An Experimental Investigation of Boiling Heat Transfer and Pressure-Drop Characteristics of Freon 11 and Freon 113 Refrigerants

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Vapor cycle cooling equipment is currently being used in certain high-performance aircraft and in military ground cooling applications. Its use for space-vehicle cooling applications has also been suggested. Because of the importance of efficiency, weight, and volume limitations in the above applications, the use of centrifugal compressors is indicated. Among the commonly used and readily available refrigerants, Freon 11 and Freon 113 are most suitable for this application (*1, 2*) because of their low refrigerating effect per unit volume of refrigerant circulated by the compressor. A knowledge of the boiling heat transfer data for these refrigerants is of primary importance for the accurate design of evaporators in such vapor cycles.

Although several authors have dealt with the problem of evaporation of refrigerants in horizontal tubes (*3* to *6*), and boiling heat transfer data of specific refrigerants has also been published, very little work has been done on the measurement of boiling data for Freon 11 and Freon 113 refrigerants (*7, 8*).

This paper deals with the experimental determination of boiling heat fluxes of Freon 11 and Freon 113 refrigerants in horizontal tubes as functions of several independent variables. The heat transfer data are correlated by using Rohsenow's correlation (*9*) for nucleate boiling. The effects of nucleate boiling and forced convection are separated, and the constant coefficient appearing in the correlation is determined for the various refrigerant and tube material combinations.

Two-phase flow pressure drops are presented as a function of mass flow rate and exit quality. The experimental results are compared to analytical values calculated by a procedure based on Martinelli's method (*10*) for two-phase pressure drop with heat addition.

DESCRIPTION OF THE APPARATUS

A flow diagram of the heat transfer loop is shown in Figure 1. The system consists of three circuits: the refrigerant circuit, the water circuit, and the steam circuit. The refrigerant is circulated by the compressor, flows through an oil separator, an air-cooled condenser, a subcooler, and an expansion valve. After the expansion valve the refrigerant stream is split up into a main stream and a secondary stream. The main stream enters the water-heated test section, passes through the steam-heated superheater and returns to the compressor. The secondary stream passes through the subcooler, a steam-heated heat exchanger, and joins the main stream in the suction line before returning to the compressor. The secondary stream serves two purposes: it facilitates the control of the mass flow of the refrigerant through the test section at a given test-section pressure, and it subcools the refrigerant flowing toward the expansion valve. This enables the maintenance of very low quality at the test section inlet.

The steam is supplied from a high-pressure steam line through a pressure-reducing station and a liquid separator. The role of the steam is to prevent liquid refrigerant from entering the compressor. This is accomplished by superheating the main refrigerant stream in the superheater and the secondary stream in the heat exchanger.

The water circuit is used to heat the test section. The water is circulated by a pump, flows through an electrically heated section, a flowmeter, and the test section.

The water-heated test section is a true counterflow heat exchanger. The refrigerant flows in a 0.25-in. I.D. stainless steel (or copper) tube having a length of 18 in. in the test section. This tube is placed along the center line of a 2½ in. I.D. well-insulated copper tube. The heating water flows between the two tubes in the opposite direction than the refrigerant.

The system was deaerated through an air valve in the condenser, and an oil separator with an estimated separating efficiency of 98%, was used to separate the oil from the refrigerant stream.

INSTRUMENTATION AND CONTROL

The temperature measurements were conducted with 28-gauge copper-constantan thermocouples, and a millivolt potentiometer was used to record the readings. The temperature of the Freon stream was measured upstream from the expansion valve in order to determine the quality of the stream at the test-section inlet. The Freon temperature was also measured at the test-section exit to ensure that the leaving refrigerant was not superheated. The inlet and exit water temperatures were recorded and the water flow rate was adjusted in such a way that the temperature drop of the water in the test section was to be small. The water temperature drop varied between

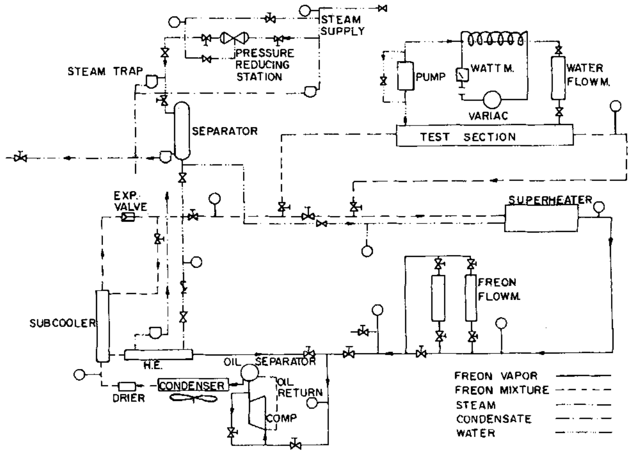

Fig. 1. Flow diagram of the heat transfer loop.

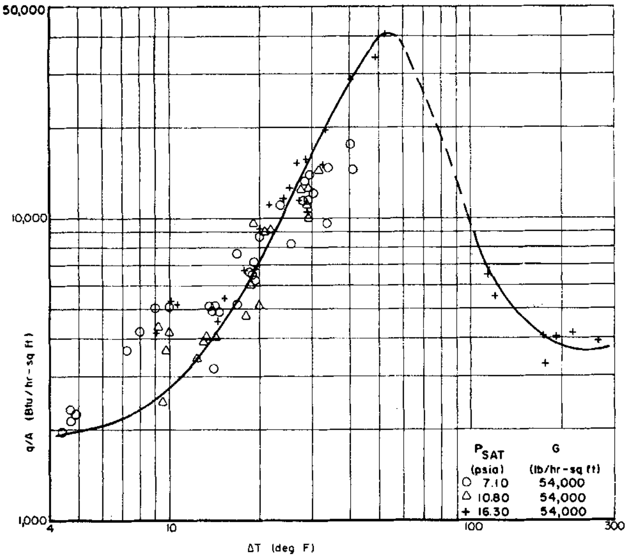

Fig. 2. q/A **vs.** ΔT **at different saturation pressures (Freon 11).**

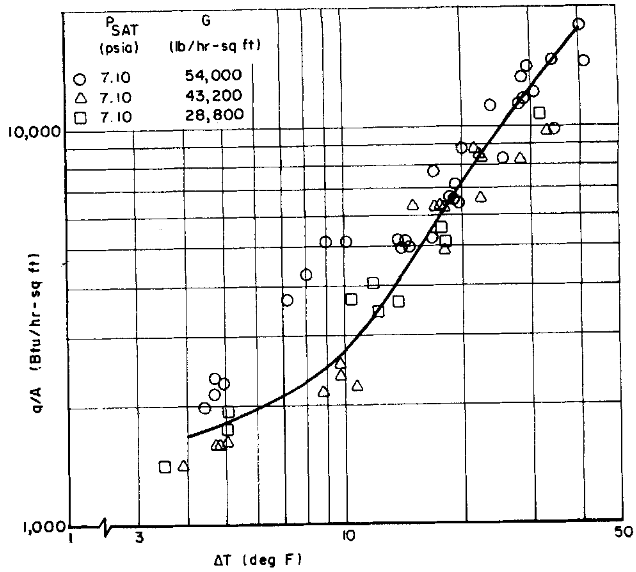

Fig. 4. q/A **vs.** ΔT **at different mass flow rates (Freon 11).**

1 and 4 deg. during the experimental program. Wall temperatures were measured at four different locations along the test tube.

Absolute pressures were determined with pressure gauges, and the test-section pressure drop was measured with a calibrated differential vacuum manometer.

Both the water and refrigerant flow rates were measured by rotameters. The Freon flow rate was measured in the suction line where the refrigerant is superheated vapor.

The heat input to the water was established by the use of an electric heater and was controlled by a variable resistance. The test section was insulated and the heat loss (or gain) of the water was established. Corrections were made during the evaluations of the data whenever it was necessary.

The refrigerant flow rate was controlled by the use of an automatic expansion valve and a bypass line. The steam-control system included an automatic pressure-reducing valve which maintained the required downstream pressure constant independently of upstream and downstream disturbances. The flow rate of the steam was controlled by an automatic valve in order to maintain the required superheat of the refrigerant at the heat exchanger and superheater exits. The water flow rate was controlled by using bypass valves to maintain small water temperature changes in the test section. All instruments were recalibrated several times during the experimental program.

EXPERIMENTAL PROCEDURE

Steady state conditions were assumed when neither the wall temperature nor the water inlet and exit temperatures changed during four consecutive measurements. The average time period between measurements was between 15 and 20 min.

The liquid film superheat, the refrigerant flow rate, the evaporation pressure, and the tube material were the independent variables. (The inlet quality of the refrigerant was kept relatively constant, between zero and 2%, except for the highest flow rates where the inlet quality was between 6 and 8%.)

The heat transfer rate was calculated by the flow rate and enthalpy change of the water (less corrections for heat transfer through the insulation whenever correction was necessary). To check the results, tests were conducted during which the refrigerant was superheated in the test section. In these cases a heat balance could be made by comparing heat transfer rates calculated on the water and refrigerant sides, respectively. The average deviation of the calculated heat balance was within ±6%, while the maximum deviation was 10%. These tests were used for calibration purposes only.

For the calculations the average of the four measured wall temperatures was used; however, the axial variation of the wall temperature was very small. Therefore the difference between local and average heat transfer coefficients is small.

RESULTS

Heat Transfer

Figures 2 and 3 show the heat flux as a function of saturation pressure and wall-to-liquid temperature difference in the nucleate boiling regime for Freon 11 and

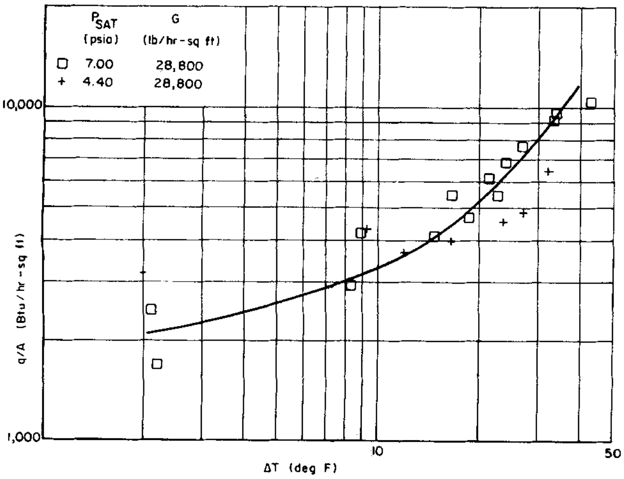

Fig. 3. q/A **vs.** ΔT **at different saturation pressures (Freon 113).**

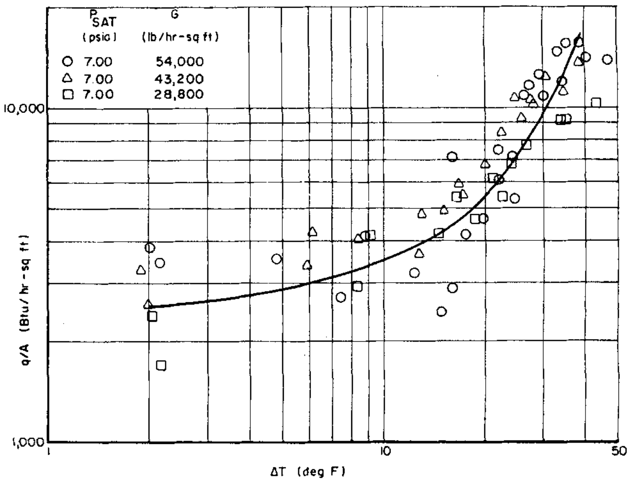

Fig. 5. q/A **vs.** ΔT **at different mass flow rates (Freon 113).**

Freon 113, respectively. Some data obtained for Freon 11 in the film boiling region are also presented. The results show no significant variation of the boiling heat flux with saturation pressure.

Figures 4 and 5 show the heat flux as a function of mass flow rate and wall-to-liquid temperature difference for Freon 11 and Freon 113, respectively. The results show that the variation of mass flow rate does not affect the boiling heat flux in the examined range of flow rate. Based on numerical calculations it is estimated that during the experiments described herein nucleate pool boiling accounted for approximately 94 to 96% of the heat transfer coefficient, while forced convection effects amounted to about 4 to 6%. This explains the relatively small effect of mass flow rate on the boiling heat transfer coefficient.

The effects of the quality on the heat transfer coefficient were examined in detail. No significant effect of the quality or the heat transfer could be established. This observation is in agreement with the results of Anderson (see Figure 10 of reference *11*) for Freon 22, who showed the variation of the local heat transfer coefficient with exit vapor quality for different mass flow rates per unit area and constant quality change. Anderson found that the variation of local heat transfer coefficient with exit vapor fraction diminishes toward lower mass flow rates. At a mass flow rate per unit area of 75,000 lb./hr. sq. ft., there is almost no variation of h with x. Considering that the maximum flow rate examined during this study was 54,000 lb./hr. sq. ft., our findings seem to agree with Anderson's results. (The reason for the relatively low mass flow rate range examined in this program is that both Freon 11 and Freon 113 are low-pressure refrigerants. A small pressure drop results in a relatively large change in saturation temperature in the range of interest for vapor cycle evaporators. Consequently, no large pressure drops are acceptable in such evaporators, and this limits the mass flow rate for this application.)

The experimental results are correlated using Rohsenow's method (*9*) for correlating nucleate pool boiling heat transfer data. It has been shown (*12, 13*) that in the case of nucleate boiling with convection, the forced convection effects can be superimposed on the bubble motion effect, and the total heat flux then is

$$q/A = (q/A)_b + (q/A)_c \tag{1}$$

The forced convection heat transfer coefficient is calculated from McAdams' equation (*14*)

$$N_{Nu} = 0.023\, N_{Re}^{0.8}\, N_{Pr}^{0.4} \tag{2}$$

The combination of Equations (1) and (2) results in $(q/A)_b$, representing that portion of the heat transfer rate which is due to bubble motion effects. The experimental results are correlated by Rohsenow's equation

$$\frac{C_L}{h_{fg}}(T_W - T_{SAT}) = C_{SF}\left[\frac{q/A_b}{\mu_L h_{fg}}\sqrt{\frac{g_o \sigma}{g(\rho_L - \rho_V)}}\right]^{1/3}\left[\frac{C_L \mu_L}{k_L}\right]^{1.7} \tag{3}$$

Using Equations (1) and (2), $(q/A)_b$ was calculated, and the coefficient C_{SF} in Equation (3) was determined for Freon 11 in a stainless steel tube (0.006), Freon 11 in a copper tube (0.010, see Figure 6), Freon 113 in a stainless steel tube (0.004), and Freon 113 in a copper tube (0.007).

Rohsenow's correlation using the experimentally determined value for the coefficient C_{SF} correlates all data in the nucleate boiling regime with an average deviation of approximately 30%.

Attempts were made to correlate the results using Gilmour's correlation (*15*). The experimentally obtained heat transfer coefficients, however, were substantially below those predicted by the correlation.

Finally, Chang's correlation (*16*) was considered. This correlation, however, involves physical constants for which reliable values for the examined refrigerants could not be obtained. Therefore, this correlation could not be used for this program.

Figure 7 shows the Nusselt number plotted against the exit vapor Reynolds number $\left(\frac{Gd}{\mu_v} x_e\right)$ for Freon 11. The results show a slightly increasing trend in Nusselt numbers toward increasing Reynolds numbers. In Figure 7 some data points of reference 7 are also presented for comparison.

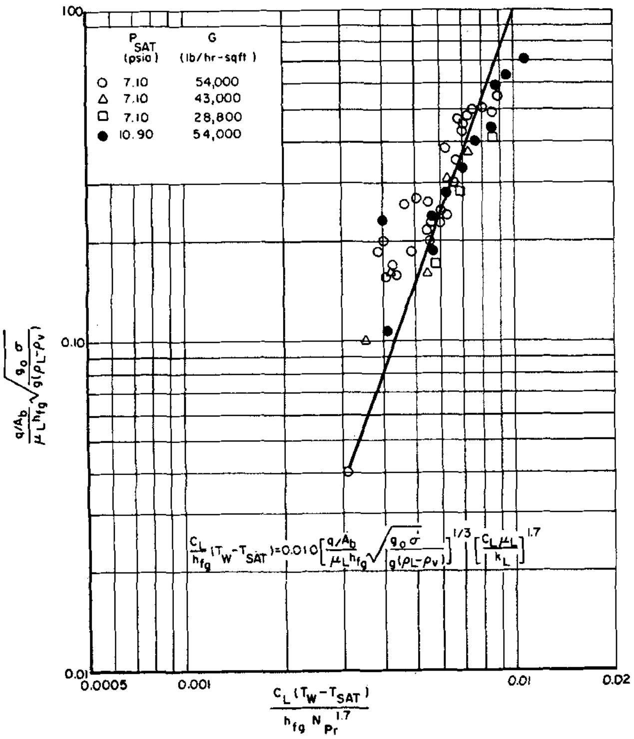

Fig. 6. Correlation of boiling heat transfer data (Freon 11-copper).

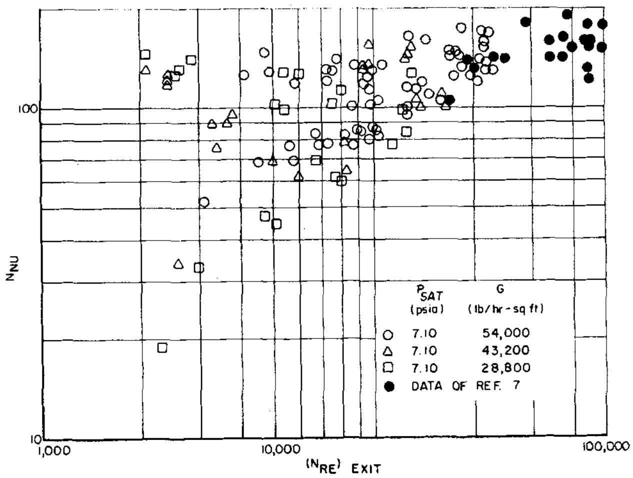

Fig. 7. Nusselt number vs. exit vapor Reynolds number (Freon 11).

Pressure Drop

Two-phase flow pressure drops as functions of mass flow rate and exit quality are presented in Figures 8 and 9 for Freon 11 and Freon 113, respectively. The experimentally obtained points are correlated by a procedure based on

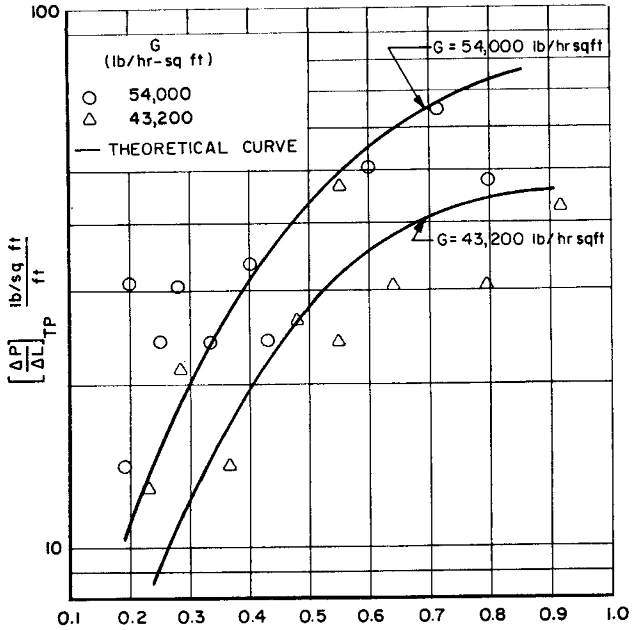

Fig. 8. Two-phase flow pressure drop (Freon 11).

Martinelli's method (*10*) for calculating two-phase pressure drop with heat addition.

The total pressure drop is the sum of the pressure drop due to friction and momentum change, that is

$$\Delta P_{TP} = \Delta P_F + \Delta P_M \tag{4}$$

The frictional pressure drop is calculated from

$$\frac{\Delta P_F}{\Delta P_o} = \frac{1}{x_e} \int_{x_i}^{x_e} (1-x)^{2-n} \phi_{tt} \, dx \tag{5}$$

The exponent $(2 - n)$ was determined from single-phase data, and Martinelli's $\phi_{tt} = f(X_{tt})$ relationship, experimentally obtained for water, was assumed to be correct for fluids in the same Prandtl number range. X_{tt} and ΔP_o were determined by the properties and flow characteristics of the fluids involved

The pressure drop due to momentum change is calculated from

$$\Delta P_M = \frac{G^2}{g_o} v_L \left[(1-x) + x\left(\frac{v_v}{v_L}\right) - 1 \right] \tag{6}$$

It is assumed that there is no slip between the vapor and liquid (complete mixing), and that although the specific volume of the mixture changes, that of each component remains the same (incompressible flow).

The theoretical curves include all data (above $x_e = 0.25$) with an average deviation of 30%. Below $x_e = 0.25$ the correlation underestimates the pressure drop.

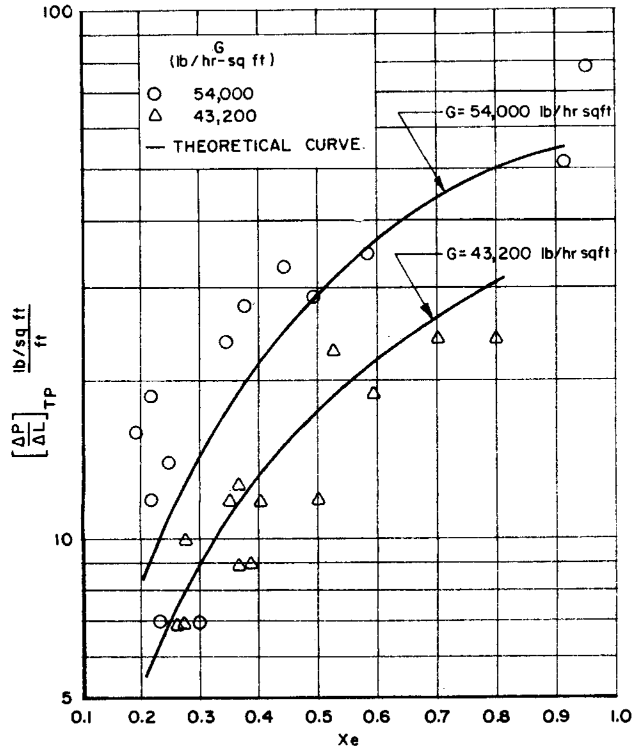

Fig. 9. Two-phase flow pressure drop (Freon 113).

SUMMARY AND CONCLUSIONS

Heat transfer and pressure-drop data were experimentally determined for boiling Freon 11 and Freon 113 refrigerants in horizontal copper and stainless steel tubes. The coefficient appearing in Rohsenow's correlation was determined for each refrigerant-tube material combination and the results were correlated using Rohsenow's method. It is shown in Figure 7 that almost all data obtained for Nusselt numbers during this program fall between 100 and 200. For design purposes these values may be used.

The procedure used to correlate the pressure-drop data includes all data (above $x_e = 0.25$) with an average deviation of 30%.

ACKNOWLEDGMENT

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NOTATION

- A = heat transfer surface, sq.ft.
- C = specific heat, B.t.u./lb. °F.
- C_{SF} = constant in Rohsenow's equation
- d = diameter, ft.
- G = mass flow, lb./hr. sq.ft.
- g = acceleration, ft./sec.^{2}
- g_o = gravitational constant, lb.$_m$ ft./lb.$_f$ sec.^{2}
- h = heat transfer coefficient, B.t.u./hr. sq.ft. °F.
- h_{fg} = latent heat, B.t.u./lb.
- k = conductivity, B.t.u./hr. ft. °F.
- L = test-section length, ft.
- N_{Nu} = Nusselt number
- N_{Re} = Reynolds number
- P = pressure, lb./sq.in.abs.
- N_{Pr} = Prandtl number
- q = heat flow, B.t.u./hr.
- T = temperature, °F.
- ΔT = $T_W - T_{SAT}$, °F.
- v = specific volume, cu.ft./lb.
- x = quality, lb. vapor/lb. total
- X_{tt} = Martinelli parameter
- μ = viscosity, lb./hr. ft.
- ρ = density, lb./cu.ft.
- σ = surface tension, lb./ft.
- ϕ_{tt} = Martinelli parameter

Subscripts

- b = boiling
- c = convective
- e = exit
- F = frictional
- i = inlet
- L = liquid
- M = momentum
- SAT = saturation

TP = two-phase
v = vapor
W = wall
o = refers to all liquid flow

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Mixing on Valve Trays and in Downcomers of a Distillation Column

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Because of the increased use of the valve type of tray in distillation columns, this investigation was initiated. The mixing on each of two consecutive trays of a distillation column was determined. This column contained three Glitsch V-1 ballast trays. The residence-time distribution function and the eddy diffusivity were determined by means of the injection of a salt tracer followed by downstream monitoring. The output responses were correlated by use of an eddy diffusion model.

Considerable effort has been devoted to the theoretical description of mixing in flowing streams. The importance of mixing in distillation columns results from the fact that it is closely related to the plate efficiency.

Danckwerts (*6*) advanced the residence-time-distribution theory for an agitated flow system. This theory is based on the idea of an infinite number of entering streams which are destined to reside in the flow system a certain time so that a complete distribution of fluid-residence times is produced. Taylor (*17*) formulated the theory of turbulent mixing or dispersion by the cumulative effect of the action of many small eddy fluctuations. Danckwerts (*6*), Taylor (*17*), and Tichacek (*18*) developed the diffusion model for the analysis of the dispersion of a nonreacting tracer in an agitated flow process. The diffusion model is frequently approximated by the cascaded stirred-cell model. The overall dispersion in such a model is described by the choice of the proper number of perfect mixers. The similarity of the diffusion model and the cascaded stirred-cell model has been shown by Kramers and Alberda (*12*) and by Aris and Amundson (*2*). Variations of this model with recirculation and bypass streams have been discussed by Levenspiel (*14*).

In order to determine the residence-time-distribution function, dynamic methods of testing are required. Kramers and Alberda (*12*) and many other investigators have employed step function and frequency response methods. For certain boundary conditions, Levenspiel and Smith (*15*), van der Laan (*13*), Aris (*1*), and Bischoff (*4*) describe the use of the means and variances of the impulse response for the analysis of experimental results.

Recently several investigators have studied liquid mixing on distillation trays. Kirschbaum (*11*) described mixing on distillation trays by use of the cell model. Johnson and Marangozis (*10*) employed a splashing factor and the cell model in the description of the results for a single perforated plate. In the analysis of the results for bubble-cap trays, Oliver and Watson (*16*) made use of the concept of recirculation without mass transfer. Gilbert (*9*) utilized the diffusion model to interpret the frequency response results for both bubble-cap and perforated trays. The diffusion model was employed by Gerster et al. (*8*) in the analysis of the results for bubble-cap trays. Foss et al. (*7*) employed step functions in the investigation of sieve trays. Baker and Self (*3*) investigated longitudinal diffusion on a large sieve plate.

EQUIPMENT AND OPERATING PROCEDURE

The distillation column (Figure 1) employed in this investigation contained three plates that were 27 in. in diameter and were spaced 18 in. apart. A drawing of one of the plates is presented in Figure 2.

Air was blown into the column through a 5-in. line by a centrifugal blower that was capable of delivering 1,200 cu. ft. of air/min. (at standard conditions) against a head of 27 in. of water. Water was pumped into the top of the column through a 2-in. line by two centrifugal pumps, each of which was capable of delivering 180 gal./min. of water under the conditions of the experiment. Both the air and water lines were fitted with the necessary orifices and manometers required for making flow measurements. Water was recirculated though two storage tanks, each of which had a capacity of 180 gal.

External conductivity cells were attached to the downcomers of the first and second plates. In the first twenty-nine runs, the bare conductivity probes extended to the downcomer. In runs 30 through 69, the probes were located about 2 in. externally from the downcomer. In both cases liquid was withdrawn through the cells. The conductivity probes in these cells were connected to a conductivity monitor which was essentially a full wave rectifier. The conductivity monitor was connected to a d.c. source circuit in reverse bias in order to suppress the background signal for effective utilization of the full chart width. A brush recorder with two channels was used to record the conductivity signals. The data obtained are given in reference 20 and elsewhere.*

* Tabular material has been deposited as document 7919 with the American Documentation Institute, Photoduplication Service, Library of Congress, Washington 25, D. C., and may be obtained for $1.25 for photoprints or 35-mm. microfilm.