volumetric heat transfer coefficient was independent of the dispersed phase volume fraction and the inlet temperatures, but varied roughly proportional to the 1.6 power of the total linear velocity. In the present study the liquid velocities were sufficiently high to preclude stratification. It should not be surprising then if the dependence of the volumetric heat transfer coefficient on the operating variables is different in this study from that found by Grover and Knudsen (1) or by Wilke et al. (*14*) with stratified

Table 1. Test Section Dimensions

I. D., in.	Total heat transfer length, ft.	Length to diameter ratio	Total heat transfer volume, cu.ft.
0.325	4.50	166	0.00259
0.531	7.67	173	0.0118
1.00	9.00	108	0.0492
1.50	11.20	90	0.137
3.00	7.00	28	0.344

flow in view of the difference in the hydrodynamics.

EXPERIMENTAL APPARATUS AND PROCEDURE

The test stand used for the experimental work in this study was designed by Richeda (*8*) and is described in detail in a project summary. It had been used to obtain the preliminary data reported by Wilke et al. (*14*). The initial equipment arrangement formed a closed loop in which both water and oil flowed continuously through the system. Oil and water were stored in separate tanks and were preheated by passing each liquid through a closed loop containing a steam heat exchanger. After the desired temperatures were achieved, the oil and water were metered into a mixing tee followed by a section of pipe in which they flowed together in turbulent flow and exchanged heat. A vapor-liquid separator at the end of the test section received the oil and water (and steam in the case of boiling) mixture. From the vapor-liquid separator the liquid mixture flowed by gravity into a liquid-liquid settler and after separation the two liquids were returned to their respective tanks. Due to problems associated mostly with inadequate liquid-liquid settling, the experimental data in this study were taken on a batch basis. The above system was retained intact except that the oil-water mixture was diverted from the vapor-liquid separator into a large holding tank where it was allowed to remain until settling was complete.

Five test sections of different inside diameters were used. Each was a glass pipe preceded by a mixing tee of the same diameter. The dimensions of the test sections are listed in Table 1.

Thermocouples were fabricated by spot welding 2-mil diameter chromel and alumel wires. The junction was suspended by passing the wires through thin glass capillaries inserted in a rubber stopper, pulling the wires taut, and filling the capillaries with epoxy resin. A thermocouple was then placed in the test section through a tubulation provided in the glass pipe. The thermocouple leads were connected to a Kintel Model 112A d.c. amplifier which was capable of amplifying the thermocouple voltage signal up to 1,000-fold and of following a 10 kc. signal to ± 3%. The output of the amplifier was the input to a Honeywell 906 C Visicorder. In the Visicorder, a galvanometer with an attached mirror rotates according to the voltage signal it receives. The mirror then reflects light from a mercury vapor lamp onto a light-sensitive paper thus giving a permanent trace. The speed of response of the galvanometer used in this study was 600 cycles/sec.

The liquids used in this study were tap water and a hydrocarbon, 90 Neutral Oil supplied by Union Oil Company. The pertinent physical properties of the 90 Neutral Oil are listed in Table 2. These are either catalog values or were determined by standard means. To assess the importance of the interfacial tension in the heat transfer process, a second hydrocarbon, 100 Spray Base, supplied by Shell Oil Company, was used for several runs. This oil has essentially the same physical properties as 90 Neutral Oil except that its interfacial tension is 55, 40, and 35 dynes/cm. at 50°, 100°, and 150°F., respectively.

Table 2. Physical Properties of 90 Neutral Oil

Temperature °F.	Density, lb./cu.ft.	Thermal conductivity B.t.u./hr.°F.ft.	Heat capacity, B.t.u./lb.°F.	Viscosity, lb./hr.ft.	Interfacial tension, dynes/cm.
50	54.4	0.0800	0.44	130	25.5
100	53.3	0.0787	0.47	35	19.2
150	52.2	0.0774	0.50	15	16.6

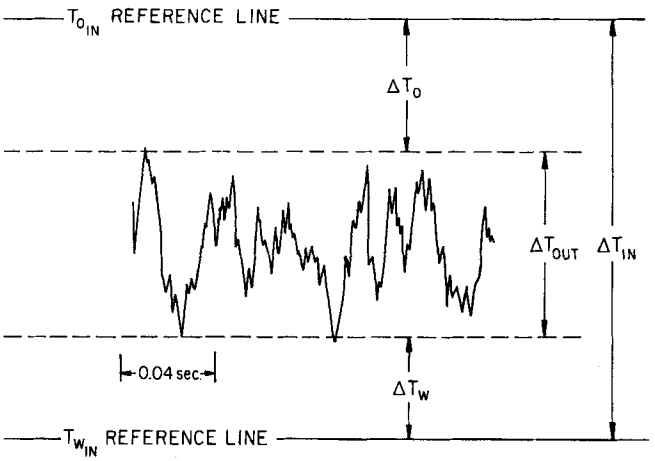

Fig. 1. Typical thermocouple trace.

When the two liquids had obtained the desired temperatures, one of them was fed into the test section and the temperature indicated by the thermocouple was noted. The same procedure was then carried out for the second liquid. In this way two reference temperatures, the inlet oil temperature and the inlet water temperature, were obtained. Oil and water were then caused to flow through the test section together at the desired rate and the temperature signal from the thermocouple was monitored on the Visicorder. A schematic of a typical trace is reproduced in Figure 1.

INTERPRETATION OF THE THERMOCOUPLE TRACES

The widely varying peak heights and trough depths seen on the trace in Figure 1 were typical and presented a problem in inferring the "local" oil and water temperatures. By examining many traces a consistent method of determining these temperatures was developed. This method assumes that the "true" (or average) temperatures of the oil and water correspond to the extreme maxima and minima of the trace. This is taken to be true even though the number of these maxima or minima is very small (0.05 to 5%) in comparison with the large number of intermediate peaks and troughs. In most cases, traces of 1-min. duration gave five or more absolute maxima or minima for the dispersed phase temperature through which a straight horizontal line could be drawn. When there was a preponderant amount of one phase (volume fraction greater than 0.7) the temperature of this phase was easily determined by the large number of peaks or troughs indicating this temperature.

Several experiments were carried out to test the validity of the above procedure for obtaining the local temperatures. First, the thermocouple was placed only 4 in. away from the mixing tee. At this short distance very little heat transfer has occurred, so that the temperature trace should oscillate essentially between the inlet liquid temperatures if the response is sufficiently rapid. The extreme maxima and minima did reach these temperatures for flow velocities up to 20 ft./sec. for 2-mil thermocouple. (It was observed that a 10-mil thermocouple did not respond rapidly enough to reach the two inlet temperatures.)

In a second experiment, the temperatures and flow rates of the two liquids were adjusted so that boiling of the water occurred in the test section. The thermocouple was placed in a section of the pipe where boiling was taking place and the temperature of the water determined in the usual manner. The pressure at this point in the test section was also measured and the saturation temperature for water at this pressure was noted. These two temperatures were in very good agreement with each other (within 2° F.).

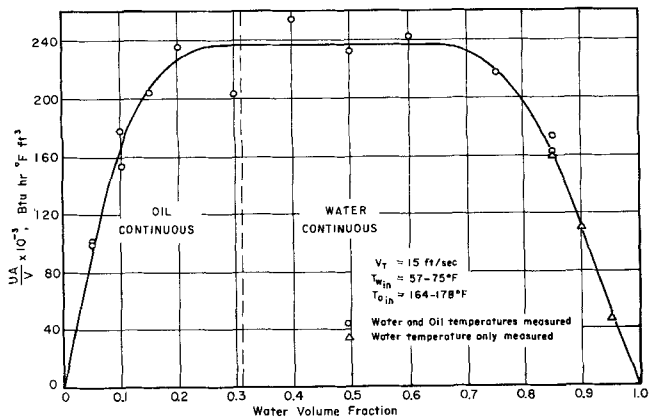

Fig. 2. Volumetric heat transfer coefficient as a function of volume fraction of water. The conditions shown are for 90 Neutral oil-water flowing in the 0.325-in. diameter test section at a total linear velocity of 15 ft/sec.

As a third check on the procedure, heat balances were made for every run. The average heat balance error was about 7% for the one hundred and sixty-nine heat transfer runs. Since the temperature determination of the continuous phase for volume fractions greater than 0.7 is thought to be quite reliable, agreement in heat balances indicates that the dispersed phase temperature determination is also reliable. Thus the procedure adopted for determining the temperatures of the two fluids from the thermocouple trace appears to be valid and the error in the calculated amount of heat transfer should not exceed 7%.

An explanation of the variability in peak heights would probably include the following: (1) Different drops could be at different temperatures due to their variability in size and past history. (2) Many drops may strike the thermocouple with glancing blows or large velocities thus having contact times shorter than the response time of the thermocouple. (3) The thermocouple may be wetted by one phase which partially shields it from the other phase. It appears that the thermocouple responds completely only when it is hit by the largest drops. A very rough order of magnitude calculation shows that the thermocouple will reach the droplet temperature only for drops with diameters about twenty times larger than the expected average size computed from an empirical equation (3). Because most of the volume of the dispersed phase is carried in the large drops and because frequent coalescence and redispersion of the drops would tend to even out the temperature among the drops, measurement of the temperature of the largest drops appears to be adequate in determining the volumetric heat transfer coefficients.

Referring to Figure 1 again, we may write the heat transferred between the two phases as

$$Q_w = W_w C_{pw} \Delta T_w = Q_o = W_o C_{po} \Delta T_o \qquad (1)$$

The average of the heat taken up by the water and that given up by the oil was then used with a log mean temperature difference and the test section volume V to calculate a volumetric heat transfer coefficient:

$$\frac{UA}{V} = \frac{(Q_w + Q_o)/2V}{(\Delta T_{in} - \Delta T_{out})/\ln \dfrac{\Delta T_{in}}{\Delta T_{out}}}. \qquad (2)$$

EXPERIMENTAL RESULTS AND DISCUSSION

Data on direct contact heat transfer between 90 Neutral Oil and water and between 100 Spray Base and water were taken for a variety of flow conditions. In the following sections we discuss the dependence of the volumetric heat transfer coefficient on the volume fraction of water, the total linear velocity, the pipe diameter, the interfacial tension, the inlet oil viscosity (that is, inlet temperature), and the pipe length.

Volume Fraction of Water

Most of the heat transfer data in this study were taken at water volume fractions of 0.2 to 0.6. In this region, the water volume fraction was found to have little effect on UA/V. This lack of dependence of the volumetric heat transfer coefficient on the dispersed phase volume fraction is in agreement with the observations of Grover and Knudsen (4). However, for ϕ_w less than 0.2 or greater than 0.8, the volumetric heat transfer coefficient was found to increase roughly linearly with increasing dispersed phase volume fraction. This effect is illustrated in Figure 2 where we have plotted UA/V as a function of ϕ_w for the heat transfer between 90 Neutral oil and water flowing in the 0.325-in. diameter pipe at a total linear velocity of 15 ft./sec.

For low volume fractions of the dispersed phase where coalescence is not important, it would seem correct to assume that the interfacial area is roughly proportional to the amount of dispersed phase present. The increase in UA/V with dispersed phase volume fraction for $\phi_w < 0.2$ or > 0.8 therefore reflects the increase in interfacial area. At higher volume fractions of the dispersed phase there are two opposing factors. Increasing the amount of dispersed phase tends to increase the interfacial area, but due to more frequent coalescence the average drop size is increased so that the rate of increase is significantly less than the first power of the volume fraction. In addition, the freedom of movement of the drops relative to the continuous phase would be restricted at higher dispersed phase volume fractions and this should lead to a decrease in the heat transfer coefficient to an individual drop.

The vertical dashed line in Figure 2 at $\phi_w \approx 0.3$ which separates the diagram into an oil continuous region and a water continuous region was determined (by an extrapolation procedure) as the lowest volume fraction of water at which a significant amount of current could be passed between two electrodes placed in the two phase flow. It is perhaps surprising to note that in this case and in all

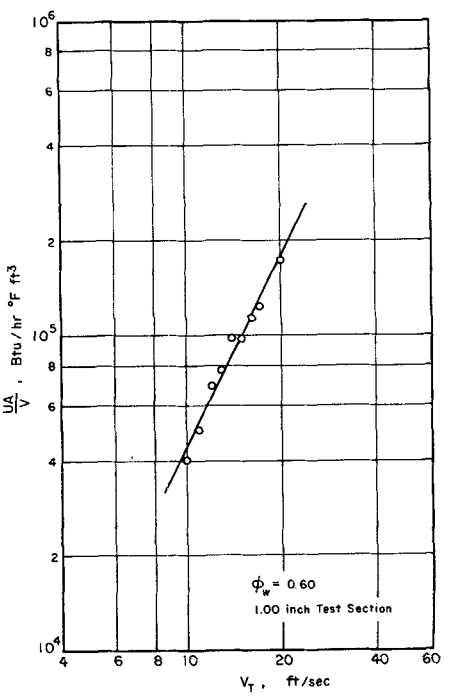

Fig. 3. Typical plot of volumetric heat transfer coefficient as a function of the total linear velocity. The conditions shown are for 90 Neutral oil-water flowing in the 1.00-in. diameter test section with the volume fraction of water being 0.6.

TABLE 3. LEAST SQUARES SLOPE FOR LOG (UA/V) VS. LOG V_T

System	Pipe diameter, in.	No. of points	Slope
90 Neutral oil-water	3.00	8	2.0
90 Neutral oil-water	1.50	37	2.0
90 Neutral oil-water	1.00	51	2.1
90 Neutral oil-water	0.531	21	1.8
90 Neutral oil-water	0.325	13	1.7
100 Spray base-water	1.00	9	2.2
100 Spray base-water	0.325	6	1.6

others investigated there was no sharp change in the volumetric heat transfer coefficient as the identity of the continuous phase was changed. A possible explanation is the absence of a clear-cut transition point at $\phi_w = 0.3$ as suggested by the observation of large fluctuations in current near this water volume fraction.

Velocity

Heat transfer data were taken at various linear velocities, all other parameters being held constant. Plots of the volumetric heat transfer coefficient vs. total linear velocity on log-log paper were found to give straight lines with slopes ranging from 1.6 to 2.2. A typical plot is shown in Figure 3. This plot is for the heat transfer between 90 Neutral Oil and water flowing in the 1.00-in. diameter test section with the volume fraction of water being 0.6. Bearing in mind that the volumetric heat transfer coefficient was independent of ϕ_w for $0.2 < \phi_w < 0.8$, we next plotted UA/V vs. total linear velocity for all the 90 Neutral oil-water data for the 1.00-in. diameter pipe having ϕ_w this range. This is shown in Figure 4. The slope of the least squares line for the fifty-one points on this plot was found to be 2.10. The data for the other test sections were treated in the same way. See, for example, Figure 5 where the 90 Neutral oil-water data for the 0.325-in. diameter test section are given. Least squares slopes were calculated for all heat transfer data taken; the slopes are listed in Table 3. The slopes for the two smallest pipes are significantly lower than those for the other three, and an examination of the data indicated that this is probably due to a falling off of the rate of increase of the volumetric heat transfer coefficient as the linear velocity is raised above a certain level in these two pipes. If we restrict ourselves to velocities less than 15 ft./sec., the data for the 0.325-in. diameter test section can be adequately represented by a line with slope 2.1. This is the slope of the line shown in Figure 5.

Grover and Knudsen (*4*) found that the volumetric heat transfer coefficient varied roughly proportional to the 1.6 power of the linear velocity for three different mixing conditions. However, their velocities were sufficiently low that stratification of the two fluids occurred in the pipe. The higher exponent found in this work may be ascribed to the higher degree of turbulence accompanied by more intimate mixing and greater heat transfer.

Pipe Diameter

The data for each pipe were plotted separately as log UA/V vs. log V_T as described above and as shown in Figures 4 and 5. Then lines of slope 2.1 were drawn through the data. It was found that the three largest pipe diameters gave approximately equal volumetric heat transfer coefficients for a given linear velocity. As will be seen shortly, there is the possibility of a significant entry length effect and the heat transfer coefficient may decrease substantially with distance from the mixing tee. Thus the data for the 3.00- and 1.50- (and perhaps the 1.00-) in. diameter pipes might be expected to overestimate the heat transfer coefficients which would obtain for large pipe lengths and to

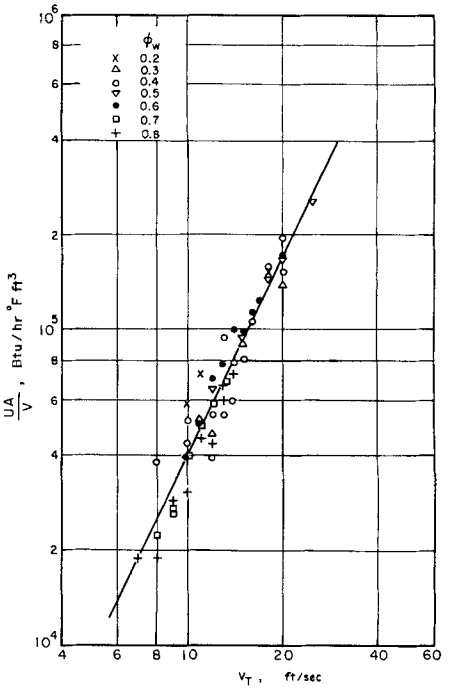

Fig. 4. Volumetric heat transfer coefficient as a function of total linear velocity for 90 Neutral oil-water flowing in the 1.00-in. diameter test section with $0.2 < \phi_w < 0.8$.

give unreliable results for the evaluation of the diameter effect since only 28 and 90 (and 108) pipe diameters were contained in these pipe lengths, respectively.

The three smallest test sections were then used to evaluate the diameter dependency of the volumetric heat transfer coefficient. Figure 5 shows UA/V vs. V_T for 90 Neutral oil-water for each of these pipes; the dashed lines are the lines of slope, 2.1, through the experimental data for the indicated pipes. It is seen that the volumetric heat transfer coefficient decreases with increasing pipe diameter at a fixed linear velocity. The power on the diameter found

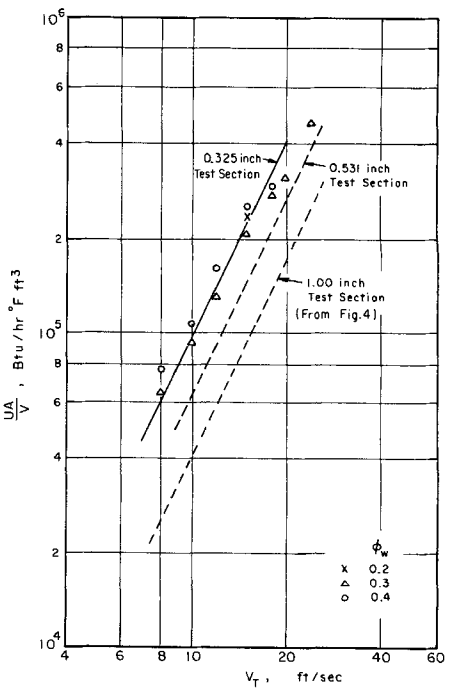

Fig. 5. Volumetric heat transfer coefficient as a function of total linear velocity for different pipe diameters.

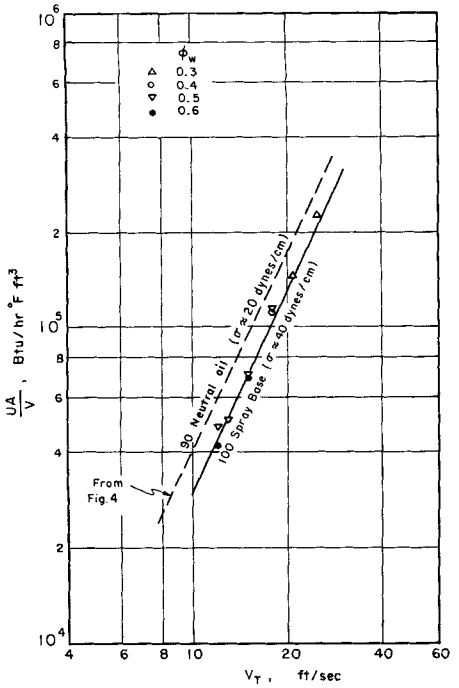

Fig. 6. Volumetric heat transfer coefficient as a function of total linear velocity for two different interfacial tensions.

necessary to superimpose the 0.325- and 0.531-in. data is —0.89. The exponents on the diameter necessary to superimpose the 0.325- and 1.00-in. data and the 0.531- and 1.00-in. data are —0.78 and —0.71, respectively. It is reasonable that the volumetric heat transfer coefficient decreases with increasing pipe diameter at a fixed velocity primarily because the drops would become larger and there would be less interfacial area per unit volume available for heat transfer. (See the section on the correlation of data.)

Interfacial Tension

The effect of the interfacial tension on the volumetric heat transfer coefficient was investigated by using two oils with significantly different interfacial tensions with water but with otherwise quite similar physical properties. As measured with a ring tensiometer, the interfacial tension for the fresh 90 Neutral oil-water system at 100°F. was 19 dynes/cm. and the interfacial tension of the fresh 100 Spray Base-water system was 40 dynes/cm. Impurities in the system can have a marked effect on the interfacial tension, and because of the large volumes being handled no attempt was made to keep the liquids clean. In fact, measurements of the interfacial tensions after the liquids had been used showed that both had been reduced significantly (but in the same ratio). Consequently, the influence of the interfacial tension on the volumetric heat transfer coefficient found here can be regarded only *semi-quantitatively.*

Heat transfer data were taken for both heat transfer systems flowing in the 1.00- and 0.325-in. diameter pipes. The data were treated as before, by plotting log UA/V vs. log V_T, and drawing a slope of 2.1 through the data for the 1.00-in. diameter pipe and a slope of 1.7 through the 0.325-in. diameter pipe. See, for example, Figure 6 where the data for the 1.00-in. diameter pipe are presented. The exponent on σ necessary to effect coincidence of the two systems is —0.43 and —0.25 for the 1.00- and 0.325-in. pipes, respectively. Again, one would expect the volumetric heat transfer coefficient to be decreased as the interfacial tension is raised, all other factors held constant, because of the decrease in the interfacial area.

Inlet Oil Viscosity (Temperature)

To examine the effect of the oil inlet viscosity on the volumetric heat transfer coefficient, data were taken with the oil inlet temperatures varying from 100° to 190°F.; that is, the inlet oil viscosity varied from 36 to 10 lb./(hr.)(ft.). Of course, the other physical properties also vary with temperature, but Table 2 shows the variation to be small compared with that of the viscosity, the largest variation occurring in the surface tension which changes by about 30%. For this range of variables no significant variation of the volumetric heat transfer coefficient was found.

Pipe Length

For several runs, thermocouple traces were obtained at various distances along the test section. The results for one of these experiments, 90 Neutral oil-water flowing in the 1.50-in. diameter test section at a velocity of 9.3 ft./sec. and water volume fraction of 0.22, are shown in Figure 7 where the local temperatures of the two phases are plotted vs. pipe length. The volumetric heat transfer coefficient calculated at each point is also indicated on the plot. The figure and similar data indicate the possibility of a significant entrance effect giving larger volumetric heat transfer coefficients for small pipe lengths. Unfortunately, the effect of pipe length was not investigated in detail and there are insufficient data to establish the magnitude of the entrance effect.

CORRELATION OF DATA

Any attempt to predict the rate of heat transfer between two immiscible liquids in turbulent pipe flow must take into account two principal factors: the dependence of the

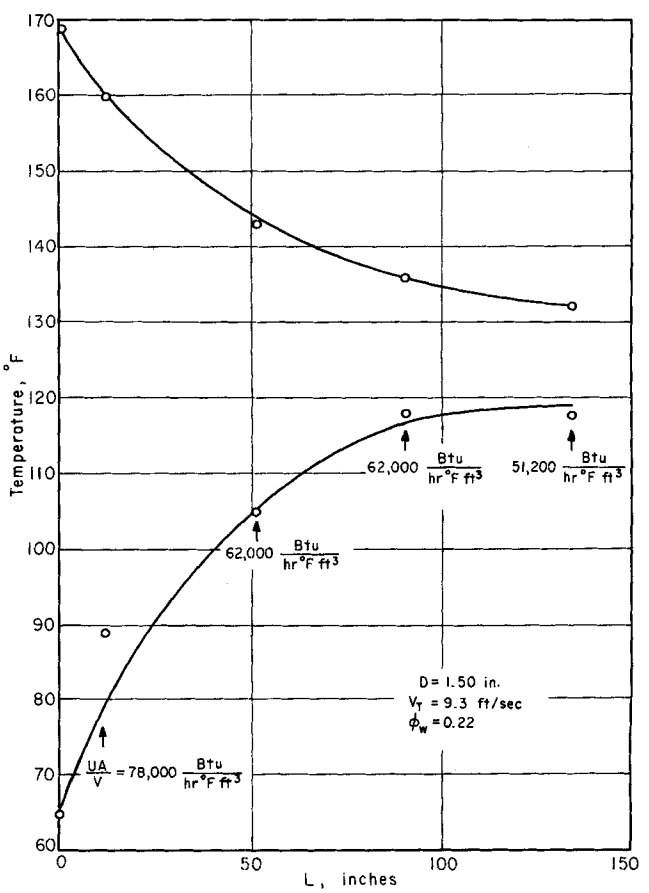

Fig. 7. Typical temperature profile giving temperatures as functions of the distance from the mixing tee.

drop size (hence the interfacial area) on the physical properties and operating conditions, and the rate of heat transfer to or from an individual drop. Results from the statistical theory of turbulence have been applied with success in estimating a characteristic drop size in liquid-liquid turbulent flow. For a review of the treatment, one should consult Hinze (3) and Levich (4). When applied to two-phase liquid-liquid pipe flow of liquids of low viscosity, the treatment gives the characteristic drop size d as

$$d/D \sim We^{3/5} \sim (\sigma/\rho_c V_T^2 D)^{3/5} \tag{3}$$

where ρ_c is the density of the continuous phase which is assumed to be very nearly that of the dispersed phase. The proportionality factor is of the order of unity. This estimate is valid only for very large Reynolds numbers and for drops larger than the microscale of the turbulence for which conditions viscous forces are not important in the breakup of the drops. For brevity we have omitted a factor in Equation (3) given by the 0.08 power of the Reynolds number (12); this is justified since it introduces only minor changes in the dependence of the volumetric heat transfer coefficient on the variables. With the proportionality factor recommended by Hinze and the 0.08 power of the Reynolds number included, the drop diameter predicted by Equation (3) is about 1.2×10^{-3} ft. for 90 Neutral oil-water flowing in a 1-in. diameter pipe at 10 ft./sec. A characteristic relative velocity u for two fluid elements separated by a distance d will be needed in the ensuing development; according to the statistical theory of turbulence this is given by

$$u/V_T \sim (d/D)^{1/3} \tag{4}$$

Simple volumetric considerations show that if the drops can be treated as having a representative diameter d, the interfacial area in the test section of volume V is

$$A \sim \phi V/d \tag{5}$$

where ϕ is the volume fraction of the dispersed phase. As noted above, one should expect A to increase linearly with ϕ only for small values of ϕ. Denoting the external, internal, and overall heat transfer coefficient for an individual drop as h_c, h_d, and U, respectively, with $1/U = 1/h_c + 1/h_d$, we may write

$$\frac{UA}{V} \sim \frac{U\phi}{d} \sim \frac{\phi/d}{\frac{1}{h_c}+\frac{1}{h_d}} \sim \frac{\phi/d^2}{\frac{1}{k_c N_{Nu_c}}+\frac{1}{k_d N_{Nu_d}}} \tag{6}$$

In order to proceed we need to make some estimate of the heat transfer coefficients for an individual drop. For this purpose we assume that the heat transfer coefficients are given by standard correlations using the drop diameter d and relative velocity u as the characteristic parameters. This type of analysis has recently been advanced by Middleman (5) and was used successfully to correlate mass transfer data for spheres of ion exchange resin in a stirred tank. Sideman and Barsky (10) have used the statistical theory of turbulence to estimate the drop size in a liquid-liquid system where one of the liquids is evaporating. Some of these ideas were also used by Heuss et al. (2) in gas-liquid mass transfer in cocurrent froth flow.

To illustrate the procedure, let us assume the external heat transfer is given a correlation of the well-known form (8)

$$N_{Nu_c} = \frac{h_c d}{k_c} = a' + b'\left(\frac{C_{pc}\mu_c}{k_c}\right)^{1/3}\left(\frac{du\rho_c}{\mu_c}\right)^{1/2} \tag{7}$$

and the internal heat transfer is given by (2)

$$N_{Nu_d} = \frac{h_d d}{k_d} = c' \tag{8}$$

These correlations, which are thought to be the most appropriate for the small drops forming the dispersion, represent a combination of conduction and forced convection for the external heat transfer and conduction only (with the approach to equilibrium greater than 30 to 40%) for the internal heat transfer. An extended review of heat transfer to individual drops is given by Sideman and Shabtai (9). When these correlations are substituted into Equation (6) and when the quantities d and u are eliminated by use of Equations (3) and (4), we find

$$\frac{UA}{V} = \frac{\phi D^{-2} N_{We}^{-6/5}}{\{k_c [a + bN_{Pr_c}^{1/3} N_{Re}^{1/2} N_{We}^{2/5}]\}^{-1} + \{k_d c\}^{-1}} \tag{9}$$

where a, b, and c are dimensionless constants. It frequently happens that one phase or one mechanism is heat transfer

Table 4. Comparison of the Exponents for UA/V from Various Heat Transfer Models and Experiment

Model	ϕ	V_T	D	μ_c	Variable σ	ρ_c	k_c	C_p	k_d	μ_d
External phase controlling, conduction: N_{Nu_c}~const. that is, $b=o$, $c=o$ in Equation (9)										
Drop size predicted from Equation (3)	1	2.4	−0.8	0	−1.2	1.2	1	0	0	0
Drop size predicted from Equation (15)	1	5.0 to 3.6	0	1	−3.0 to −1.6	2.0	1	0	0	0 to −1.4
External phase controlling, forced convection: $N_{Nu_c} \sim N_{Pr}^{1/3} N_{Re}^{1/2}$ that is, $a=o$, $c=o$ in Equation (9)										
Drop size predicted from Equation (3)	1	2.1	−0.7	−0.167	−0.8	1.3	0.667	0.333	0	0
Drop size predicted from Equation (15)	1	3.8 to 2.9	−0.167	0.5	−2. to −2.17	3.67	0.667	0.333	0	0 to −0.933
Internal phase controlling, conduction: N_{Nu_d}~const. that is, $a=o$, $b=o$ in Equation (9)										
Drop size predicted from Equation (3)	1	2.4	−0.8	0	−1.2	1.2	0	0	1	0
Drop size predicted from Equation (15)	1	5.0 to 3.6	0	1	−3.0 to −1.6	2.0	1	0	0	0 to −1.4
Range of experimentally observed exponents	1 or 0	1.7 to 2.1	−0.89 to −0.71	0	−0.43 to −0.25	x	x	x	x	x

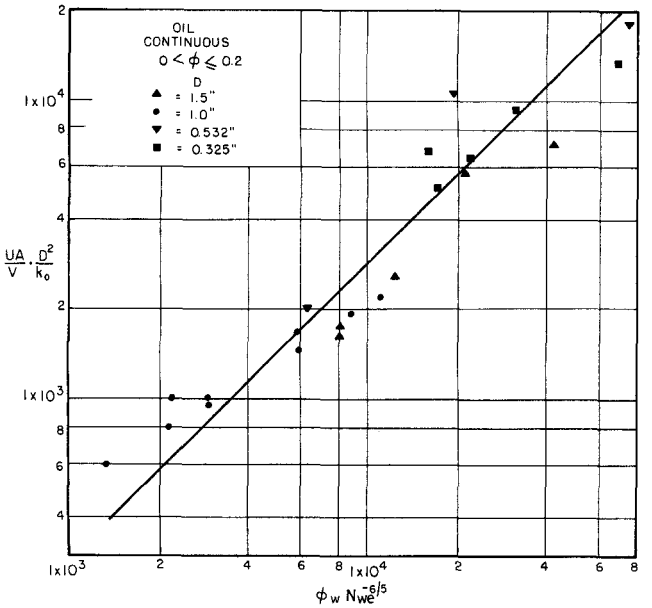

Fig. 8. Correlation of the heat transfer coefficient for oil being the continuous phase: $0<\phi_w<0.2$.

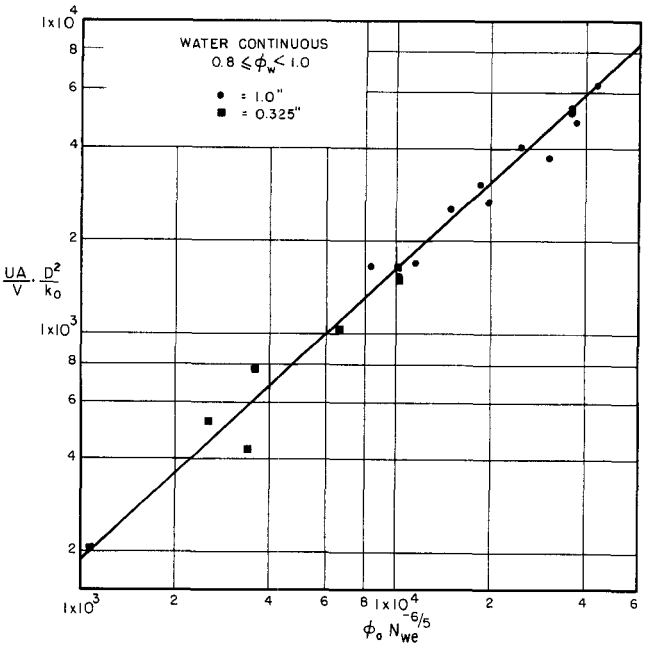

Fig. 10. Correlation of the heat transfer coefficients for water being the continuous phase: $0.8<\phi_w<1.0$.

controlling. Table 4 shows the exponents predicted from Equation (9) on the variables on which (UA/V) depends when one of the mechanisms is controlling. The table also gives the range of exponents found experimentally. It is seen that the predicted exponents on V_T, D, and μ are very nearly the same for the three models and are also near the experimental values. There is considerable deviation for the exponent on the surface tension and this is ascribed to the contamination of the fluids. Thus, with the data at hand, it cannot be inferred that one or the other of the mechanisms presented here is controlling, and they each do an adequate job in correlating the available data. However, due to the low thermal conductivity of the oils, it is likely that the major resistance is always in the oil phase (6). With this in mind, the data for the 90 Neutral oil-water system was divided into four groups and treated as follows:

1. For $0 \leq \phi_w \leq 0.2$ (oil continuous) log (UAD^2/Vk_o) was plotted vs. log $\phi_w N_{We}^{-6/5}$. This is shown in Figure 8.
2. For $0.4 \leq \phi_w \leq 0.8$ (water continuous) we plot $\log(UAD^2/Vk_o)$ vs. log $N_{We}^{-6/5}$ where the volume fraction of the dispersed phase is omitted, because in this region it is found to have no effect on the volumetric heat transfer. This is shown in Figure 9.
3. For $0.8 \leq \phi_w \leq 1.0$ (water continuous) we plot $\log(UAD^2/Vk_o)$ vs. log $\phi_o N_{We}^{-6/5}$ in Figure 10.

The physical properties used in making these plots were for a temperature of 150°F. and the interfacial tension of the fresh oils. It is seen that all three plots can be adequately fitted with straight lines with a slope of unity. The major reason for the scatter in the plots is the lack of diameter dependence of UA/V for the larger pipes, and the line in Figure 9 is drawn through the data for the three smaller pipes. The equations found may be summarized as follows:

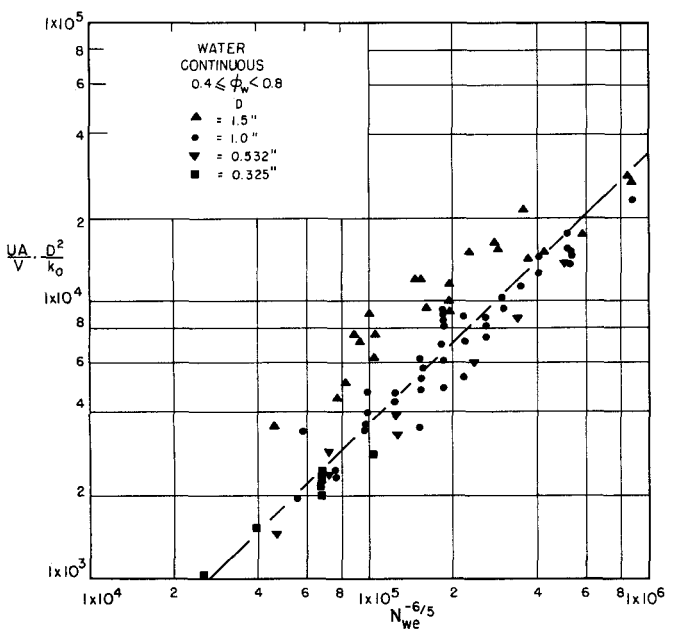

Fig. 9. Correlation of the heat transfer coefficients for water being the continuous phase in the region where UA/V is independent of ϕ: $0.4<\phi<0.8$.

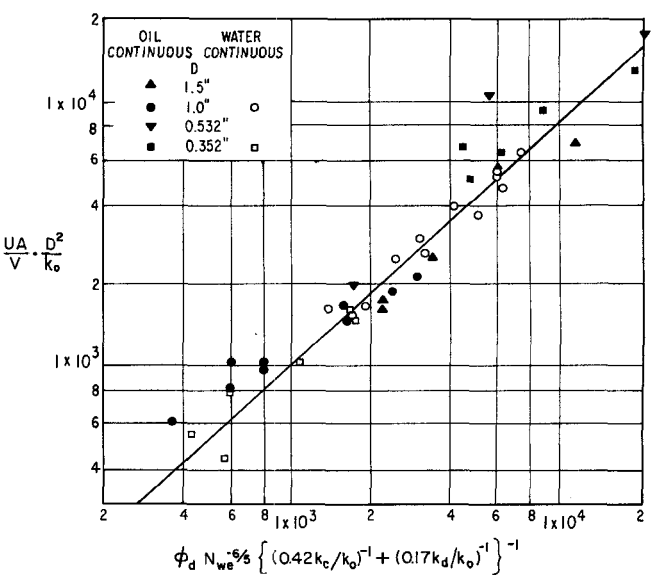

Fig. 11. Final correlation of the heat transfer coefficients with the dispersed phase volume fraction less than 0.2.

$$0 \leq \phi_w \leq 0.2 \quad \frac{UAD^2}{Vk_o} = 0.28\,\phi_w N_{We}^{-6/5} \qquad (10)$$

$$0.2 \leq \phi_w \leq 0.4 \quad \text{No correlation sought, but data could be approximated by (12).} \qquad (11)$$

$$0.4 \leq \phi_w \leq 0.8 \quad \frac{UAD^2}{Vk_o} = 0.033\,N_{We}^{-6/5} \qquad (12)$$

$$0.8 \leq \phi_w \leq 1.0 \quad \frac{UAD^2}{Vk_o} = 0.16\,\phi_o N_{We}^{-6/5} \qquad (13)$$

Bearing the form of Equations (6) or (9) in mind and considering the heat to be transferred by conduction only in both phases, we can combine Equations (10) and (13) to give the correlation

$$\frac{UAD^2}{Vk_o} = \frac{\phi_d\,N_{We}^{-6/5}}{(0.415\,k_c/k_o)^{-1} + (0.173\,k_d/k_o)^{-1}} \qquad (14)$$

The data on which this correlation is based are shown in Figure 11.

Again, it should be pointed out that an equally good correlation can be obtained with the group $\phi N_{Pr}^{1/3} N_{Re}^{1/2} N_{We}^{-4/5}$. These correlations should be regarded only as tentative until more data on the effect to fluid physical properties, pipe length, and pipe diameter are forthcoming.

One of the reviewers has pointed out that the conditions of isotropic turbulence and of sufficiently high Reynolds number for there to be an inertial subrange of the turbulent energy spectrum are not met in the present experiments for Equation (3) to be theoretically justified. He suggested instead of Equation (3) the use of the correlation of Sleicher (*12*) and Paul and Sleicher (*6*):

$$d\rho_c V_T^2/\sigma = C(\sigma/\mu_c V_T)^{0.5}\{1 + 0.7\,(\mu_d V_T/\sigma)^{0.7}\} \qquad (15)$$

where C is at most very weakly dependent upon the pipe diameter. This correlation was established from the observation of the breakup of individual drops introduced into turbulent pipe flow and it was observed that breakup occurred predominately near the pipe wall. It predicts the drop diameter to be independent of the pipe diameter and to vary with the −1.8 to −2.5 power of the velocity, whereas Equation (3) predicts the drop diameter to vary with the 0.4 power of pipe diameter and the −1.2 power of velocity. The exponents predicted from Equation (6) on the variables one which UA/V depends when Equation (15) is used for the drop size are shown in Table 4. Two numbers are given depending upon whether the first or second term in Equation (15) dominates. It is seen that this correlation does a significantly poorer job in predicting the observed exponents on the variables studied, V_T, D, μ_c, and σ, than does Equation (3) for each of the three heat transfer models considered. The one exception is for the pipe diameter dependence for the larger pipes where Equation (15) predicts a zero or small dependence and a zero dependence was found experimentally.

Does this mean that Equation (3) does a better job in predicting the drop size in the present system than Equation (15)? It may be that the drop size is controlled by the inlet conditions at the mixing tee where intense mixing was observed and where conditions may be more appropriate for Equation (3) to hold than in the test section proper. We do not have any direct information on the change of drop size with pipe length, although the slight decrease in the volumetric heat transfer coefficient with pipe length may be indirect evidence of an increase in the average drop size. In a study with hydrodynamics slightly different than those in the present work, but in the same Reynolds number range as this and Sleicher's work, Sjoblom and Goren (*10*) found a velocity dependence of the −1.28 power for the drop diameter using light scattering as a measure of the average drop diameter. Unfortunately the velocity was the only parameter varied in that study.

ACKNOWLEDGMENT

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NOTATION

a', b', c', a, b, c = constants
A = interfacial area per unit volume, sq.ft./cu.ft.
C_p = heat capacity, B.t.u./(lb.)(°F.)
d = drop diameter, ft.
D = pipe diameter, ft.
h = heat transfer coefficient, B.t.u./(sq.ft.)(hr.)(°F.)
k = thermal conductivity, B.t.u./(ft.)(hr.)(°F.)
N_{Nu} = Nusselt number, hd/k
N_{Pr} = Prandtl number, $\rho C_p/\mu$
N_{Re} = Reynolds number, $\rho V_T D/\mu$
N_{We} = Weber number, $\sigma/\rho D V_T^2$
Q = heat transfer rate, B.t.u./hr.
T = temperature, °F.
ΔT = difference between temperature after heat transfer and the inlet temperature
U = overall heat transfer coefficient, B.t.u./(sq.ft.)(hr.)(°F.)
V = volume of test section, cu.ft.
V_T = linear velocity in pipe, ft./sec.
W = mass flow rate, lb./hr.

Greek Letters

μ = viscosity, lb./(ft.)(hr.)
ϕ = volume fraction
ρ = density, lb./cu.ft.
σ = surface tension, dyne/cm.

Subscripts

c = continuous phase
d = disperse phase
o = oil phase
w = water phase

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