and slug material disperse as the slug progresses from injection well to production well. This then will tell whether or not miscibility is maintained. Figure 18*b* shows schematically the relative positions of fluids in this process in a porous medium.

In one group of experiments the authors determined the composition of the effluent stream from the displacement of naphtha from a 3-1/3 ft. long laboratory porous system by different size propane slugs which were in turn displaced by a lean gas. These composition data are plotted on the triangular diagram of Figure 18*a* to show the relationship between the composition profile through the dispersed slug and the phase boundary curve. For an initial slug size of 37.5% of the core pore volume and for the dispersion characteristic of this system and porous medium the composition profile cuts the phase boundary curve. This indicates that this slug size was not large enough to give a miscible displacement throughout the core. In a like manner two-phase production occurred for the 45% slug-size experiment. For a 55% slug size the entire effluent was single phase with a concentration profile estimated as shown. Since this profile just touches the phase boundary curve, any lower slug size would not yield a miscible displacement throughout the core. This then is the minimum slug size required for a miscible displacement for this length of sandstone core. The peak propane concentration for this slug size was 56%. Since the peak propane concentration in the slug initially was 100% (pure propane was injected), this shows that the peak concentration declined to 56% while the propane traveled through the core. This was the result of the dilution by the lean gas and naphtha.

The results of a series of experiments show that the initial slug length required for a miscible displacement throughout a porous medium is proportional to the square root of the length of the medium (*1*). This means that whereas a 1.83 ft. (55% of the core length) slug of propane was required in the 3.33 ft. core, a slug only 31.7 ft. (3.2% of the total length) would be required for 1000-ft. porous medium, which is a typical reservoir path length. That relatively small slugs are needed in reservoir applications of the miscible slug process makes this process economically promising.

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Manuscript received December 23, 1959; revision received June 13, 1960; paper accepted June 15, 1960. Paper presented at A.I.Ch.E. San Francisco meeting.

The Effect of Mixer Design on the Efficiency of a Pump-Mix Mixer-Settler

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The mixing efficiency of simple and pitched paddles, marine propellers, and centrifugal and disk impellers was determined by a heat transfer method in a single stage of a large pump-mix mixer-settler. The mixing devices were from 3 to 5 in. in diameter and were tested with a refined kerosene as the organic phase and water as the aqueous phase.

A heat conductance was calculated from flow parameters and the efficiency and was correlated against impeller design and speed variables .For centrifugal and disk impellers *UA* was proportional to $n^{1.2}L^3b^{0.3}$; for the paddles *UA* was proportional to $n^{1.67}L^{2.5}b^{0.42}$. At normal operating speeds controlled variation of the internal recirculation of mixed phases through a centrifugal impeller did not change the efficiency.

Horizontal, box type of mixer-settlers are becoming increasingly popular in solvent extraction because of their high efficiency and versatility. A particularly versatile box unit is the pump-mix mixer-settler, described by Coplan and Zebroski (*1*). Excellent operating characteristics have been described in the literature for four pump-mix units, whose sizes are as follows: a miniature unit with a holdup of 8 cc./stage (2), a laboratory-scale unit with a holdup of 0.5 gal./stage (3, 4), and two pilot-scale units with holdups of 2.0 and 6 gal./stage (5, 6).

The subject of the present work is a single stage of a pump-mix mixer-settler with a holdup of 20 gal. The objective of the work was to compare the relative efficiencies of a number of different types of rotating mixing elements in the mixer-settler at the same tip speeds.

Experiments of a similar nature are described in a British paper by Roberts and Bell (7).

DESCRIPTION OF THE EQUIPMENT AND TEST METHOD

The mixing devices were tested in a single stage of a pump-mix mixer-settler. The stage was a long rectangular box, 15 in. high and 8 in. wide. The box was divided into two parts, a mixing section 8 in. long and a settling section 48 in. long. A set of vertical louvers separated the mixing and settling sections. As shown in Figure 1 four smaller boxes were attached to the sides of the stage at the ends to simulate the flow functions of adjacent stages in a multistage mixer-settler. The entire assembly was made of transparent Plexiglas and held about 20 gal of liquid when filled to a height of 10 in.

The mixing section was covered with an 8 by 8-in. plate with a 6-in. hole in it. The plate fit snugly in the top of the mixing section at an elevation of 10 in. A 6-in. pipe, which was welded to the top of the plate, served as the support for the motor for the impeller. As shown in Figure 2 the mixing section was subdivided into mixing and premixing sections by a horizontal plate which had a 3-in. diameter hole in its center. The horizontal plate was located about 5 in. above the bottom of the stage. Although the mixing devices discharged liquid in a horizontal plane, much of the liquid recycled to the premixing section through the 3-in. hole even when simple paddles were used.

Except for the light phase entrance, which was a duct that crossed the top of the mixing section and entered the side of the 6-in. pipe, liquid flowed into the various chambers through 1 by 7-in. slots in the side walls. As shown in Figure 1 the heavy phase slot into the mixing section was located about 4 in. above the bottom of the stage. Light phase flowed from the feed chamber over the top and into the mixing section around the impeller shaft. The heavy phase left the settler through a slot at the bottom of the stage and flowed over an adjustable weir. The light phase passed through a similar slot at an elevation of 10 in.

In operation the entire assembly was filled to a height of 10 in. with equal amounts of the two phases. When the impeller was rotated at pumping speed, the two phases were drawn into the mixing section and discharged as a mixture through the louvers into the settling section. The phases separated, passed into the appropriate outlet chambers, and were pumped through rotameters back to the inlet chambers by two standard centrifugal pumps.

The flow pattern in the stage as seen from the side is shown in Figure 2.

The heavy phase used in the tests was tap water. The light phase was a specially refined kerosene which had a density of 0.78 g./cc., a viscosity at 25°C. of 15.7 mpoises., an interfacial tension with water of 45 dynes/cm., and a specific heat of 0.48 cal./(g.)(°C.) at 25°C.

The mixing efficiency of the various impellers was determined by a heat transfer technique described by Mottel (8). The difference in temperature of the two phases after mixing compared with the temperature difference before mixing was used as a measure of how efficiently the two phases had been contacted. As shown in Figure 3 the kerosene phase was heated about 0.5°C. above room temperature before it entered the mixing section, and the water phase was cooled as it left the settler. The difference in temperature of the two phases was measured by thermopiles in the slots where the liquids entered the mixing section and left the settler. Each thermopile consisted of ten iron-constantan thermocouples sealed in glass tubes. The differential readings between the phases at the inlet and outlet of the stage were recorded on a multirange potentiometer. A run was continued at constant impeller speed, flows, and heat input until the temperature differences became constant.

DESCRIPTION OF THE MIXING DEVICES AND TEST CONDITIONS

Mixing devices of two basic types were tested: open impellers, which include simple and pitched paddles, marine propellers, and a turbine impeller; and closed impellers, which include centrifugal and disk impellers. Sketches of each type of mixing device are given in Figures 4 and 5.

Table 1 lists the conditions under which the efficiency of the various mixing devices was studied. The range of variables was necessarily restricted to flows and mixing speeds which did not exceed the phase-separating capacity of the settling section.

PROCEDURE FOR PLOTTING THE MIXING EFFICIENCY

The thermal efficiency of a system in which two liquids initially at different temperatures are mixed was derived by Mottel (8) in terms of a heat conductance and a heat load. The result is as follows:

$$\ln(1-\eta) = -UA\left[\frac{1}{W_{org}} + \frac{1}{W_{aq}}\right]$$

The equation represents a heat balance around a mixer into which two liquids at different temperatures have been fed, with cocurrent flow assumed. It is independent of the type of mixing element, the shape of the mixing section, and the speed of the mixing element. The input flows of both phases are represented in the equation, although internal recirculation in the mixing section and the slight change of conditions taking place in the settling section are not. The heat conductance can be affected by the mixer configuration, speed, and flows and may be affected by internal recirculation.

In analyzing the data the other variables are held constant, and the effect of mixing speed on the efficiency is obtained by plotting $\ln(1-\eta)$ against speed. (For convenience an efficiency scale is shown and the log scale is inverted.)

RESULTS

Mixing Speed and Flow Variables

Simple Paddles. The effect of the mixing speed of five paddles on the thermal efficiency is shown in Figure 6 for a constant flow rate. For the same speed the paddle with the largest facial area gave the highest efficiency,

TABLE 1. CONDITIONS TESTED

Mixer type	Diameter, in.	Height, in.	Design	Speed, rev./min.	Total flow, gal./min.	Flow ratio, A/O
OPEN PADDLES*						
Simple paddle	2⅞,5	1,2	Solid and perforated blades	80 to 250	4 to 17.5	0.3 to 1
Pitched paddle	4	2 (30-deg. pitch)	Two opposite-thrust paddles, 1 in. apart	100 to 300	13.4	0.45
Marine propellers	5	1	Two opposite-thrust propellers, back to back	150 to 450	4 to 16	0.45 to 1
Turbine	5	1½	(see Figure 3)	170 to 260	13.4	0.45
CLOSED IMPELLERS						
Centrifugal impeller	5	½,1	Single suction, with and without blades or caps	130 to 400	4 to 16	0.25 to 4
		1	Double suction			
Disk impeller	3,5	1,1¼,1¾	Single suction	150 to 800	13.4	0.45

* The marine propellers have three blades; all other mixing devices have four.

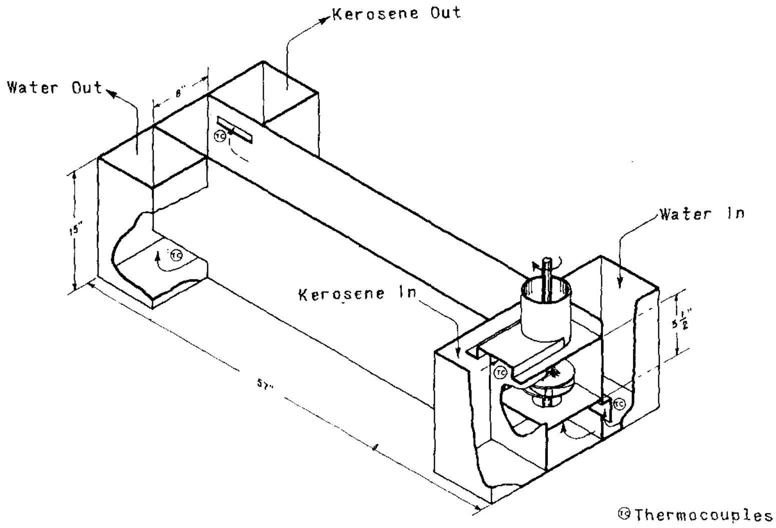


Fig. 1. Isometric view of the mixer-settler stage.

as was expected. The efficiency was lower for smaller paddles, for paddles with perforated vanes, and for the turbine paddle. About the same amount of dispersion was observed in the settling section at the same tip velocities. Typical dispersion thicknesses were: 1½ in. at 250 rev./min., 1 in. at 200 rev./min., ½ in. at 150 rev./min., and less than ½ in. below 150 rev./min.; all readings were ± ¼ in. The 2.9-in. diameter paddle gave ½ in. dispersion at 250 rev./min.

Pitched Paddles and Marine Propellers. The two pitched paddles with opposite thrust were noticeably more efficient than the two marine propellers at the same tip velocity. In both cases the efficiency increased with the speed, as shown in Figure 7. Two propellers spaced 1 in. apart on the same shaft were not as efficient as two propellers touching. The combination of two propellers with a paddle in between was only slightly more efficient than the paddle alone. Four propellers on one shaft were no more efficient than two. Dispersion thicknesses at 13.4 gal./min. were as high as 3½ in. at 450 rev./min., less than ½ in. at 150 rev./min. as shown in Figure 7.

Centrifugal and Disk Impellers. The effect of speed on the efficiency of a number of centrifugal and disk impellers is shown in Figure 8 in comparison with the open impellers. In general the closed impellers were less efficient than the open impellers at the same tip speed. The larger disk impeller was more efficient than the centrifugal impeller of the same diameter at the same flow and mixing speed, as was expected because of its greater axial length. The efficiency of the centrifugal impeller decreased when an orifice cap was attached to the inlet of the suction tube. There was an increase in efficiency when mixing blades were placed on top of the centrifugal impeller. The efficiency of a double-suction centrifugal impeller was higher than the single-suction impeller of the some diameter and width presumably because the two phases were forced to mix inside the impeller. Dispersion thicknesses varied between ½ and 1 in. at 200 rev./min., and between 3 and 3½ in. at 430 rev./min. At the same tip speed slightly more dispersion was produced by the centrifugal impeller with mixing blades than by one without mixing blades but with an orifice cap.

Effect of Recirculation—Centrifugal Impeller

The design of the mixing section of the pump-mix mixer-settler is such that part or all of the mixed phases will be pumped through the impeller more than one time, depending on the speed of the impeller. The hydraulic operation of a multistage unit depends on this recirculation feature because each impeller fixes the position of the interface in the preceding stage in the aqueous flow path and the recirculation makes the interface height less sensitive to impeller speed. In effect the impeller and recirculation system is a pump with a large bypass line from its discharge to its suction.

The amount recirculated was varied by changing the clearance between a disk on the outside of the suction tube and the horizontal plate which separates suction from discharge. An appropriately sized orifice cap was attached to the tip of the suction tube so that constant net flows could be maintained. With reference to Figure 2 the recirculation ratio was determined by sampling the phases at the recirculation plate (1) and at the tip of the suction tube (2). For example the volume fraction of water in one run was 0.7 at point (1) and 0.9 at point (2). The volume fraction of kerosene of course was 0.3 and 0.1, respectively. The kerosene entering the impeller snout (2) could come only from point (1). A kerosene balance then gives $V_2 = 3V_1$. The water entering the snout comes from point (1) and the new water (V_o) entering the premixing section. A water balance gives $V_o + 0.7V_1 = 0.9V_2$. Since V_o is the known water feed rate, the two equations give $V_1 = 1/2V_o$ and $V_2 = 3/2V_o$. Two sets of runs were made, one at 250 rev./min. and the other at 350 rev./min. The net flows for all the runs was 9.2 gal./min. of kerosene and 4.2 gal./min. of water. The recirculation ratio for the impeller without a disk or cap was about 0.6.

The results, plotted in Figure 9, show that the mixing efficiency was not affected by recirculation at 350 rev./min. when the amount recirculated was increased from near 0 to 70% of the net aqueous phase pumped. At 250 rev./min. the efficiency increased rapidly as the recirculation was increased from 0 to 25%, but then it remained constant up to the maximum recirculation. It was noted that the mixture being recirculated was essentially all aqueous phase at the low mixing speed and was approaching the feed ratio at the high speed. (The total pumping rate of the centrifugal impeller is many times the net water flow rate.)

The turbulence at the tip of the impeller was apparently sufficient to mix the organic phase with the aqueous phase at the higher speed. At 250 rev./min. however it is concluded that there was insufficient turbulence and

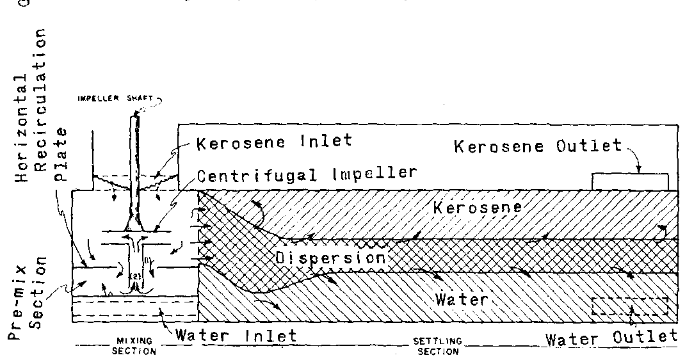


Fig. 2. Flow pattern in the mixer settler.

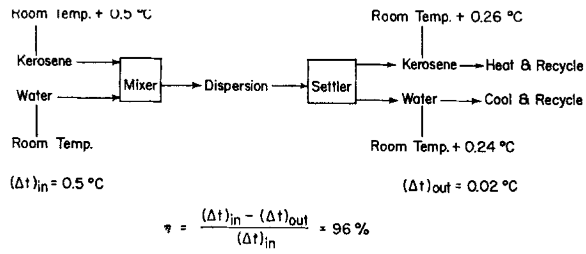


Fig. 3. Operational diagram.

that high recirculation was needed to draw the organic phase down to the intake of the impeller to help it mix with the aqueous phase.

At the normal operating speed of 350 rev./min. therefore the efficiency was not affected by recirculation. The value of recirculation seems to lie in improving the hydraulic operation of the mixer-settler. High recirculation rates did not increase the amount of dispersion in the settling section although it was thought that recirculating mixed phases into the turbulent zone might make finer dispersions.

Correlation of Mixing Efficiency

Effect of Mixer Variables at Constant Flow. In accordance with the parallel flow equation the mixing efficiency and flow can be represented by UA. The most effective impeller is the one that produces the highest UA; therefore the effect of impeller design and speed on UA should be determined. This was done, first at constant flow, for some of the mixers discussed in preceding sections.

The values of UA for centrifugal and disk type of impellers were correlated against the term $\pi n L^{2.5} b^{0.25}$.

The values of UA for several of the simple open paddles were correlated against the term $\pi n L^{1.5} b^{0.25}$ in a similar manner. When screens were placed around the paddle to coalesce the dispersion, or when holes were placed in the blades to reduce the dispersion, higher tip speeds were required to get the same value of UA.

Effect of Flow on the Correlation. The heat conductance was plotted against the total flow at constant speed and flow ratio for three different impellers, as shown in Figure 10. For the closed (centrifugal) impeller the UA increased in proportion to the cube root of the flow. For the open impellers (paddles and propellers) UA was directly proportional to the flow. The increase of UA with flow at constant speed is most likely due to the increase in interfacial area which was evident during the runs from a proportional increase in the amount of dispersion.

The above flow effects were incorporated into the correlations as shown in Figures 11 and 12. The following equations represent the straight-line portions of the curves:

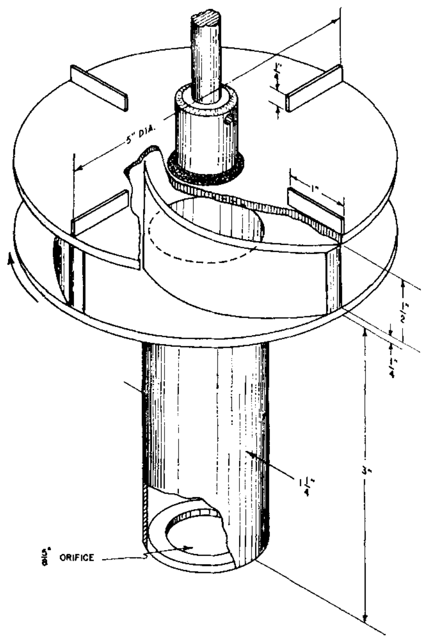


Fig. 5. Isometric view of centrifugal impeller with mixing blades.

Closed impellers:

$$\frac{UA}{(\Sigma W)^{1/3}} = 8.6(\pi n L^{2.5} b^{0.25})^{1.2}$$

Open impellers:

$$\frac{UA}{\Sigma W} = 0.5(\pi n L^{1.5} b^{0.25})^{1.67}$$

As correlating techniques the effective diameter of the turbine paddle was assumed to be 80% of the actual diameter because the vanes did not extend from the axis of the paddle. The effective diameter of the propellers was assumed to be 64% of the actual diameter, an 80% reduction because of the rounded blades and another 80% because the propellers had only three blades whereas all the other impellers had four.

DISCUSSION

In analyzing the data the inefficiency (or UA) was plotted against

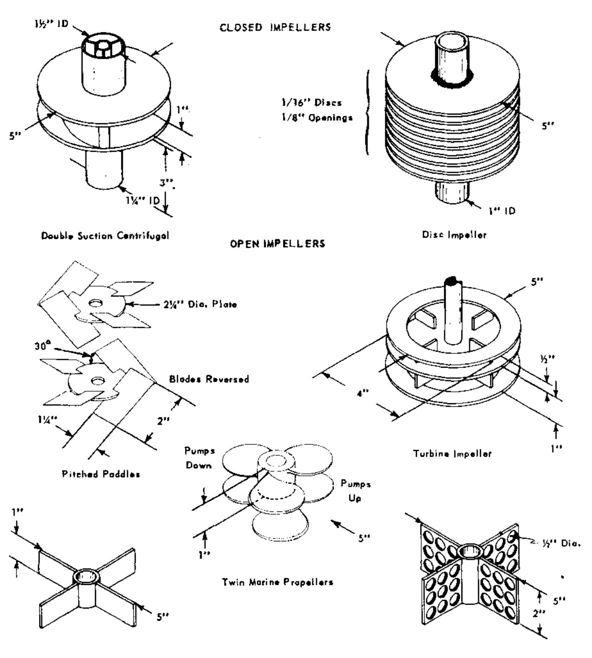


Fig. 4. Types of impellers tested in the mixer settler.

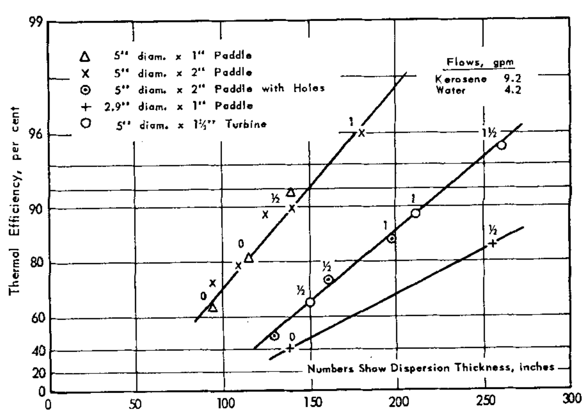


Fig. 6. Effect of mixing speed on the efficiency of open impellers at 13.4 gal./min.

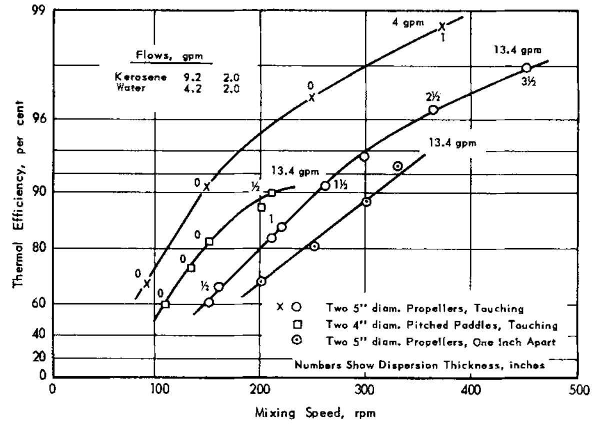


Fig. 7. Effect of mixing speed on the efficiency of propellers and pitched paddles.

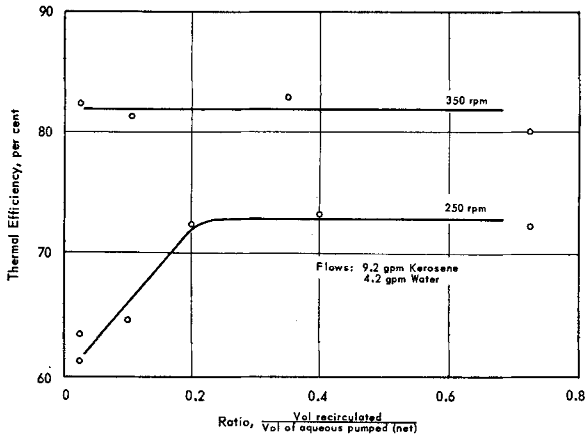


Fig. 9. Effect of recirculation on mixing efficiency for the centrifugal impeller, no blades, $b = \frac{1}{2}$ in.

mixing speed. A more conventional approach might have been to determine the power input to the various types of impellers. In fact it is fairly certain that the differences in efficiency that appeared for the various designs at the same tip speed would have shown proportionately different power inputs. However the power input was not measured in this study for a number of reasons.

In the first place for this size of mixer-settler the motors are selected on the basis of bearing life and mechanical dependability and have many times the horsepower required for mixing the liquids. For example it was determined in other studies that the maximum power required in this contactor for mixing at the rated flows is 1/30 hp. For the reasons mentioned above ½-hp. motors are used, an order of magnitude larger than would normally be used.

The second (and perhaps overriding) factor is that one is interested in rapid phase separation, not just excellent contact of phases. One must be wary of producing so much dispersion or entrainment that harmful back mixing would result in a lowered effectiveness of the contactor. Therefore one is interested in the type of impeller that produces the highest UA and the least amount of dispersion. As noted in the results all the impellers produced about the same volume of dispersion at the same tip speed and flow conditions. Since there is a definite spread in UA for the various impellers at the same tip speed, it was considered useful to show these differences without employing power input.

The plots of the efficiency data are based on the parallel flow equation, where

$$\ln(1-\eta) = -UA\left[\frac{1}{W_{org}} + \frac{1}{W_{aq}}\right]$$

This mathematical model fits the mixer very well at high throughputs, where it is reasonable to expect parallel flow to be taking place. At low throughputs, and particularly since the aqueous phase is generally continuous and recirculates to the premixing section, a different model probably exists in which the temperature of the continuous phase is near or at the exit temperature. This recirculation model is given by the relation below, where the aqueous phase is at the exit temperature:

$$\ln\left[\frac{(1-\eta)(W_{org}+W_{aq})}{W_{aq}+W_{org}(1-\eta)}\right] = -UA\left[\frac{1}{W_{org}}\right]$$

The UA calculated according to the second model will be higher than the first because the driving force is less. The biggest difference between the two appears at high efficiencies and at low aqueous to organic flow ratios. For example the following is obtained for the water and kerosene system:

Flow ratio, A/O	$[UA]_{Recirc.}/[UA]_{Parallel}$ $\eta=0.98$	$\eta=0.90$	$\eta=0.75$
4	1.07	1.05	1.04
1	1.26	1.18	1.12
0.40	1.60	1.38	1.25
0.25	2.00	1.64	1.40

Most of the low throughput runs were made at flow ratios of 1:1. A few were as low as 0.4:1, so that a 60% error could exist in the UA calculated ac-

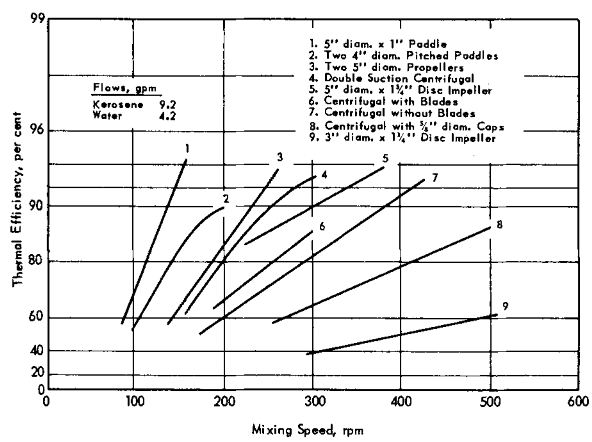


Fig. 8. Summary of the efficiencies of various mixers at 13.4 gal./min.

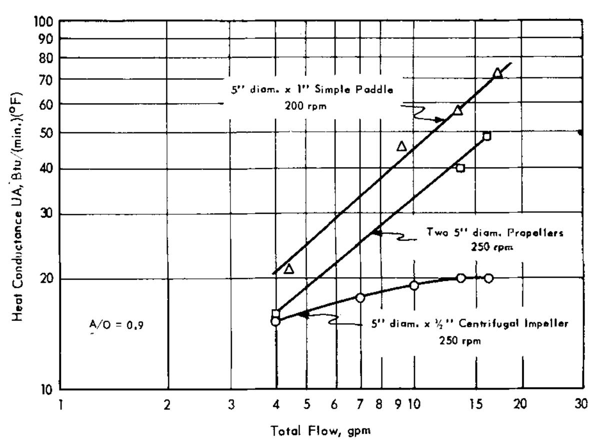
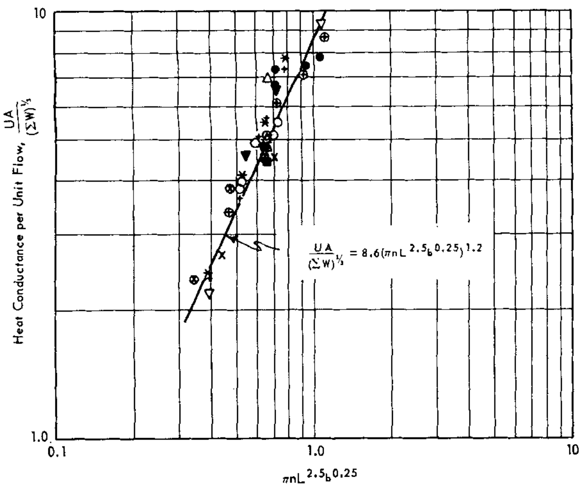
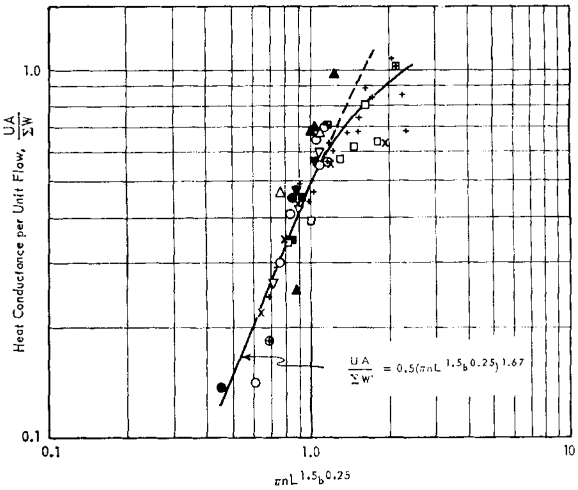


Fig. 10. Effect of total flow on the heat conductance.



Type Mixer	Dimensions, ft Dia.	Width	Flow, gpm Kerosene	Water	Symbol
Centrifugal	0.416	0.0416	2.0	2.0	+
			5.0	2.0	✱
			2.0	5.0	□
			5.0	5.0	△
			8.0	2.0	▽
			2.0	8.0	■
			9.2	4.2	○
			8.0	5.0	⊞
			5.0	8.0	▲
			8.0	8.0	⊟
		0.083	9.2	4.2	⊕
Disc	0.416	0.083	9.2	4.2	●
		0.146	9.2	4.2	▼
	0.25	0.083	9.2	4.2	×
		0.104	9.2	4.2	⊗

Fig. 11. Mixer variables vs. heat conductance per unit flow for centrifugal and disk impellers.



Type Paddle	Dimensions, ft Dia.	Width	Flow, gpm Kerosene	Water	Symbol
Simple	0.240	0.083	9.2	4.2	●
	0.416	0.083	2.3	2.1	△
			6.4	3.0	▲
			4.9	4.5	○
			9.2	4.2	▽
			6.4	5.9	▼
			9.2	8.3	■
	0.416	0.166	9.2	4.2	□
Pitched	0.333	0.166	9.2	4.2	×
Propellers	0.416	0.166	2.0	2.0	⊞
			9.2	4.2	+
			8.0	8.0	⊕

Fig. 12. Mixer variables vs. heat conductance per unit flow for open impellers.

cording to the parallel flow equation when the efficiency was very high.

The effect of throughput on UA at a flow ratio of 0.9:1 was shown for various impellers in Figure 10. For the open impellers UA increases rapidly with flow at constant speed, many times more than the recirculation correction factor could explain. It is postulated that the interfacial area increases rapidly with increasing flow. This is corroborated by the fact that a decided increase in the amount of dispersion is noted as the flow is increased. The effect with the closed impeller is not nearly as well defined, since it is not as efficient a mixer.

CONCLUSIONS

The effectiveness of various impellers was compared at constant tip speed in a single stage of a mixer settler. Rather than use power input a heat conductance was determined from a heat-balance model with parallel flow assumed. Although the UA calculated in this way is somewhat in error at high efficiencies and low flow ratios, it is still a useful and simple parameter to describe the effectiveness of an impeller. The comparison of impellers given in Figure 8 shows that higher efficiencies were produced by open impellers (paddles and propellers) than by closed impellers (centrifugal and disk impellers) at the same tip speeds, at which speeds approximately the same amount of dispersion was produced by all types. The picture will be clearer when more complete data are obtained on the amount of dispersion produced at various values of UA and when a corresponding mass conductance can be determined.

ACKNOWLEDGMENT

The authors wish to express their appreciation to Dr. W. E. Winsche of the Savannah River Laboratory for his guidance and helpful suggestions during this work.

The information contained in this report was developed during the course of work under contract AT(07-2)-1 with the U.S. Atomic Energy Commission.

NOTATION

- A = interfacial area
- b = impeller width, ft.
- c = heat capacity
- L = effective impeller diameter, ft., which is 64% of the actual diameter for the propellers, 80% of the actual diameter for the turbine, and 100% of the actual diameter for all the other impellers
- n = rotational velocity, rev./min.
- U = heat transfer coefficient
- UA = heat conductance, B.t.u./(min.) (°F.)
- V = volumetric flows
- W = $cV\rho$ = mass heat flow, B.t.u./(min.) (°F.)
- Δt = temperature difference
- η = thermal efficiency, fraction, ($1 - \Delta t$ at outlet/Δt at inlet)
- ρ = liquid density
- aq = aqueous phase
- org = organic phase

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Manuscript received April 7, 1959; revision received May 16, 1960; paper accepted May 18, 1960. Paper presented at A.I.Ch.E. Atlantic City meeting.