# helical coil heat transfer in mixing vessels

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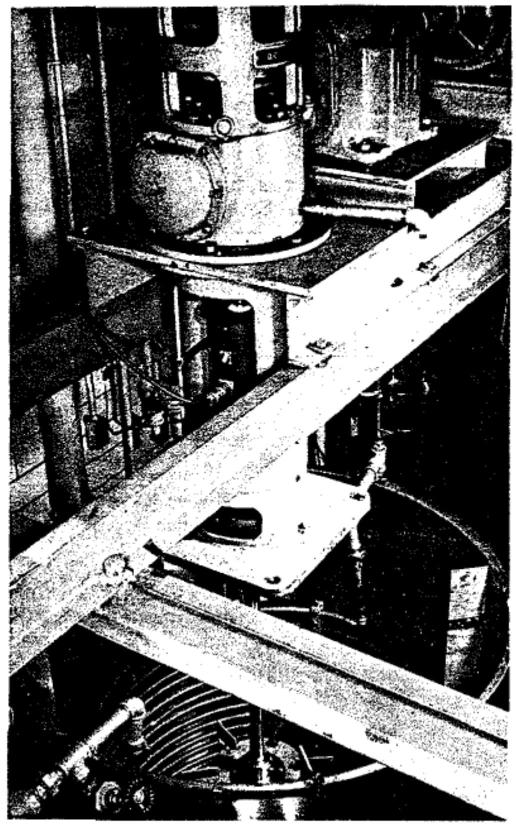


Fig. 8. Photograph of experimental equipment.

**Heat-transfer coefficients for helical coils in a mixing vessel were measured with the use of flat-blade turbine impellers. Data were obtained for baffled conditions. The power consumption of the impeller was measured using a strain-gauge torquemeter. A correlation of heating and cooling data was effected by a graphical method.**

**A Reynolds number range of 400 to 1,500,000 was used. Among the variables investigated were impeller speed, power, ratio of impeller size to tank size, baffle position and tube diameter.**

Heat transfer is a common auxiliary process requirement in mixing applications.

Common types of heat-transfer surfaces are helical coils, jackets, and vertical tubes. Heat-transfer data were recently published (*3*) for a vertical tube system in which the vertical tubes served both as baffles and as heat-transfer surfaces. Previously available data (*1, 2*) on helical-coil installations covered operation in unbaffled tanks. A summary of prior work on heat transfer (*6*) also presents basic flow patterns and the reasons for the use of baffles in a mixing vessel.

### Equipment and Procedure—Summary

Flat-blade turbines, 12, 16, 20, 24 and 28 in. in diam., illustrated in Figure 1, were used throughout this work. Fluid viscosities ranging from 0.4 to 400 centipoises, and Reynolds numbers ranging from *400 to 1,500,000*, were investigated.

The tank was 48 in. in diam., with a 48-in. liquid level. Coils were arranged for both steady- and unsteady-state operation. The copper tubes used were ⅞ in. O.D. and the stainless-steel tubes also used were 1¾-in. O.D. Figure 2 shows the arrangement of the equipment.

Thermocouples were imbedded in the tube wall; thus an experimental temperature difference across the fluid film was given.

### Experimental Results

GENERAL CORRELATION OF $h_o$

The general correlation of $h_o$ with other variables is given as

$$\frac{h_o d}{k} = 0.17\left(\frac{ND^2\rho}{\mu}\right)^{0.67}\left(\frac{C_p\mu}{k}\right)^{0.37}\left(\frac{D}{T}\right)^{0.1}\left(\frac{d}{T}\right)^{0.5} \qquad (1)^*$$

Experimental conditions used to determine Equation (1) are given in the report. The range of application is recommended as:

*Tank size:* All sizes.

Relations have been shown to hold by Rushton and Dunlap (3).

*Baffles:* Either on wall or inside coil.

Tube diameter: $0.018 \leqslant d/T \leqslant 0.036$.

*Viscosity:* Experimental data up to 400 centipoises.

*Data of Uhl (7) indicate that these relations* should hold into the viscous range, 10,000 centipoises.

*Tube spacing:* See report.

*Turbine type:* Flat-blade turbine.

HEATING AND COOLING

A 16-in. diam. flat-blade turbine is placed 16 in. off bottom in a 48-in. diam. tank equipped with helical coils. The coils have a tube diameter of 1.75 in. O.D. and a two-tube diameter spacing. (Data are shown in Table 1). On this table following the run number the letter (*S*) is used for steady state and (*U*) for unsteady state.

The variable of film temperature was studied by obtaining two coefficients, one heating $[h_h]$, and one cooling $[h_c]$. for each condition. The relation used for correlation is given by Equation (2)

$$h_c, h_h = h_o[\mu_s/\mu]^{-m} \qquad (2)$$

* Terms in Equation (1) are given in Notation. and plotted in Figure 3.

For Tables 1, 2, 3, 4, and 6, order document 4411 from A.D.I. Auxiliary Publications Photoduplication Service, Library of Congress, Washington 25, D. C., remitting $1.25 for microfilm or $1.25 for photoprints.

*Manufacturer of LIGHTNIN Mixers and Aerators

The surface temperature of the coil was calculated and the ratio of $\mu_s/\mu$ obtained, where $\mu_s$ is the viscosity at the surface temperature, and $\mu$ is the bulk viscosity. A plot was made for each condition of $h_o$ and $h_h$ vs. $\mu_s/\mu$, on a logarithmic plot, and a straight line drawn in connecting the points. Typical curves are shown on Figure 4. It should be noted on Figure 4 that the slope of the lines ($m$) is not constant under varying conditions. Two possible causes are:

**(1) The value of ($m$) is a function of fluid properties such as $\mu$, $k$ and $Cp$, and**

**(2) The value of ($m$) is dependent upon the value of $\mu_s/\mu$ and thus on the heat flux through the film.**

A plot of ($m$) vs. viscosity (Cause 1), is shown in Figure 5 for fluid viscosities. The stainless steel tubes and the copper tubes give slightly different results. The spread of the viscosity ratio $\mu_s/\mu$ was greater for the small copper tubes, but was not sufficient to enable a correlation with $\mu_s/\mu$ definitely to prove Cause 2.

An average line was drawn through the points on Figure 5 and is of sufficient accuracy for normal tube proportions and viscosities in industrial use.

For all succeeding correlations the variable of average film temperature was eliminated by making a plot of $h_o$ and $h_h$ vs. $\mu_s/\mu$. A straight line was drawn joining the points, picking off the value of $h$ at $\mu_s/\mu = 1.0$. This value of $h$ is termed $h_o$.

### PRANDTL NUMBER

A plot of $h_o d/k$, the Nusselt number, vs. Reynolds number for three different fluid conditions is shown in Figure 6. By cross-plotting these data at constant Reynolds number as a function of Prandtl number, an exponent on the Prandtl number of 0.37 was obtained. This agrees with previous data on heat-transfer coefficients.

### D/T RATIO

The ratio of impeller diameter ($D$) to tank diameter ($T$) was varied between 0.25 and 0.58. Table 2 gives the data used to determine the effect of $D/T$ ratio. In plotting this data at constant Reynolds number, one found the exponent to be 0.1.

### TUBE DIAMETER

Data given in Tables 1 and 2 were all for a tube with an O.D. ($d$) of 1.75 in., and tube spacing [$S_c$] of 3.5 in. Runs made with a tube diameter of 0.875 in., $S_c = 1.75$, gave the data listed in Table 3. Since the tube diameters used are about as large and as small as would be practical, it is assumed that it is sufficiently accurate to use an exponential relationship even though there are only two different tube sizes. Data from Tables 1, 2 and 3 were plotted at constant Reynolds number and gave an exponent of 0.5 on the $d/T$ ratio.

The use of an exponential relation even though only two points are available is justified by an analogy to data on heat transfer to tubes in other heat exchangers.

### TUBE SPACING

The spacing was varied between two diameters and four diameters for the 1.75-in. diam. tube. Table 4 includes additional data for the wide tube spacing. Table 5 shows that a wider tube spacing gives a lower coefficient, and this is more marked in the case of the high viscosity fluid. Table 5 serves as a guide in estimating the effect of tube spacing.

### BAFFLE PLACEMENT

Baffles were placed at the tank wall,

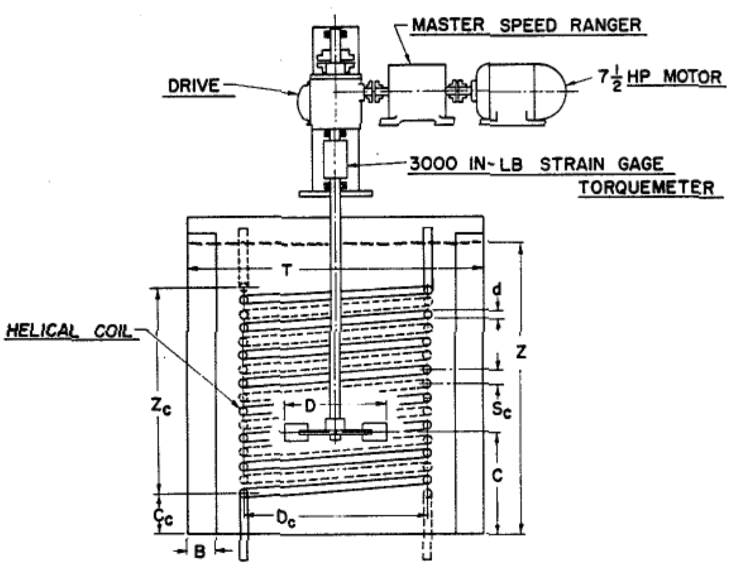
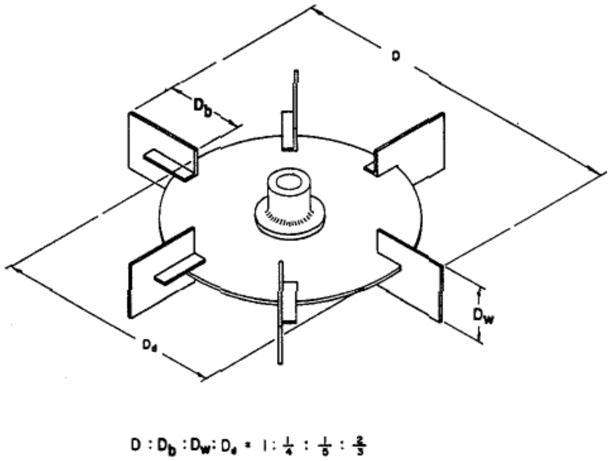



Fig. 2. Experimental equipment.




$D : D_b : D_w : D_4 = 1 : \frac{1}{4} : \frac{1}{5} : \frac{2}{3}$

Fig. 1. Flat blade turbine impeller.

**Table 5.—Effect of Coil Spacing on Heat Transfer**

Tank—48-in. diam., 48-in. liq. level, flat bottom, 365 gal. Four 4-in. wide baffles at the wall.

Impeller—16-in. diam. 6 flat blade turbine, $C = 16$ in.

Coil—1.75-in. O.D.; stainless steel tube; $d_c = 34.25$, $S_c$, variable.

| Fluid | Viscosity | $\frac{[h_o]_{S_C = 4d}}{[h_o]_{S_C = 2d}}$ |
|---|---|---|
| Water | 0.4 centipoises | 0.96 |
| Oil | 50 centipoises | 0.88 |

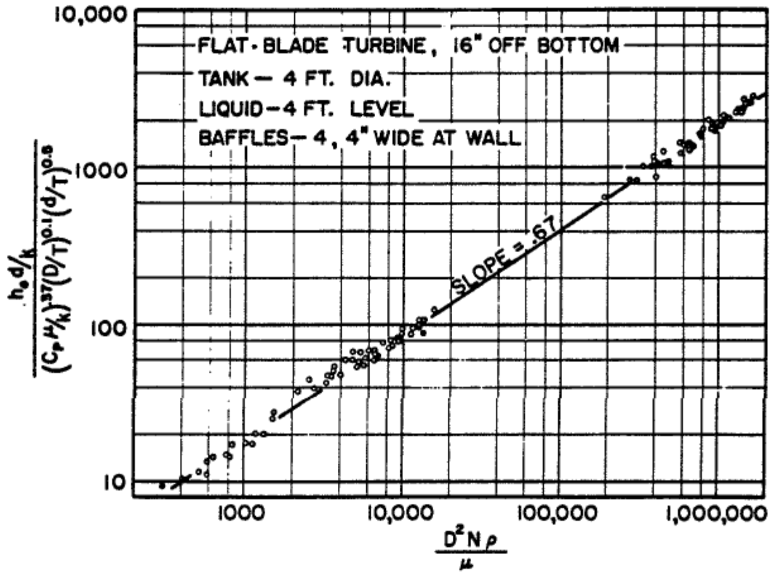



Fig. 3. Overall correlation of $h_o$.

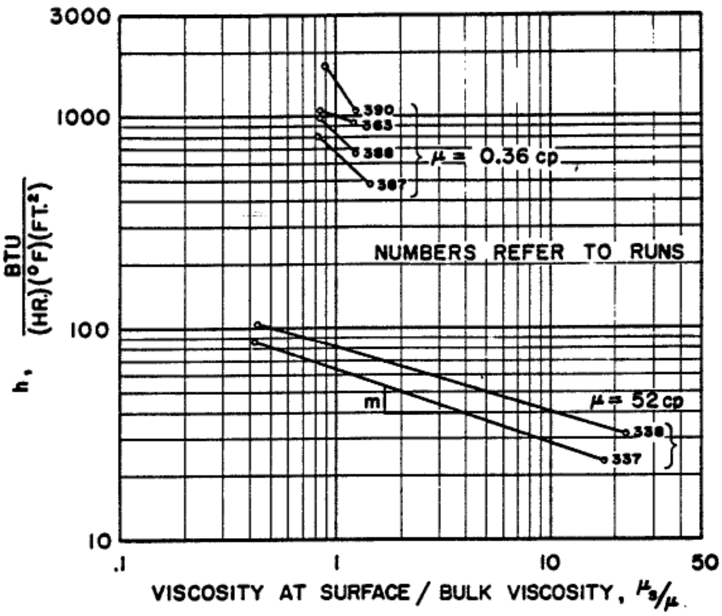



Fig. 4. Evaluation of $h_o$ and $m$.

**Table 7.—Effect of Baffle Placement on Heat Transfer**

Tank—48-in. diam., 48-in. liq. level, flat bottom, 365 gal. Four 4-in. wide baffles at the wall.

Impeller—16-in. diam. 6 flat blade turbine, C = 16 in.

Coil—1.75-in. O.D., stainless steel tube, $D_c$ = 34.25, $S_c$ = 3.5 in.

| Baffle position | Relative $h_o$ |
|---|---|
| At wall | 1.0 |
| 1 in. off wall | .95 |
| Inside coil | .95 |

1 in. off the wall, and inside the coil (data given in Table 6). It was found that wherever baffles were placed—either at the tank wall or inside the coil—little difference resulted in the heat-transfer coefficient (see Table 7). The important point was that baffle placement can be determined by other process or mechanical requirements and can still achieve good heat transfer.

#### Power Measurement

In the consideration of a mixing process from a standpoint of fluid mechanics, it is desirable to have a comparison on the basis of equal energy input to the system. Power (energy per unit time) produces an impeller flow and an impeller head. The effect of the ratio of flow to head can be evaluated best by examining certain of the variables at constant power input. In addition, power is a major item in the cost of the mixer, and comparison at constant power input yields data on the most effective conditions.

Data presented here extend data (*5*) for tanks with helical coils, and various baffle conditions. Data are shown in Table 8 which give the value of the power number, $Pg/\rho N^3 D^5$, where $P$ is power, $g$ is gravitational constant over the range of Reynolds numbers of $10^4$ and higher, which is the turbulent area.

## mixing

Further refinements of data (*5*) have been possible due to use of the strain-gauge torquemeter measuring technique. Also, the baffles used here are one-twelfth the tank diameter in width rather than one-tenth as used by Rushton (*5*).

It is seen from Table 8 that with baffles at the wall, the diameter and the spacing of the tubes have no effect on power consumption at constant speed and diameter. Placing the baffles 1 in. off the tank wall, or inside the coils, lowers the power consumption.

#### COMMENT

Correlations given in Equation (1) are in terms of dimensionless groups, with speed and diameter as the two independent variables to specify mixer

**Table 8.—Effect of Baffles and Coils on Impeller Power Consumption**

Tank—48-in. diam., 48-in. liq. level, flat bottom, 365 gal.

Impeller—16-in. diam., 6 flat blade turbine, C = 16 in.

Coil—1.75-in. O.D., stainless steel tube, $D_c$ = 34.25 in., $S_c$ = 3.5 in.

| Coil | | Baffles | $N_P$ at $N_{Re} > 10^4$ | Ratio |
|---|---|---|---|---|
| Tube diam. | Tube spacing | position | | |
| 0 | 0 | at wall | 5.4 | 1.0 |
| 1.75 | 3.5 | at wall | 5.2 | .96 |
| 1.75 | 7.0 | at wall | 5.2 | .96 |
| 1.75 | 3.5 | 1 in. off wall | 4.8 | .89 |
| 1.75 | 3.5 | inside coil | 4.5 | .83 |
| 1.75 | 3.5 | no baffles | 1.9 [at $N_{Re} = 10^6$] | .35 |
| 0.875 | 1.75 | at wall | 5.2 | .96 |

characteristics. It is desirable, however, to give a useful comparison of process results maintaining equal power input to the system.

#### $h_o$ VS. POWER

Since the heat-transfer coefficient varies with (Reynolds number)$^{0.67}$, $h_o$ varies with [Horsepower]$^{0.22}$ in the turbulent region above a Reynolds number of $10^4$. The exponent is lower than 0.22 in the transition range for Reynolds numbers from 10,000 to 400.

#### TUBE DIAMETER

In comparing the effect of tube diameter, one finds that the actual tube diameter is commonly used in the Nusselt number. This has caused a problem in comparing heat-transfer data from various sources. The fact that a given coil has a large diameter usually gives a larger value of the Nusselt number regardless of whether a higher heat-transfer coefficient has or has not been obtained.

The relationship given in Equation (1) shows that at constant speed, turbine diameter, and power, since variations in tube diameter do not affect power consumption.

$$h \propto 1/(d)^{0.5} \qquad (3)$$

The minimum tube diameter is determined thus by pressure drop through the coil to affect flow of steam or other fluids, or by the minimum spacing between the tubes that can be used without affecting other requirements in the mixing process.

The relationship above is in general agreement with relationships on heat-transfer coefficients in pipes.

#### FLOW-HEAD RATIO

Power input to a mixing impeller produces an impeller flow and an impeller head. The impeller head is used primarily to generate turbulence in the discharge stream from the impeller. In heat transfer, flow from the impeller passing around the heat-transfer surfaces is the primary source of turbulence.

It has been shown (*6*) that a large-diameter impeller running at slow speeds gives more flow and lower head than does a small-diameter, high-speed impeller.

The effect of $D/T$ ratio at a constant power input is shown in Figure 7. This indicates that the larger the diameter of the impeller, the higher the heat-transfer coefficient. Heat transfer is aided by large impeller flows. The slope of the line for water is greater than it is for the oil. This is due to the fact that when oil is used, the Reynolds

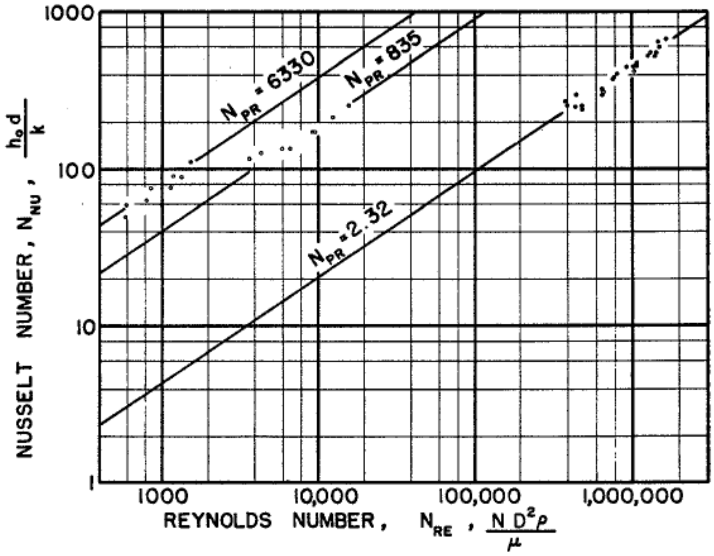



**Fig. 6. Nusselt number as a function of Reynolds number.**

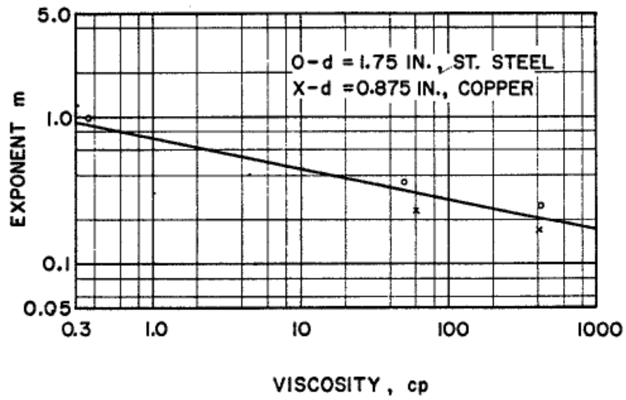



**Fig. 5. Value of exponent [m] as a function of bulk fluid viscosity.**

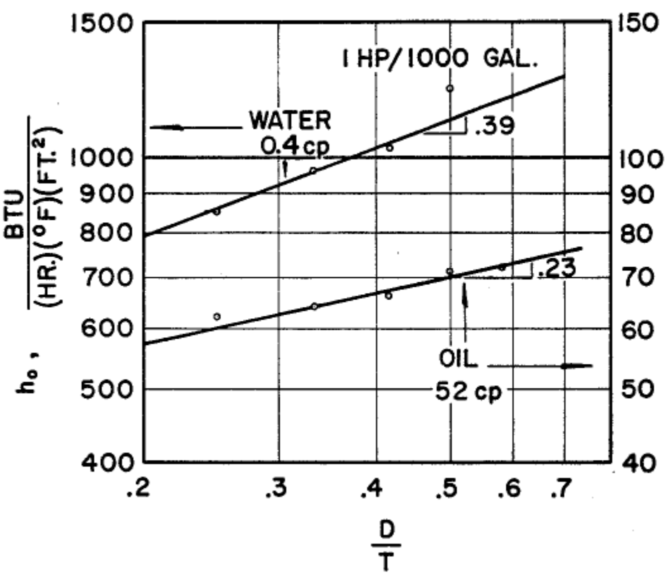



**Fig. 7. Effect of D/T ratio at constant power.**

number is in the transition region where the power number ($N_P$) is not constant with $N_{Re}$.

#### Comparison with Previous Data

Comparing the results of this investigation with the results of other investigators, it is misleading to use the Nusselt-Reynolds number correlation alone. It has been pointed out previously that investigators using a larger tube diameter come out automatically with a higher Nusselt number at the same Reynolds number. Also an investigator using an impeller which consumes more power at a given Reynolds number due to a difference in design would give a higher heat-transfer coefficient, due primarily to the higher power input.

If power data are available, comparison can be made for each case by considering the operation on the basis of heat-transfer coefficient obtained as a function of energy input to the system.

#### VERTICAL TUBES

Certain data (*3*) show that at a Reynolds number equal to $10^6$, $C/T = \frac{1}{2}$ (optimum impeller position), Prandtl number = 2.32 (water), $D/T = \frac{1}{3}$, $B$ (number of baffles) = 6, $d = 1.9$ in., $k = 0.39$ B.t.u./(hr.)(sq.ft.)(° F./ft.)

$$= (700)(2.32)^{0.3}(D/T)^{0.33}(2/B)^{0.2}\left[\frac{0.39 \times 12}{1.9}\right]$$

$h_o = 1230$

For helical coils at Reynolds number equal to $10^6$,

$C/T = \frac{1}{3}$,

Prandtl number = 2.32, $D/T = \frac{1}{3}$, $d = 1.75$, $k = 0.39$

$h_o = 1210$

$h_o = 1160$ at $d = 1.9$ in.

At the same speed and diameter the power consumption of an impeller in a tank with vertical tubes is 75% of that in a tank with standard wall baffles. On the basis of constant power input, the ratio of coefficients in a tank equipped with vertical tubes to the coefficient in a tank equipped with helical coils is given by

$$\frac{h_o(V.T.)}{h_o(H.C.)} = \frac{1230\left(\frac{1}{0.75}\right)^{0.22}}{1160} = 1.13 \qquad (6)$$

In normal installations the parallel flow liquids in vertical tubes gives a lower inside coefficient than for a helical coil, and over-all coefficients are usually quite similar. More area can usually be obtained by helical coils than with vertical tubes. The final choice of vertical tubes or helical coils should normally be made on the basis of economics, ease of installation, replacement, and required heat-transfer surface area.

#### OTHER DATA ON HELICAL COILS

The estimation of power consumption in unbaffled tanks with impellers that are not similar to impellers of known characteristics can not be made accurately. The industrial use of unbaffled tanks is quite limited and an accurate detailed comparison is not feasible or necessary.

#### HELICAL COILS VS. JACKETED TANKS

The major source of data for these two systems is (*1, 2*). When one compares the data on jacketed tanks and helical coils with $d/T$ equal to 0.042, it is found that a jacketed tank at the same Reynolds number and, therefore, at the same power input, has a coefficient about 65% of that obtained with a helical coil. This would vary with the size of tube diameter in the coil. There is some question whether this holds true in baffled vessels. This gives an approximate figure to use, however, in estimating data for jacketed tanks from data available for helical coils.

### Experimental Equipment—Details

#### VESSEL

**The vessel was a 48-in. cylindrical tank with a flat bottom. The total height was 56 in. Liquid depth was 48 in. Four 4-in. baffles were used, and bracket arranged so that they might be placed at the tank wall, 1 in. off the tank wall, or inside the helical coil. Figure 8 shows a photograph of the equipment.**

#### FLUIDS

**Two fluids were used: Rochester (N.Y.) city water passed through an ion exchanger, and Gulf Crown E turbine oil. The physical properties of the oil are given in Table 9 and the properties of water and tube metals were obtained from Perry (4). The viscosity of the oil was checked in the laboratory over the entire series of experiments and did not vary from the data supplied by the Gulf Refining Co.**

**Table 9.—Properties of Gulf Crown E Turbine Oil**

| | ° F. | | |
|---|---|---|---|
| Specific gravity .......... | 100 | | .885 |
| | 130 | | .874 |
| | 210 | | .844 |
| Viscosity centipoises ....... | 100 | 678 | |
| | 130 | 245 | |
| | 210 | 35.5 | |
| Specific heat, B.t.u./(lb.)(°F.) | 100 | | 0.457 |
| | 140 | | 0.494 |
| | 220 | | 0.513 |
| Thermal conductivity, (B.t.u.)/(hr.)(sq.ft.)(° F./ft.) .... | 60 | | 0.90 |
| | 150 | | 0.87 |
| | 212 | | 0.86 |

#### COILS

**To obtain steady-state conditions, combination coils were used having alternate turns for heating and cooling. The coils were wound on the same outside diameter, and each coil wound with a double pitch. The arrangements of these coils are shown in Figure 2.**

**Two different tube diameters were used. The essential dimensions of these coils are:**

| | | |
|---|---|---|
| $d$ = | ⅞ in. | 1¾ in. |
| $D_o$ = | 35.125 | 34.25 |
| $D_e$ = | 36.0 | 36.0 |
| $Z_o$ = | 31.5 | 31.5 |
| $C_o$ = | 7.0 | 7.0 |
| $S_o$ = | 1¾ | 3½, 7 |

For certain data it was convenient to use a single coil, for either heating or cooling, and to use a suitable unsteady-state calculation to obtain equipment performance at the given condition.

To obtain accurate temperature measurements across the fluid film, copper-constantan thermocouples were im-

**mixing**

bedded in the tube wall. A circumferential groove 0.037 in. deep was made at the desired thermocouple location (*3*). The wire thickness was 0.018 in., leaving an average thermocouple junction depth of 0.028 in. The thermocouple wires were joined in a lap joint, ¾ in. long, and soft soldered.

Thermocouple leads with Fiberglas insulation were wrapped around the remaining portion of the groove and taken off the opposite side of the tube. The groove was filled with Glyptol cement, and smoothed to avoid disrupting the flow pattern. The wires were brought over to the tank wall and up through the liquid surface.

In a baffled vessel all parts of the helical coil are not subject to the same direction of flow since the direction of flow varies with height of the coil and proximity to the baffles. To insure a valid average temperature, several thermocouple positions were used in tests. Two basic configurations were used:

**A. Eleven thermocouples on the heating coil, eleven thermocouples on the cooling coil.**

**Five thermocouples were spaced equally in a vertical plane, placed on the inside position on**

the tube circumference. One set was placed on a radius coinciding with a baffle, and another set was placed between two baffles. The set at the baffle had an additional thermocouple on the top turn mounted on the outside position on the tube circumference.

B. Ten thermocouples on one coil ($B_1$), six thermocouples on the other coil ($B_2$).

$B_1$. Five thermocouples were spaced in a vertical plane, one set on a radius coinciding with a baffle, and the other set between two baffles. The ten thermocouples were arranged in a random pattern among outside, inside, top and bottom positions on the tube circumference.

$B_2$. Three thermocouples were spaced equally in a vertical plane, one set on a radius coinciding with a baffle, another set between two baffles. These were arranged in a random pattern among inside, outside, top and bottom positions on the tube circumference.

Thermocouples were connected to a Brown 16-channel multipoint recorder.

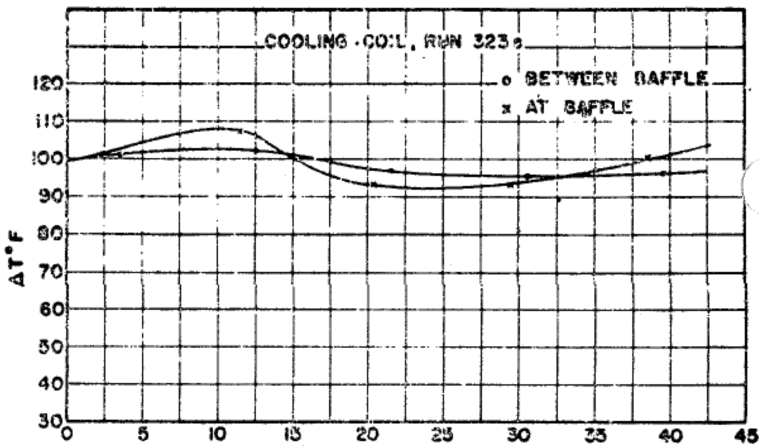



Fig. 9. Graphical integration of mean $\Delta T$.

#### IMPELLERS

Flat-blade turbine impellers with six blades were used, having proportions shown in Figure 1. The diameters used were 12, 16, 20, 24 and 28 in.

#### POWER MEASUREMENTS

A Baldwin electric resistance strain-gauge torqúemeter, 3,000 in.-lb. capacity, measured the torque in the mixer shaft. This gave a continuous record on a Brown single-channel strip-chart recorder. A hand tachometer was used to measure the speed.

The drive consisted of a 7½ hp., 1750 rev./min. motor; a variable-speed Master Speedranger, giving a speed variation of three to one upward and downward; and an E Series drive having gear ratios from 6.3 to 26.0.

#### OTHER MEASUREMENTS

The flow of water through the system was measured with a Fisher and Porter flowrator. The temperature of the water flowing in and out of the cooling coil was measured by mercury in glass thermometers mounted in pipe tees. Steam input was controlled by a pressure regulator, and the amount of condensate weighed on a Toledo scale. The temperature of the fluid in the tank was measured with a thermocouple probe, and with a mercury in glass thermometer.

### Procedure

#### STEADY STATE

The mixer was turned on and set at the desired operating speed. Steam in the heating coil, and cold water in the cooling coil were turned on, and when the tank contents reached an equilibrium temperature, a thirty minute run was made measuring flow rates and temperatures. At least three readings were taken of each measurement.

#### UNSTEADY STATE

In certain runs only one coil was used for either heating or cooling. These runs were started at 10 to 20°F. from the desired tank fluid temperature, and ended at the same temperature difference above or below the desired temperature. Time of measurements was recorded by the use of a stop watch.

For those unsteady state runs in which $S_o = 3½$ in., the second coil remained in the tank but inactive as to heating or cooling. For the runs in which $S_o = 7$ in., this inactive coil was removed and the single remaining coil was used for either heating or cooling.

### Calculation

The lowest power input used, except for natural convection data, was 0.1 hp./1,000 gal. A thermocouple probe indicated that the temperature of the fluid in the vessel was uniform throughout, including the fluid between the tubes of the coil. A constant fluid temperature was used, measured by a mercury in glass thermometer. To obtain natural convection coefficients, the mixer was turned off, and the thermocouple probe was used to get an average temperature reading throughout the entire tank.

**Table 10.—Thermocouple Temperature Data for Cooling Water Coil †**

| Thermocouple position in tank | Thermocouple No. | Thermocouple reading °F. | Tank temp. °F. | ΔT °F. | Length of tubing below liq. surface at thermocouple location ft. |
|---|---|---|---|---|---|
| Between baffles | 12 | 93.8 | 190 | 96.2 | 39.6 |
| | 13 | 94.5 | | 95.5 | 30.6 |
| | 14 | 93.5 | | 96.5 | 21.5 |
| | 15 | 88 | | 102 | 12.5 |
| | 16 | 89 | | 101 | 3.5 |
| | 17 | 89.7 | | 100.3 | 3.5 |
| At baffle | 18 | 90 | | 100 | 38.5 |
| | 19 | 96.7 | | 93.3 | 29.5 |
| | 20 | 96.7 | | 93.3 | 20.4 |
| | 21 | 83 | | 107 | 11.4 |
| | 22 | 88.7 | | 101.3 | 2.4 |

† Run 323.

### STEADY STATE

During the steady-state runs, conditions did not vary with time and Equations (4) and (5) represent the method of calculating the mean heat-transfer coefficient over the length of the tube.

$h$ = mean fluid film heat-transfer coefficient over length of tube
$Q$ = heat transferred
$c$ = circumference of tube
$L$ = length of tube
$\Delta T$ = temperature driving force between thermocouple position and bulk of fluid at a point
$t_m$ = mean temperature at the thermocouple position over the length of the coil, *determined by arithmetic average*
$T_b$ = temperature of bulk of the tank fluid
$\Delta T_m$ = difference between $T_b$ and $t_m$
$U_m$ = mean heat-transfer coefficient over entire length of tube, including tube wall resistance
$R$ = resistance of tube wall between thermocouple positions and tube surface

$$Q = U_m c \int_0^L \Delta T dl = U_m c L \Delta T_m \qquad (4)$$

$$h = \frac{U_m}{[1 - RU_m]} \qquad (5)$$

Under duplicate conditions, values of $h$ for heating and cooling were calculated from

$$\int_0^L \Delta T dl$$

obtained graphically as shown in Figure 9, and also by the use of $\Delta T_m$ calculated as the difference between $T_b$ and the arithmetic average of the thermocouple readings. The heat-transfer coefficients agreed to $\pm 0.7\%$ in all cases tested, and further calculations were all made from arithmetic averages of the thermocouple temperature readings. Table 10 shows the thermocouple readings used in Figure 9.

### UNSTEADY STATE

In certain runs only one coil heating or cooling was used. The experimental procedure was to run the test over a given temperature interval, exactly bracketing the desired fluid temperature. Initially, data from all readings were plotted as a function of time. The measurements were interpolated to coincide vith the desired fluid temperature.

A calculation also was made averaging arithmetically the values of all the variables recorded at regular increments of time over the time interval, and the results were identical, $\pm 2\%$, with that given in the previous procedure. All succeeding calculations were made averaging all measurements over the time interval.

### COMPARISON BETWEEN UNSTEADY AND STEADY STATE

Duplicate runs made with both steady-state and unsteady-state conditions give excellent agreement between results. Unsteady-state runs are included in Figure 3 and lie within the correlation.

The same heat-transfer coefficients were obtained in steady-state runs where alternate coils were heating and cooling as were obtained in unsteady-state runs in which adjacent coils were both heating and cooling.

### Notation

$B$ = width of baffles
$C$ = impeller distance off tank bottom (measured to horizontal centerline of impeller)
$C_c$ = coil distance from bottom of tank
$C_p$ = specific heat
$c$ = circumference of tube
$D$ = impeller diameter
$D_b$ = length of impeller blade for interrupted blade
$D_c$ = diameter of coil at tube centers
$D_d$ = disk diameter
$D_o$ = outside diameter of coils
$D_w$ = blade width
$d$ = tube diameter
$g$ = gravitational acceleration
$h_c$ = mean cooling coefficient, B.t.u./(hr.)(° F.)(sq.ft.)
$h_h$ = mean heating coefficient, B.t.u./(hr.)(° F.)(sq.ft.)
$h_o$ = mean heat-transfer coefficient, B.t.u./(hr.)(° F.)(sq.ft.) at $\mu_s/\mu = 1.0$
$HP$ = horsepower
$HP_v$ = horsepower per unit volume
$k$ = thermal conductivity
$N$ = impeller speed
$N_p$ = power number, $Pg/\rho N^3 D^5$
$N_{Re}$ = Reynolds number, $ND^2\rho/\mu$
$P$ = power
$R$ = tube wall resistance, (hr.)(° F.)(sq.ft.)/(B.t.u.)/(ft.)
$S_c$ = coil spacing between tube centers
$T$ = tank diameter
$U$ = over-all heat-transfer coefficient, B.t.u./(hr.)(° F.)(sq.ft.)
$Z$ = liquid depth
$Z_c$ = height of coils
$\rho$ = density of fluid
$\mu$ = viscosity at tank temperature
$\mu_s$ = viscosity at tube surface temperature

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