Reply to Professor McKelvey's Note

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Professor McKelvey's note shows that the diffusion process could contribute as much as 12% of the total resistance to mass transfer at very low heat flux, based on a Nusselt number of 7.4 and the limiting slope of the graph for run V. However, based on the same Nusselt number and the same equation, the ratio of diffusion resistance to total resistance would vary from 0.16 to 0.55 for the individual data points of run V. For data points from the other four runs this ratio is even higher, and the ratio exceeds 1 for several points of runs I and II. A ratio of 1 or greater means that all the resistance could be attributed to diffusion. A ratio greater than 1 means that the Nusselt number *is actually greater than the assumed* value of 7.4.

As another way of showing the probable importance of diffusion, the mass transfer coefficients have been calculated for typical data from runs I to V, with equilibrium assumed at the interface. The calculated Nusselt numbers range from 4 to 40, generally increasing with the over-all temperature drop. The increase could be explained partly by the increased density difference and partly by the increased flow of vapor in the system. The expected Nusselt numbers for natural convection would be about 5 to 10; the exact values are not easily calculated since the lower molecular weight at the interface tends to offset the effect of the lower temperature at the interface. The fact that the calculated values are about the same as those expected for natural convection is strong evidence that diffusion is important. Eddy currents in the condenser might make the Nusselt numbers 2 to 3 times the values for natural convection, but it seems unlikely that they would be 10 to 20-fold higher, as would be required to make the diffusion resistance negligible. (To get a Nusselt number of 100 would require a Reynolds number of about 20,000 either for flow in a pipe or flow past a flat plate.)

Nusselt Numbers for Condensation of Methanol with Air Present (Data of McKelvey and Baer)

Run	ΔT_1, °C.	$r \times 10^3$, g./sq. cm.-sec.	*Apparent Nusselt number
I*a*	0.59	0.56	41
I*f*	1.48	1.40	40
II*a*	0.53	0.49	12.7
IIf	1.56	1.44	20
III*a*	0.40	0.37	6.5
III*h*	1.33	1.23	13.3
IV*a*	0.40	0.37	5.8
IV*g*	1.44	1.33	12.9
V*a*	0.26	0.24	4.0
V*d*	0.70	0.64	7.1
V*g*	1.23	1.13	10.4

* Based on a length of 6 cm and D = 0.48 sq. cm./ sec. at 0.37 atm

CHEMICAL ENGINEERING PROGRESS SYMPOSIUM SERIES ABSTRACTS

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The *A. I. Ch. E. Journal* will publish, from time to time, abstracts of the articles appearing in the Symposium Series volumes. Recently published volumes are abstracted below.

HEAT TRANSFER—CHICAGO, Vol. 55, No. 29, 1959.

Hard-Water Scaling of Finned Tubes at Moderate Temperatures, J. G. Knudsen and H. K. McCluer. With the increasing use of finned tubes in industrial heat transfer equipment it is important to know their heat transfer characteristics. A list of recommended fouling coefficients to be used in the design of plain-tube exchangers has been published; however very little information on the behavior of finned tubes under scaling conditions has been reported. This investigation is concerned with a study of the rate of scaling of transverse finned tubes in a double-pipe heat exchanger. **The Effect of Thermal Cycling of Integral-Finned Duplex Tubes,** Edwin H. Young and Marvin L. Katz. The effect of thermal cycling to 350° and 600°F. on the heat transfer performance of duplex integral-finned tubes is reported. The primary heat transfer variable affected by thermal cycling was the bond resistance of the tubes. The phenomenon of bond resistance is discussed, and an apparatus for the measurement of bond resistance is described. Curves indicating the variation in bond resistance as a function of the number of thermal cycles for (a) four copper-liner tubes cycled to 350°F., (b) ten admiralty-liner tubes cycled to 350°F., and (c) four admiralty-liner tubes cycled to 600°F. are presented and discussed. The effect of bond resistance on the over-all coefficient of an air cooler is indicated. **Coefficients for Evaporation of Neutral Sulfite Spent Liquors,** S. T. Han, B. D. Andrews, and W. G. Dedert. This paper presents the results of an investigation of the evaporation of neutral sulfite spend pulping liquor in a semicommercial, forced-circulation, external heat-exchanger evaporator. A series of runs was made at each of four conditions simulating quadruple-effect operation. The principal variable studied at each condition was the liquor velocity through the tubes of the heat exchanger. Data are presented on over-all heat transfer coefficients, and these are analyzed to yield film coefficients on both the liquor and the steam sides.

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The liquor-side coefficients indicate that little boiling occurred in the tubes. **Description and Experimental Results of Two Regenerative Heat Exchangers,** E. K. Dabora, M. P. Moyle, R. Phillips, J. A. Nicholls, and P. L. Jackson. Two pebble-type regenerative heat exchangers capable of operating at pressures and temperatures beyond the limits of commercially available equipment have been designed and constructed by personnel of the Aircraft Propulsion Laboratory. These heat exchangers were required to produce stagnation temperatures of about 2,500°R. at pressures of about 1,000 lb./sq.in. in experiments designed to achieve a standing detonation wave. In the initial operation the outlet temperature of the small (hydrogen) heat exchanger was considerably less than anticipated, and a theory was therefore developed to predict the performance of a regenerative heat exchanger with heat loss. The design and operating characteristics of the two heat exchangers are presented in this report, and their performance is compared with the analysis. The experimental results agree with the theory. **Local Shell-Side Heat Transfer Coefficients in the Vicinity of Segmental Baffles in Tubular Heat Exchanges,** M. S. Gurushankariah and J. G. Knudsen. Local heat transfer coefficients were studied in detail in the central baffle space between two segmented baffles in a tubular heat exchanger. Two baffle spacings and three flow rates were investigated. Average over-all Nusselt numbers determined from the measured local coefficients compare favorable with average Nusselt numbers measured by other workers. The data are presented in picture form, which permitted a schematic diagram of the flow pattern in the baffle space to be drawn. The results indicate the presence of three flow zones between two baffles. The longitudinal-flow zone occurs in the baffle windows; the cross- and eddy-flow zones occur on the downstream and upstream sides of the baffle space respectively. Heat transfer rates were in general higher in the eddy-flow zone. The heat transfer rate varies along individual tubes from a maximum at the baffle to a minimum midway in the baffle. The minimum value is one-fourth to one-half the value at the baffle. **Heat Transfer and Pressure Drop of Air in Forced Convection Across Triangular Pitch Banks of Finned Tubes,** D. J. Ward and E. H. Young. The effects of tube geometry on the heat transfer and pressure-drop characteristics of equilateral-triangular-pitch tube banks containing smooth, integral, helically finned tubes with air drawn by forced convection in cross flow through the banks were studied in seven finned-tube banks ranging from four to eight rows deep, with air velocities from 200 to 3,000 ft./min. based on the minimum cross section of the tube banks. The tubes used in the investigation ranged from ¾ to 2¼ in. in diameter over the fins, with tube spacings equal to the nominal fin diameters plus 3/16 in. for all tube banks. The velocity distribution and turbulence of the air approaching the tube banks were also studied. A single plain tube bank was included in the study for comparison with previous investigations, and rates of heat transfer from individual rows of tubes in one tube bank were investigated. The air-pressure-drop data were compared with a relationship proposed by Gunter and Shaw. An alternate correlation is also presented. **Heat Transfer and Fluid Friction During Flow Across Banks of Tubes: VII. Bypassing Between Tube Bundle and Shell,** O. P. Bergelin, K. J. Bell, and M. D. Leighton. This paper presents information on the flow between the outer row of tubes and the shell wall in rectangular tube banks. The results presented include isothermal, heating, and cooling runs in both laminar and turbulent regimes for two tube-bank geometries with various bank-to-shell clearances. The effect of sealing strips to block the bypass stream is also shown. Methods of generalizing the results for commercial exchanger design are considered. **Forced-Convection, Local-Boiling Heat Transfer in Narrow Annuli,** Louis Bernath and William Begell. Heat transfer data in the region of fully developed local boiling have been collected, analyzed, and correlated. The results of this analysis permit the prediction of the wall superheat for the range of the variables studied. A proposed local boiling film concept is shown to agree with experimental observations. **Performance of Vaporizers: Heat Transfer Analysis of Plant Data,** Charles H. Gilmour. This paper records observed data from several vaporizers in a chemical plant and shows how the data are analyzed on the basis of heat transfer theory to establish the mechanism, to determine the probable film coefficients, and to obtain data on fouling. **Heat Transfer Rates to Boiling Freon 114 in Vertical Copper Tubes,** H. L. Foltz and R. G. Murray. The prediction of heat flux along the length of a tube boiler is a problem particularly in the design of high-efficiency gas coolers in which heat is transferred from the gas to a boiling liquid. Heat transfer rates from condensing steam to boiling Freon 114 were measured in vertical copper tubes to determine where the assumption of uniform heat flux could be safely used and how tube length, tube diameter, flow rates, pressure, and temperature difference affected uniformity. **The Prediction of Surface Temperatures at Incipient Boiling,** S. G. Bankoff. Superheat required for bubble nucleation with respect to surface conditions is considered. It is shown that cavity type of surfaces may require relatively low superheats which are determined by the cavity dimensions but that grooves, which are the more common type of primary roughness element, are ineffective vapor traps unless very poorly wetted or steep walled. An approximate theory derived for predicting superheats required for initiation of ebullition at low pressures, based on a limiting real solution of the equations for the rate of penetration of the liquid into the capillary roughness element. A semiempirical expression for minimum superheat at elevated pressures is given. Agreement is shown with the literature within the experimental uncertainties involved. These observations should also be pertinent to cavitation phenomena. **Bubble Growth Rates in Highly Subcooled Nucleate Boiling,** S. G. Bankoff and R. D. Mikesell. By comparison of the Rayleigh solution for bubble growth and collapse with experimental data, it is found that the pressure of the vapor within bubbles arising in very subcooled nucleate boiling is less by a small, fairly constant amount than the pressure at a great distance. The effect is the same as if the bubble surroundings were imparted an initial impulse while the minute bubble was still entirely within the laminar-wall layer. The kinetic energy imparted by this initial impulse and the restricting pressure difference determine the bubble trajectory. These parameters are computed for Gunther's data. It is then postulated that the heat flux from the portion of the bubble projecting into the turbulently-flowing core depends primarily upon turbulent and convective heat flow rather than laminar heat conduction and hence is relatively constant during the bubble lifetime. The proposed mechanism gives qualitative agreement with the observed trends. It is suggested that latent heat transport may be an important mode of heat transfer in subcooled nucleate boiling. **The Effects of Superimposed Forced and Free Convection on Heat Transfer and Pressure Drop in a Horizontal Rectangular Duct,** M. Altman and F. W. Staub. Average and local heat transfer coefficients and flow friction data are reported. Variables include a range of Grashof and Reynolds numbers based on equivalent duct diameter. Axial wall temperature distributions, duct position, and inlet

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shape were varied. Average heat transfer coefficients and duct friction factors were correlated for both linear and nonlinear axial wall temperature distributions. **The Effects of Superimposed Forced and Free Convection on Heat Transfer in a Vertical Rectangular Duct,** M. Altman and F. W. Staub. Average and local experimental heat transfer coefficients are reported for a vertical duct of rectangular cross section where the mean air flow is from the top to the bottom of the duct and where heat is being added to the air at the walls. Results are given for a Reynolds number range of 1,800 to 10,000 based on the equivalent duct diameter for a duct length to equivalent diameter ratio of 69. The characteristic dip region in the transition zone between turbulent and laminar flow condition is absent for Grashof numbers greater than 2×10^5 based on diameter. **The Radial Heat Flux,** S. W. Churchill and R. E. Balzhiser. The ratio of the radial heat flux at any radius to the radial heat flux at the wall provides insight into the structure of heat transfer in fully developed flow in a tube and can be used to calculate the temperature field and local heat transfer coefficient for laminar and slug flow from the effective thermal conductivity. This important quantity is examined in detail for the first time and values are presented for slug, laminar, and turbulent flow with a uniform heat-flux density at the wall and with a uniform wall temperature. The explicit or implicit assumptions concerning the radial heat flux which have been made in the development of existing theoretical expressions for heat transfer are noted. **Heat Transfer in Scraped-Surface Exchangers,** Peter Harriott. A simple equation for the film coefficient of a scraped-surface exchanger is obtained from the theory for transient conduction in a thick slab. The theoretical coefficient is proportional to the square root of the heat capacity, density, thermal conductivity, and rotational speed. Experimental data for fluids of moderate viscosity show reasonable agreement with the theory, although lower coefficients for very viscous fluids or pastes would be expected. **Mechanically Aided Heat Transfer,** D. Q. Kern and H. J. Karakas. Heat transfer devices with moving parts implement phase changes to and from viscous Newtonian and non-Newtonian fluids. By combining certain principles of heat and mass transfer, hydrodynamics, and rheology, equations are developed for the design of these machines and the prediction of their performance. A numerical example illustrates the use of the derived equations. **Film Coefficients for Heat Transfer to Liquid Drops,** E. R. Elzinga, Jr., and J. T. Banchero. Heat transfer rates to drops in liquid-liquid systems have been experimentally measured, and the effects of physical properties and drop size on the continuous-phase heat transfer coefficient have been determined. Internal drop circulation has been shown to increase greatly the continuous-phase coefficient for some systems, and conversely the presence of surface-active materials reduces the continuous-phase coefficient by decreasing internal drop circulation. **Natural Convection in Horizontal Liquid Layers,** Ernst Schmidt and P. L. Silveston. Heat transfer through a horizontal liquid layer bounded on top by a cold surface and on bottom by a heated surface was measured, and results were correlated by dimensionless numbers. Optical observations were made on the patterns formed in the layer by convection. The data are satisfactorily correlated. **Condensing Heat Transfer Within Horizontal Tubes,** W. W. Akers, H. A. Deans, and O. K. Crosser. This investigation studied the effect of vapor velocity, temperature difference, and fluid properties upon the heat transfer coefficient of a vapor condensing within a horizontal tube. Condensing heat transfer coefficients for propane and Freon 12 were measured over a wide range of conditions. The data would not correlate when the existing equations for condensation were used. **Calculation of the Performance of a Mixed-Vapor Condenser by Analogue Computation,** N. G. O'Brien, R. G. Franks, and J. K. Munson. An analogue-computer program for computing the condenser size and composition changes encountered in the simultaneous condensation of a binary vapor mixture is described, and the effects on performance of liquid-phase resistance to mass transfer are computed. **Optimum Air-Fin Cooler Design,** D. Q. Kern. In an induced-draft air-fin cooler the heat transfer coefficient for each successive row of tubes normal to the air flow increases exponentially with the number of tube rows for a given air velocity. The air pressure drop and power consumption increase very nearly in direct proportion to the number of tube rows. Combining both functions yields equations which permit the design of the air-fin cooler having the lowest annual cost for any pay-out term. In this solution the number of rows and air velocity are dependent variables. Data and a numerical problem illustrate the method of application. **Transient Heat Conduction in Annular Fins of Uniform Thickness,** A. J. Chapman. The transient behavior of an annular cooling fin of uniform thickness is considered. For a cooling fin which is initially at the same temperature as the surroundings and is suddenly heated at its inner radius, equations are developed in a dimensionless form which give the distribution of the temperature in the fin, the heat removed from the source, the heat given up to the surroundings, etc., all as functions of time. These equations are also reduced to graphical form for the use of design engineers. **The Graetz-Nusselt Problem (With Extension) for a Bingham Plastic,** E. H. Wissler and R. S. Schechter. The Graetz-Nusselt problem is extended to include heat transfer to a slurry which behaves as a Bingham plastic. The eigenvalues, normalized expansion coefficients for constant inlet temperature, and the norms of the orthogonal functions are tabulated. These values permit the computation of heat fluxes and average fluid temperatures with and without internal heat generation. **Heat Transfer During Laminar Flow Past Flat Plates: An Extension of Pohlhausen's Solution to Low- and High-Prandtl-Number Fluids,** F. D. Fisher and J. G. Knudsen. The Pohlhausen solution for heat transfer during laminar flow past an isothermal flat plate has been extended to low- and high-Prandtl-number fluids. The integral obtained by Pohlhausen is rearranged so that it may be evaluated easily for low-Prandtl-number fluids.