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UNIVERSITY OF LOUISVILLE

CORRELATION OF COOLING DATA FOR KEROSENE

11

A Thesis

Submitted to the Faculty

of the Graduate School

of the University of Louisville

in Partial Fulfillment of the

Requirements for the Degree of

MASTER OF CHEMICAL ENGINEERING

Department of Chemical Engineering

by

James Franklyn Scott

111

1939

JAMES FRANKLYN SCOTT

CORRELATION OF COOLING DATA FOR KEROSENE

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TABLE OF CONTENTS

	Page
LIST OF TABLES	v
LIST OF FIGURES	v
TABLE OF NOMENCLATURE	vi
ACKNOWLEDGMENT	viii
INTRODUCTION	1
HISTORICAL	3
THEORETICAL	14
EXPERIMENTAL	26
Apparatus	27
Operation	33
Test Procedure	34
Physical Properties	36
Data	42
RESULTS AND CONCLUSIONS	50
BIBLIOGRAPHY	63

LIST OF TABLES

	Page
TABLES	
I Engler Distillation Range of Eocene	37
II Heat Transfer Data	44

LIST OF FIGURES

FIGURES	
1 Heat Exchanger	30
2 Sketch of Apparatus	31
3 Effect of Temperature on Viscosity	38
4 Specific Heat at Temperatures of -20 - 140°F.	40
5 Effect of Temperature on Specific Gravity	41
6 Dittus and Boelter Correlation of Eocene Data	52
7 Correlation of Reynolds Number vs. Nusselt Number for Eocene Data	54
8 Friction Factor - Reynolds Number Curve	55
9 Cooling Correlation of Friction Factor vs. Nusselt Number for Kerosene	57
10 Cooling Correlation of Friction Factor vs. Nusselt Number for Oils	58
11 Heating Correlation of Friction Factor vs. Nusselt Number for Oils	59
12 Correlation of Heating and Cooling Coefficients	60

TABLE OF NOMENCLATURE

A = Heat transfer surface	sq. ft.
C = Specific heat at constant pressure	B.T.U./ (lb.) (°F.)
D = Inside diameter of tube	ft.
f = Friction factor in Fanning equation	$\Delta H = \frac{2f L u^2}{g^D}$
g = Acceleration due to gravity	4.18 x 10 ft./ (hr.) ²
G = Mass Velocity	#/ (hr.) (ft.) ²
h = Average surface coefficient of heat transfer between tube surface and fluid	B.T.U./ (hr.) (ft) ² (°F)
ΔH = Drop in fluid head due to friction	ft.
k = Thermal conductivity	B.T.U.-ft/ (hr) (ft) ² (°F)
L = Heated length of tube	ft.
M = Mass	lbs.
Q = Quantity of heat transferred	B.T.U./hr.
r = Factor in Prandtl equation	
t = Temperature difference (in discussion of dimensions)	°F.
t ₁ = Average temperature of fluid entering test section	°F.
t ₂ = Average outlet temperature of fluid	°F.
T ₁ = Average tube temperature at inlet end	°F.
T ₂ = Average tube temperature at outlet end	°F.
u = Average fluid velocity	ft./hr.

U	= Overall coefficient	B.T.U./ $(\text{hr})(\text{ft})^2(^{\circ}\text{F})$
W	= Weight rate of flow	lbs./hr.
Δt	= Temperature rise of fluid	$^{\circ}\text{F.}$
ΔT	= Arithmetical temperature difference	$^{\circ}\text{F.}$
$\ln \Delta T$	= Log mean temperature difference from metal surface to fluid	$^{\circ}\text{F.}$
ρ	= Density of fluid	lbs./cu. ft.
μ	= Viscosity of fluid at main-stream temperature	Centipoises
μ	= Viscosity of fluid at main-stream temperature	lbs./ $(\text{ft})(\text{hr})$
μ_f	= Viscosity of fluid at mean film temperature	lbs./ $(\text{ft})(\text{hr})$
μ_w	= Viscosity of fluid at temperature of inner tube surface	lbs./ $(\text{ft})(\text{hr})$
ϕ	= Time	hr.

Subscripts:

h, c = Represents condition of heating and cooling

Symbols are from the system recommended by the Council of American Institute of Chemical Engineers.

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INTRODUCTION

INTRODUCTION

The purpose of this investigation was to study the behavior of the coefficient of heat transfer under conditions of low temperature cooling. Flow under viscous, critical, and lower turbulent conditions was obtained. A hydrocarbon oil, a purified water-white kerosene known as Eocene, was used in this experimental work.

The experimental data was obtained in a horizontal double-tube heat exchanger having a six foot test section and using a one inch brass pipe. Eocene was also used for a cooling medium.

Liquid temperatures ranging from 60° to 10° F. covered a temperature region not previously investigated. Liquid rates of flow covered a range of Reynolds numbers of from 10 to 13,000.

The principle of dimensional analysis was used in an attempt to obtain a correlation of various groups. The Dittus and Boelter equation and others were compared. A correlation of Nusselt numbers versus friction factors gave satisfactory results with this and other experimental data.

HISTORICAL

HISTORICAL

Osborne Reynolds⁽²⁷⁾ in 1874 predicted an analogy between heat transmission and fluid friction. He proposed the dimensionless equation

$$h = f c_p G / 2$$

relating the heat transfer coefficient with the drop in pressure due to fluid friction. The Reynolds analogy presupposes that all the thermal resistance is in the fluid core.

In 1855 Graetz⁽¹⁰⁾, assuming heat transfer to a fluid in viscous flow was by conduction only, obtained the relation:

$$(t_2 - t_1) / (T - t_1) = 1 - 8P_2$$

where P_2 represents the convergent infinite series.

$$P_2 = 0.10238^{-14.6272n} + 0.01220^{-89.22n} + 0.00237e^{-212n} + \dots$$

and $n = kL / 4Wc_p$. This latter group is the so-called Graetz number which has been correlated by other experimenters. The equation does not involve the coefficient of heat transfer.

In 1909 Nusselt⁽²⁵⁾ applied the principle of dimensional similarity and obtained his well-known general equation for turbulent flow in pipes:

$$hD/k = \phi(DG/\mu) \phi' (c\mu/k) \phi'' (D/L)$$

Assuming heat transfer by conduction and convection to be controlled by the factors entering the differential equations for both hydrodynamics and heat conduction. Nusselt suggested that the introduction of the term D/L , the ratio of the diameter to the heated length, would affect the correlation of the equation. In 1910 Nusselt re-discovered the equation proposed by Graetz for the heat transfer of liquids in viscous flow.

Prandtl⁽²⁶⁾, in 1910, based his analogy on the assumption that part of the thermal resistance is in the core, and the remainder in a film flowing in viscous motion along the pipe wall. He proposed the equation:

$$h = (f c_p G / 2) / (1 - r) + r (c_p z_f / k_f)$$

where $r(c_p z_f / k_f)$ is the thermal resistance of the film and $(1 - r)$ that of the core.

McAdams and Frost⁽²¹⁾, in 1922, discussed the relation between heat transfer and fluid friction and proposed a simplified form of the Nusselt equation for the film coefficient of heat transfer for fluids flowing inside pipes in turbulent motion. They gave the equation:

$$h = 22.6 \quad k/D \quad (DG/\mu)^{.796}$$

obtained from the plot of hD/k vs. DG/μ using experimental data for water from four sources and data for light oils from two sources.

Rice⁽²⁸⁾ recalculated the data of several investigators for both heating and cooling of water and oils and suggested the equation:

$$hD/k_f = 1/80 (Du_f/\mu_f)^{0.83} (c_f\mu_f/k_f)^{0.5}$$

Morris and Whitman⁽²³⁾ correlated data covering heating and cooling of three petroleum oils and water in a horizontal steel pipe. They evaluated the physical properties of the liquids at the film temperature and found that the cooling data values were more than double those for heating. The data were recalculated on main stream temperatures in the final correlation. These data plotted according to the Nusselt co-ordinates did not form a straight line but were of the form:

$$y = (hD/k)(c\mu/k)^{0.37} \text{ vs. } DG/\mu$$

They suggested that for cooling a value of 75% of the heating values be used.

In 1930 Dittus and Boelter⁽⁶⁾ noted the data for both heating and cooling of fluids in the region of turbulent flow could be correlated the equation:

$$hD/k = 0.023 (DG/\mu)^{0.8} (c\mu/k)^m$$

where m is .4 for heating and .3 for cooling.

Lawrence and Sherwood⁽¹⁸⁾ reported data on heating water in four lengths of pipe. They correlated these data using the equation:

$$hD/k = 450 (DG/\mu)^{0.7} (c\mu/k)^{0.5}$$

and concluded that although the length of pipe varied from 59 to 224 diameters, the effect of the length was negligible. They plotted the data of a number of investigators using the method proposed by E.E. Wilson in which the reciprocals of the overall coefficients U is plotted against $s^{0.24}/v^{0.73}$. The slope of the line substantiates the conclusion of neglecting the tube length for equations in the region of turbulent flow.

Drew, Hogan, and McAdams⁽⁹⁾ presented data of Holden⁽¹³⁾ and White⁽³⁴⁾ for heating a hydrocarbon oil flowing in a stream-line motion in a horizontal copper pipe. The data, when plotted as required by the Graetz theory, gave a smooth curve of the required shape. However, the empirical curve calls for substantially greater heating than the theory based on parabolic distribution of mass velocity. Additional data from the literature gave results of the same order of magnitude, although cooling runs fall below heating data. The data plotted with the (DG/μ) replacing the group (Wc/kL) as abscissae showed that for stream-line flow the use of the co-ordinates Wc/kL is preferable to the Reynolds number.

Kirkbride and McCabe⁽¹⁷⁾ reported data on the transfer of heat to liquids flowing at velocities below the isothermal critical velocity. They correlated their data an

those of Morris⁽²²⁾ and Morris and Whitman⁽²⁵⁾ by the Nusselt-Grober theory which expresses the film coefficient in the form:

$$hD/k = (kL/D^2 V_0)$$

The theory correlated the data from the three sources representing a range of viscosity of from 0.6 to 50 centipoises although the co-ordinates do not include a viscosity term. They suggested a general correlation of data for both viscous and turbulent flow using the equation:

$$hD/k = f(k/\rho GD, DG/\mu, L/D)$$

Sherwood, Kiley and Mangsen⁽³⁰⁾ found the effect of the length of the pipe was negligible in the correlation of their data for heating of hydrocarbon oil in turbulent flow through several different lengths of horizontal copper pipe. They plotted data using the co-ordinates suggested by Morris and Whitman⁽²³⁾ and obtained a curve roughly paralleling that of Morris and Whitman, but somewhat lower. For values of (DG/μ) below 2,000, the points were widely scattered although a rough correlation was obtained for each individual tube length. They reported that although a reasonably smooth curve was obtained above the critical point, the Morris and Whitman co-ordinates were clearly inadequate for the correlation of the data in the viscous flow region.

Drew⁽⁸⁾ heated glycerol in a horizontal pipe in viscous flow and plotted his results using the method of Drew, Hogan, and McAdams⁽⁹⁾. The limited amount of data made it impossible to determine a general correlation of the viscous region of flow. Drew suggested that the method of plotting:

$$(t_2 - t_1)/(T - t_1) \text{ vs. } WC/kL$$

is inadequate when data on different liquids were considered and probably unsatisfactory when even the same liquid was handled at different temperature levels.

Lawrence and Hogan⁽¹⁹⁾ in presenting data using the Prandtl-Taylor equation, concluded that this method did not provide sufficient correlation. They recommended correlation of data for fluids flowing in turbulent flow in vertical pipes by means of the Dittus and Boelter equations.

Sherwood and Petrie⁽²⁹⁾ presented data on heat transmission to water, acetone, benzene, kerosene, and n-Butyl alcohol flowing in viscous and turbulent flow through a horizontal copper pipe. The data obtained in the turbulent region were found to be well correlated by the Dittus and Boelter equation:

$$hD/k = 0.024 (DG/\mu)^{0.80} (C\mu/k)^{0.40}$$

Their results indicating that the slope of the curve for kerosene was definitely greater at velocities 1 to 10 times the critical than for other liquids may be compared with data of Morris and

Whitman⁽²³⁾ and of Sherwood, Kiley and Mangsen⁽³⁰⁾ who obtained similar results with other petroleum oils. Their results indicated that the equation of Dittus and Boelter is clearly unreliable when applied to viscous flow.

Cryder and Gilliland⁽⁵⁾ in the heating of different liquids discovered that a dimensionally sound equation, representing the data for all liquids used in these experiments, was not obtainable without the use of variable surface tension. They plotted their data using an equation which best correlated the value; however, this equation was not dimensionless.

Murphree⁽²⁴⁾ presented a method of calculating heat transfer coefficients for fluids flowing in pipes in turbulent motion from existing data on pipe friction and velocity distribution. The method, if applicable only to cases where the temperature gradient, is such that the physical properties of the fluid do not appreciably vary across the pipe cross section. The calculated coefficients are compared with published experimental results on gases, water, and oil. The correlation

$$hD/k \quad \text{vs.} \quad (f_t/f_c) (DG/\mu_f)$$

where f_t is the friction factor for the tube in question and f is the friction factor for the copper tube, resulted in separate curves for the group $(c\mu/k)$.

Kaye and Furnas⁽¹⁵⁾ advanced a theory to explain the difference between the film coefficients of heat transfer on heating and cooling fluids in turbulent flow in contact with solid surfaces. They based their assumptions on the behavior of the liquid film boundary and suggested the relation:

$$h_o/h_h = (\mu_h/\mu_o)^a$$

where $a = n/m$, a constant evaluated by trial. This constant was found empirically to be best taken as 0.5. The correlation requires evaluation of the data on the basis of the mean film temperature. Heating or cooling coefficients may be obtained when data only for the opposite case are known.

J.F. Downie Smith⁽³²⁾ correlated data on heating and cooling of an oil in viscous and turbulent flow in a horizontal copper pipe. He noted close agreement with data of other investigators for turbulent flow data when plotted with the Dittus and Boelter co-ordinates:

$$(hD/k) / (c\mu/k)^n \text{ vs. } (DG/\mu)$$

where $n = 0.3$ for cooling and 0.4 for heating. In viscous flow, the points showed no satisfactory correlation although excellent curves would result if n were evaluated for heating $= 0.21$ and for cooling $= -0.21$. He plotted $t_1 - t_2/t_1 - T_1$ and hD/k as ordinates and W_o/kL as abscissae and obtained

agreement of the same order of magnitude as previously obtained. Distinct curves occurred for definite temperature differences of tube and oil.

Jurgensen, Jr. and Montillion⁽¹⁴⁾ heated water in an inclined pipe and plotted:

$$(hD/k)/(e\mu/k)^{0.4} \text{ vs. } DG/\mu$$

obtaining good correlation of their data. In the horizontal position, the data were compared with equations recommended by Lawrence and Sherwood⁽¹⁸⁾, and Dittus and Boelter⁽⁶⁾. The slope of the Lawrence and Sherwood line is checked almost exactly, however, the data fall distinctly above these three lines. The data indicated that higher coefficients were obtained below a Reynolds number of 20,000 for downward flow than those for horizontal pipes, while upward flow resulted in lower coefficients than those for horizontal pipes. They suggested that the effect of thermal flow in relation to convection mixing in the fluid stream was responsible.

Sieder and Tate⁽³¹⁾ presented data on heat transfer to three oils of widely different temperature coefficients of viscosity in viscous flow in a tube for both heating and cooling. The results are correlated in terms of the main-stream fluid properties by a method which takes into account the viscosity gradient of the fluid in the tube. The data for

both heating and cooling were brought into agreement. Additional data from the literature also correlated well on this basis. They plotted:

$$(hD/k)(\rho/k)^{-1/3} (L/D)^{1/3} (\mu/\mu_w)^{-0.14} \text{ vs. } (DG/\mu)$$

and obtained a straight line in the viscous region. They pointed out that when dealing with liquids with small temperature coefficients of viscosity, the μ/μ_w group requires supplementation by the addition of the Grashof group. The data of Drew⁽⁸⁾ for heating of glycerol in viscous flow were brought into agreement when plotted with these co-ordinates. This method shows definite curves for the ratio of L/D in the viscous region. The effect tapers off through the critical region and disappears entirely in turbulent flow.

Browne and Finger⁽³⁾ and Donahew⁽⁷⁾ designed and constructed the heat exchanger used in this investigation. Previous experimental work of Groth⁽¹¹⁾ and Barnes⁽²⁾ cooling kerosene in a horizontal one inch pipe was correlated by the Dittus and Boelter equation using the Prandtl group as the 0.3 power. These data were predominately in the critical region. The slope of the line best representing their data was greater than 0.8. They obtained the equation:

$$h = k/D (\rho \mu_f/k)^{.3} (DG/\mu)$$

The data were considerably scattered in the viscous region. The values of the heat transfer coefficient were higher than obtained by other investigators.

THEORETICAL

THEORETICAL

Fluids flowing through pipes in isothermal flow are generally considered as possessing either of two types of motion. If the velocity is low, stream-line or viscous flow occurs while for greater velocities turbulent flow exists. In isothermal flow of fluids at low velocities, the fluid has been assumed to have a parabolic velocity gradient across the tube section. This parabolic gradient changes logarithmically as the fluid passes into turbulent flow. A region exists between turbulent and viscous flow in which either viscous or turbulent flow may predominate depending on various variables. The type of flow which will predominate in this region is probably a function of the past history of the fluid stream.

Various investigators have correlated the mass rate of flow of fluids in the form of a dimensionless group with the fluid friction. In the viscous and turbulent regions this method of correlation has resulted in a well established curve for fluid flowing under isothermal conditions. The relation between the friction factor and the Reynolds number which is a criteria of the type of flow existing in a pipe has not been as definitely established at the lower critical region.

Under conditions in which heat is transferred to a liquid body flowing through pipes, a distortion of the type of flow occurs. This has been attributed to convection currents introduced by the temperature gradient.

The transfer of heat produces a temperature gradient across the liquid stream which causes convection currents to be set up perpendicular to the mass velocity of flow. Colburn suggested the term "thermal flow" as applying to the convection currents, or eddy effects set up in the fluid stream due to a temperature gradient. The magnitude of this thermal flow is undoubtedly a function of the temperature gradient existing across the tube section. Also, the effect of these convection currents will be of greater magnitude in the isothermal viscous region than in the turbulent region.

The theoretical equations proposed by Reynolds were based on the assumption that the resistance to heat transfer occurred entirely in the fluid core. Nusselt, in modifying Reynolds' original analogy between fluid friction and heat transfer assumed that part of this resistance occurred in a fictitious film at the tube wall. Graetz's theoretical analogy for fluids flowing in viscous motion is based on the assumption that heat is transferred through the fluid stream entirely by conduction. Later investigators have attempted correlations

of their data along lines proposed by these early theories. These investigators have made use of the original theory of a stagnant or viscous film existing at the tube wall. In most cases they have considered this film the controlling factor in the transmission of heat. Considerable substantiation of this assumption exists since the coefficient of heat transfer definitely increases as some function of the mass rate of flow. Increasing the velocity of flow increases the frictional resistance at the surface of the film. A decrease in the film thickness occurs by actual removal of the molecules of the film.

Some investigators have attacked the problem from a rigorous mathematical viewpoint in an attempt to develop an equation which takes into account the effects produced by this type of eddy flow. These attempts have failed to solve the problem and at best have resulted in individual correlation of data or groups of data. Thermal flow is a function of the viscosity, density, coefficient of thermal conductivity, coefficient of thermal expansion, and gravitational effect, and therefore might be expected to be a function of the liquid employed and temperature gradient. Correlations which are of an empirical nature usually serve to fit some investigators' data for similar conditions and liquids, but, as a whole, do

not provide a general correlation adaptable in any region.

The theory of dimensional similarity has been applied in by far the greatest number of correlations and seems to possess the greatest advantages of any proposed method. Correlations on the basis of dimensionless groups are more logical, and more flexible in application. The advantage of this principle lies in the complete universality which places the experimental data of various investigators on a common basis. Experimental variables of the individual apparatus are thus eliminated. A correlation of heat transfer data may be ultimately obtained by this method. It must be borne in mind, however, that although the relation of the variables has been absolutely defined by this analogy, that these relations apply only if the original assumptions of the variables entering into the phenomena were valid. This limitation has long been recognized and the failure of experimenters to obtain a complete correlation is not due to the method employed. It indicates a failure to consider all the variables affecting the problem in the original assumptions.

Methods of analysis of experimental data have indicated the necessity of the introduction of the tube length for flow in viscous and critical regions. It is unfortunate that in this respect the effect of the tube length upon the heat transfer coefficient in short tubes might be attributed

to end turbulence effects masking the actual relation of tube length to the coefficient. On another basis this effect might not be observed. This is especially true in the region of viscous flow. The Prandtl group $(c\mu/k)$ appears in the Nusselt type equation usually to some odd power. This group, which relates only the physical properties of the liquid, has been introduced to extend the range of a simpler correlation by making it applicable to various liquids. Other functional groups have been added to this type of correlation to evaluate the effects of convection currents such as the ratio of μ/μ_f and μ/μ_w and the Grashof group.

In correlation using the Nusselt type equation some few investigators, Sherwood and Petrie⁽²⁹⁾, Sieder and Tate⁽³¹⁾, investigated kerosene among other liquids. Generally it has been found that the value of the kerosene coefficients have increased at a greater rate than the coefficients of other liquids. It has generally been considered that the slope of the line representing kerosene data was definitely greater than 0.8 in the region of from 1 to 10 times the critical velocity. This seemed to indicate that the method of correlation was not adequate for all conditions. Sieder and Tate obtained correlation of kerosene data of the same order of magnitude as other liquids by their assumption of the factors affecting the heat transfer coefficient through a liquid film was more nearly correct.

The equation of Sieder and Tate, with the exception of the group μ/μ_w which is evaluated at some fractional power, is the Nusselt type employed by most investigators. While this equation is the most common type in turbulent flow, its use in the viscous region seems to be an important contribution.

The ratio of the viscosity at the main temperature to the viscosity at the tube wall temperatures seems to be an arbitrary selection which is similar to ratios used by Colburn⁽⁴⁾ and others in their methods of correlation. These investigators employed, however, a ratio in which the viscosity was taken at the film temperature rather than the wall temperature. This dimensionless group is a correction factor employed as a method of compensating for the effects of thermal flow. The use of viscosity at the wall temperature is based on the assumption of heat flow through the liquid film and core. The use of an average film temperature assumes heat flow through a portion of the film and the liquid core. The effect is somewhat more quantitative than previous partial assumptions. Unfortunately, as Sieder and Tate have pointed out, this method of correlation is limited to large temperature gradients and consequently large ratio of μ/μ_w . For small temperature gradients where the viscosity of the liquid has undergone a relatively small change across the

tube section, this method of analysis is inadequate. It requires the possible addition of the Grashof number appearing as some power. This agrees with the method of Colburn⁽⁴⁾ in which an analogy between fluid friction and heat transfer lead to the proposal of a heat friction group similar to the dimensionless friction factor group. This heat transfer term containing the ratio of μ/μ_w and a functional power of the Grashof number is plotted against the Reynolds group. Colburn plotted friction factors versus Reynolds numbers also and showed the similarity existing between the two fields.

These investigators have resulted in correlations for both heating and cooling data of film coefficients. Very few data are available for cooling; most of the work appearing in literature is heating data. However, the correlations with certain corrections for heating and cooling show a similarity. Of the correlations for cooling to date the Dittus and Boelter and its modifications by Sieder and Tate seem to represent the best.

A rigorous study of the variables encountered in this field was undertaken. Since it was felt that an empirical form did not furnish any contribution toward a general solution of such a complex phenomena, this study was based on the use of the principle of dimensional similitude. Originally, a correlation of temperature versus various functional groups was quite

thoroughly investigated using data from our laboratory.

The physical properties were determined experimentally except "k" which was assumed constant. The value was consistent with that of other investigators. Correlations on the basis of these liquid properties and the mass rate of flow as functions of the temperature and in mean temperature drop were employed. The Reynolds, Nusselt, Prandtl, and Graetz groups were employed in this work. Other groups containing the coefficient of thermal expansion were not used as this value was not known for the material investigated. The investigation in this direction was abandoned finally, as no correlation could be obtained. Other methods of grouping were attempted however.

A plot of the Reynolds number versus the ordinate suggested by Dittus and Boelter was found to give a fair degree of correlation of the data through the critical region. The data could be represented by a straight line breaking off toward the horizontal at the lower critical.

Correlations involving bulk data are desirable since no arbitrary assumptions are necessary. Data of this nature are readily attainable. Since the Reynolds number-friction factor relationship is evaluated at bulk conditions, it seems possible that correlations of other groups could be made in the

same manner. Other experimenters have indicated the desirability of using bulk-liquid temperatures.

Analogies have been continually advanced between fluid friction and the heat transfer coefficient. This type of correlation employs bulk temperature evaluation of the variables. Therefore, functional relations of this nature were investigated.

It has been recognized by Keevil and McAdams⁽¹⁶⁾ and others that the heating and cooling of a fluid affect the friction factor. The friction factor is, however, a function of various variables of which the pressure drop and velocity are the controlling factors. It is to be expected from the analogy of a fictitious film at the tube wall, that thermal flow would alter the characteristics of this film. The pressure drop will be a measure of not only the velocity flow but also the thermal flow. On this basis the coefficient of heat transfer which is dependent upon the flow conditions becomes a function of the pressure drop.

The heat transfer coefficient, h , is dependent upon the factors affecting the pressure drop and the thermal conductivity of the fluid, k .

$$h = \phi (k, -dP, D, DL, V, \rho)$$

In absolute units this relationship appears as:

$$\frac{M}{\rho^3 t} = \left(\frac{ML}{\rho^3 t} \right)^a \left(\frac{M}{L \rho^2} \right)^b (L)^c (L)^d \left(\frac{M}{L^3} \right)^f$$

Solving the fundamental equations simultaneously gives the relationship of the variables.

$$h = (k/D)^1 (-dP/V^2)^b (dL/D)^d$$

This functional expression can be arranged to obtain the pressure drop:

$$-dP/V^2 \rho = \phi \ hD/k, (dL/D)$$

Applying the pi theorem to the analysis, where there are seven variables, and four fundamental units, there are, therefore, three dimensionless groups. While this is the final result obtainable by dimensional analysis alone, it is physically obvious that fluid friction will be proportional to the length of the pipe,

$$\text{or} \quad -dPg/V^2 = dL/D \ \phi(hD/k)$$

Rearranging and converting pressure drop to engineering units by dividing by g:

$$-dPgD/V^2 dL = \phi \ hD/k$$

This left member is the familiar form of the Fanning equation and therefore may be replaced by $2f$.

$$2f = -dPgD/V^2 dL = \phi \ hD/k$$

Which relates friction factor and heat transfer coefficient as:

$$f = \phi \ hD/k$$

The equation of Dittus and Boelter for cooling will be plotted as well as a study of hD/k vs. f for this and other data. The equation of Sieder and Tate will also be discussed in correlation of the data.

EXPERIMENTAL

APPARATUS

APPARATUS

The apparatus, Figs. 1 and 2, consists of a horizontal counter-current liquid-liquid heat exchanger (A) and auxiliary equipment. The heat exchanger, pumps, and storage tanks are completely insulated for low temperature operation.

The auxiliary equipment consists of a refrigerating unit (B), refrigerant storage (C), test liquid storage (D), and circulating pumps (E). The heat exchanger consists of a double-tube test section (1), mixing chambers (2), calming sections (3), and piezometer rings (4). Liquid and pipe thermocouples are designated as TC1-16.

The inner tube is a standard one inch brass pipe, inside diameter 1.049 inches, 9 feet 6 inches long. An annular cast iron shell forms a test section of exactly 6 feet. The remaining lengths of pipe on either end are calming sections. These prevent end turbulence from affecting the pipe temperature measurements. The entrance calming section contains a copper calming cross.

The shell is assembled in eight sections. Three split-sleeves (5) are provided for thermocouple installation. Thermocouple leads are removed from the liquid space through

FIG. 1

HEAT EXCHANGER

bolts in these sleeves.

Mixing chambers are provided in the inlet and outlet lines of the test section. The mixing chambers are preceded or followed by calming sections. Thermocouples were installed at the outlet end of these chambers. All liquid temperatures are average or bulk temperatures determined in this manner.

Sixteen #24 B&S copper-constantan thermocouples were used for temperature measurements. Liquid temperatures were measured by the thermocouples in the mixing chambers. Twelve thermocouples were used to determine the average pipe temperature at three points. Entrance, middle, and exit measurements were obtained by four thermocouples spaced 90° around the circumference of the pipe. Pipe thermocouples were installed by the method of Hebbard and Badger⁽¹²⁾. The thermocouples were originally checked for uniformity of measurement.

The leads were connected to two 16 point selective switches. A single cold junction in series with the selective switches was maintained in a freezing mixture in a vacuum flask. A Leeds and Northrup type K potentiometer was used for reading all thermocouples. The accuracy of the instrument was 0.01 millivolts, corresponding to 0.03° F.

Thermometer wells allowed a continual check on the thermocouples. These thermometers also provided a means of visual operation.

The liquids were circulated through the heat exchanger by small gear pumps. Liquid rates of flow were controlled by by-pass valves around the circulating pumps. Provisions were made for by-passing the liquid stream into a weighing tank. The test liquid flow was weighed over a measured period ranging from 2 minutes to 15 seconds. The liquid was returned to the storage by gravity flow.

The temperature of the test liquid was measured as it left the entrance mixing chamber. The liquid transversed the isothermal calming sections before entering the test section. On leaving the test section it flowed through a similar isothermal calming section. The exit temperature of liquid was obtained after passing through a mixing chamber. The liquid was returned to the storage or by-passed to the weighing tank.

The cooling kerosene was recirculated through the annular chamber surrounding the inner tube. The inlet and outlet temperatures from this section were measured after passing the mixing chambers.

TEST PROCEDURE

TEST PROCEDURE

The kerosene for cooling was refrigerated for a period of 3 to 6 hours prior to circulation in the exchanger. The test and refrigerant liquid pumps were started when the cooling kerosene reached the desired temperature. Liquid temperatures and rates of flow were controlled by means of by-pass lines around the pumps and test liquid reservoir. A period of from 6 to 8 hours was then required for the entire system to reach equilibrium conditions.

Each run consisted of obtaining readings for a definite rate of flow. From 50 to 100 runs were obtained after the system had been brought to a steady state. Pressure drops, mass rates of flow, liquid and pipe temperatures were recorded for each set of conditions.

The quantity of heat transferred was determined from the test liquid data. The rate of flow of the cooling kerosene was constant throughout the experiment.

PHYSICAL PROPERTIES

bolts in these sleeves.

Mixing chambers are provided in the inlet and outlet lines of the test section. The mixing chambers are preceded or followed by calming sections. Thermocouples were installed at the outlet end of these chambers. All liquid temperatures are average or bulk temperatures determined in this manner.

Sixteen #24 B&S copper-constantan thermocouples were used for temperature measurements. Liquid temperatures were measured by the thermocouples in the mixing chambers. Twelve thermocouples were used to determine the average pipe temperature at three points. Entrance, middle, and exit measurements were obtained by four thermocouples spaced 90° around the circumference of the pipe. Pipe thermocouples were installed by the method of Hebbard and Badger⁽¹²⁾. The thermocouples were originally checked for uniformity of measurement.

The leads were connected to two 16 point selective switches. A single cold junction in series with the selective switches was maintained in a freezing mixture in a vacuum flask. A Leeds and Northrup type K potentiometer was used for reading all thermocouples. The accuracy of the instrument was 0.01 millivolts, corresponding to 0.03° F.

Thermometer wells allowed a continual check on the thermocouples. These thermometers also provided a means of visual operation.

The liquids were circulated through the heat exchanger by small gear pumps. Liquid rates of flow were controlled by by-pass valves around the circulating pumps. Provisions were made for by-passing the liquid stream into a weighing tank. The test liquid flow was weighed over a measured period ranging from 2 minutes to 15 seconds. The liquid was returned to the storage by gravity flow.

The temperature of the test liquid was measured as it left the entrance mixing chamber. The liquid transversed the isothermal calming sections before entering the test section. On leaving the test section it flowed through a similar isothermal calming section. The exit temperature of liquid was obtained after passing through a mixing chamber. The liquid was returned to the storage or by-passed to the weighing tank.

The cooling kerosene was recirculated through the annular chamber surrounding the inner tube. The inlet and outlet temperatures from this section were measured after passing the mixing chambers.

TEST PROCEDURE

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The kerosene for cooling was refrigerated for a period of 3 to 6 hours prior to circulation in the exchanger. The test and refrigerant liquid pumps were started when the cooling kerosene reached the desired temperature. Liquid temperatures and rates of flow were controlled by means of by-pass lines around the pumps and test liquid reservoir. A period of from 6 to 8 hours was then required for the entire system to reach equilibrium conditions.

Each run consisted of obtaining readings for a definite rate of flow. From 50 to 100 runs were obtained after the system had been brought to a steady state. Pressure drops, mass rates of flow, liquid and pipe temperatures were recorded for each set of conditions.

The quantity of heat transferred was determined from the test liquid data. The rate of flow of the cooling kerosene was constant throughout the experiment.

PHYSICAL PROPERTIES

PHYSICAL PROPERTIES

The physical properties of Eocene were determined experimentally with the exception of the thermal conductivity. The thermal conductivity was taken as 0.0875⁽²⁷⁾. The distillation range, Table I, was similar to that of other experimenters using this value for kerosene.

TABLE I

ENGLER DISTILLATION RANGE OF EOCENE

Distilled over	Eocene
%	°F.
0	325.
10	355.
20	370.
30	387.
40	402.
50	417.
60	432.
70	449.
80	471.
90	492.

The viscosity was determined using a Hoeppler viscosimeter. The variation of the viscosity over the range of 5 to 70° F. is shown in Fig. 3.

Fahrenheit

Specific heats were determined by measuring the temperature rise of a known weight of liquid, brought about by the introduction of a measured quantity of electricity. Specific heats, Fig. 4, were determined in a Dewar flask equipped with a heating element and stirrer. A watt-hour meter recorded the energy input. The temperature rise in the liquid was measured by means of a thermocouple. Calorimeter corrections were obtained by determining the specific heats of water under similar conditions. These corrections were applied to obtain the actual specific heats of the kerosene. Extrapolation of the corrections to be used were necessary for values below 32° F.

The specific gravity was obtained over a range of from -20° to 80° F. by a Westphal balance. The curve obtained, Fig. 5, was checked at several points by pycnometer determinations.

Fig. 4 Specific Heat at Temperatures of -20 to +140 °F.

Fig. 5 Effect of Temperature on Specific Gravity

Temperature in Fahrenheit

Specific Gravity

DATA

DATA

The experimental data is tabulated in Table II. The data has been tabulated in the order of increasing Reynolds numbers. That data marked * is from the work of Barnes⁽²⁾ and Scott, and that data marked + is from the work of Groth⁽¹¹⁾ and Barnes.

The inlet liquid temperature t_1 and outlet temperature t_2 are bulk temperatures obtained after the liquid has passed through mixing chambers. T_1 and T_2 are average pipe temperatures at the liquid inlet and outlet. The average liquid temperature is the arithmetic average of t_1 and t_2 . The mass rate of flow is given as lbs. per hour obtained by weighing the liquid flow.

The ln mean ΔT was calculated from the temperature differences at inlet and outlet liquid conditions. The heat transfer coefficient was calculated on the basis of the ln mean ΔT . The thermal conductivity k , appearing in the Nusselt and Prandtl groups has been evaluated at a constant value of 0.0875. The physical properties of the liquid entering the Reynolds and Prandtl groups are taken at bulk conditions. The friction factor, f , was obtained from an enlarged plot of the isothermal friction factor vs. Reynolds number. The run numbers correspond to consecutive numbering of the experimental data in order of increasing Reynolds numbers and not in chronological order.

TABLE II HEAT EXCHANGER DATA

Rum No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	In Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{hD}{k} \left(\frac{c_p \mu}{k} \right)^{.3}$
1	48.162	27.471	21.788	19.302	15.56	37.817	69.4	179.	.083	26.0	25.9	29.8	9.35
2	47.884	30.046	23.209	21.033	15.42	38.815	75.6	196.	.083	24.2	24.1	29.6	8.70
3	41.724	31.689	22.454	21.699	14.13	36.707	135.6	246.	.065	27.2	27.1	30.2	9.72
4	41.857	29.469	21.255	19.524	14.62	35.663	100.2	282.	.065	23.8	23.7	30.6	8.50
5	42.700	32.222	23.632	21.877	14.67	37.461	116.4	298.	.054	24.7	24.6	30.1	8.53
6	42.478	34.131	24.363	23.475	14.34	38.305	139.0	361.	.045	23.3	23.2	29.7	8.41
7	41.546	34.309	26.006	25.917	11.61	37.928	147.0	380.	.043	25.8	25.7	29.6	9.32
8	40.347	32.977	23.830	24.097	12.31	36.662	161.0	411.	.039	27.2	27.1	30.3	9.78
9	40.614	33.421	21.655	21.211	15.40	37.018	184.0	470.	.035	24.2	24.1	30.2	8.67
10	38.971	32.488	21.122	21.699	14.05	35.730	210.0	528.	.032	27.8	27.7	30.6	9.91
11	38.088	32.633	21.122	20.545	14.42	35.286	261.0	655.	.026	27.9	27.8	30.7	9.95
12	37.994	33.021	20.456	19.790	15.28	35.508	265.8	659.	.026	24.2	24.1	30.7	8.64
13 ⁺	59.72	53.24	43.7	36.2	16.67	56.48	225.0	728.	.0218	25.1	25.0	24.2	9.6
14	25.718	22.632	11.951	10.599	13.06	23.675	413.0	898.	.0177	27.1	27.0	35.0	9.3
15	26.050	22.765	9.400	9.489	14.98	24.408	413.0	905.	.0176	25.2	25.1	34.7	8.7
16	26.517	22.987	10.599	9.978	14.42	24.653	413.0	908.	.0175	26.5	26.4	34.7	9.1
17	23.475	20.456	7.580	6.492	14.73	21.966	458.0	962.	.0155	26.0	26.0	36.2	8.9
18	22.054	19.701	8.335	8.157	12.47	20.878	505.0	1041.	.0158	26.3	26.3	36.7	9.0
19	23.298	20.856	9.090	8.912	12.99	22.077	510.0	1073.	.0156	27.5	27.5	36.0	9.4
20	36.529	33.421	20.678	20.101	14.93	34.975	435.0	1088.	.016	25.2	25.1	30.7	8.99
21	21.832	19.524	8.113	8.601	12.32	20.678	533.	1100.	.0155	27.6	27.6	36.8	9.4
22	22.720	20.145	8.290	7.580	13.39	21.443	533.	1115.	.0155	28.4	28.4	36.3	9.7
23	22.276	19.790	7.935	7.713	13.11	21.053	540.0	1120.	.0154	28.4	28.4	36.6	9.7
24	21.300	18.946	7.447	7.180	12.63	20.123	563.0	1150.	.0153	28.9	28.9	37.1	9.8
25 ⁺	63.75	59.72	46.8	42.0	17.07	61.74	341.0	1180.	.0151	23.2	23.1	22.8	9.1

* Data of Barnes and Scott

+ Data of Groth and Barnes

TABLE II HEAT EXCHANGER DATA

Run No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	In Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{hD}{k} \left(\frac{c_p \mu}{k} \right)^{.3}$
26	22.864	20.589	9.001	8.557	15.00	21.722	570.0	1198.	.0150	27.6	27.6	36.2	9.4
27+	65.84	62.11	4.97	4.48	16.78	63.98	549.0	1543.	.0150	22.3	22.2	22.2	8.7
28*	63.09	59.00	45.5	39.7	18.23	61.06	598.0	1300.	.0147	25.6	25.5	23.0	9.9
29	22.543	20.234	9.711	8.912	12.12	21.389	623.0	1302.	.0147	32.7	32.7	36.3	11.2
30*	42.731	40.654	33.315	27.984	11.00	41.693	488.0	1329.	.0146	26.0	25.9	28.4	9.5
31+	63.09	59.00	45.5	39.7	18.23	61.06	598.0	1370.	.0145	25.6	25.5	23.0	9.9
32+	64.56	62.76	45.75	41.36	21.10	64.66	394.0	1410.	.0144	20.4	20.3	22.1	8.0
33*	43.354	40.827	31.342	24.869	13.95	42.091	518.0	1418.	.0143	26.4	26.3	28.3	9.6
34	21.300	19.612	8.157	7.447	12.38	20.456	690.0	1419.	.0143	26.2	26.2	36.9	8.9
35	23.564	22.143	13.574	13.796	8.91	22.854	683.0	1460.	.0141	30.3	30.5	35.6	10.4
36	19.657	18.236	7.713	7.846	11.09	18.947	720.0	1460.	.0141	25.4	25.4	37.6	8.5
37	60.06	56.21	41.5	35.6	19.38	58.15	454.0	1802.	.0140	26.1	26.0	23.8	10.0
38+	20.278	18.502	6.115	5.493	13.50	19.390	855.0	1727.	.0138	31.2	31.2	37.5	10.5
39	56.21	52.52	39.6	34.6	17.27	54.37	551.0	1755.	.0133	31.9	31.8	24.6	12.4
40+	63.43	60.44	48.29	44.11	15.60	61.94	506.0	1760.	.0133	23.6	23.7	22.7	9.3
41	20.772	19.346	9.445	9.178	10.59	20.034	870.0	1777.	.0132	31.4	31.4	37.1	10.6
42	24.319	22.498	10.022	9.134	13.831	23.409	833.0	1810.	.0131	30.6	30.4	35.0	10.5
43	21.966	20.456	10.555	10.999	10.30	20.643	915.0	1888.	.0129	37.0	36.9	36.8	12.5
44	19.257	17.925	6.159	6.559	12.29	18.891	945.0	1890.	.0129	28.2	28.2	37.9	9.5
45	21.966	20.456	10.555	10.999	10.31	21.211	915.0	1904.	.0128	37.0	37.0	36.5	12.6
46*	42.731	41.173	36.233	32.000	7.79	41.952	705.0	1920.	.0128	39.8	38.7	28.4	14.5
47	18.680	17.170	6.426	6.426	10.50	17.925	1020.0	2011.	.0127	40.6	40.5	38.4	13.5
48	23.875	23.321	12.952	11.842	10.51	23.098	960.0	2056.	.0125	39.4	39.4	35.5	13.5
49	21.966	20.767	12.286	11.842	9.13	21.367	990.0	2060.	.0125	35.9	35.9	36.5	12.2
50	20.145	19.124	11.709	11.443	8.05	19.635	1060.0	2128.	.0124	36.8	36.8	37.4	12.5

TABLE II HEAT EXCHANGER DATA

Run No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	In Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{hD}{k} \left(\frac{c_p \mu}{k} \right)^{.3}$
51	18.414	17.215	7.180	7.180	10.63	17.815	1148.0	2255.	.0121	35.7	35.7	38.4	12.0
52	20.190	10.035	9.400	9.223	10.22	19.035	1125.0	2264.	.0121	35.4	35.4	37.6	12.0
53 ⁺	55.44	52.285	41.450	30.146	17.77	53.842	720.0	2270.	.0121	38.0	35.3	24.9	13.7
54	21.832	20.323	10.999	11.398	9.85	21.078	1110.0	2300.	.0120	47.0	48.9	36.6	16.0
55	48.997	46.681	42.523	39.269	7.45	48.339	803.0	2380.	.0119	40.4	40.3	26.2	15.1
56 ⁺	55.24	52.88	41.6	33.8	11.15	54.06	754.0	2387.	.0119	45.7	45.6	24.8	17.4
57	21.122	19.834	11.754	11.398	8.82	20.478	1170.0	2412.	.0118	47.3	47.2	36.8	16.1
58	30.801	28.670	16.504	15.972	13.48	29.736	1035.0	2413.	.0118	45.7	45.6	32.8	16.0
59 [*]	42.318	40.516	32.798	20.023	10.44	41.416	915.0	2480.	.0117	44.3	44.1	28.6	16.2
60	15.972	14.906	5.538	5.982	9.60	15.459	1320.0	2500.	.0117	40.3	40.2	39.9	13.3
61 [*]	42.246	41.138	36.327	33.593	5.732	41.692	923.0	2510.	.0117	50.3	50.1	28.4	18.3
62	12.819	11.931	3.140	3.540	8.90	12.370	1395.0	2510.	.0117	38.3	38.3	41.8	12.6
63	25.029	23.875	15.350	14.773	9.391	24.452	1200.0	2630.	.0115	40.9	40.8	37.4	14.1
64 ⁺	59.36	57.56	51.4	46.3	9.52	58.46	795.0	2640.	.0115	43.1	43.0	23.7	16.7
65	25.740	24.807	18.991	19.390	6.09	25.274	1223.0	2710.	.0114	52.0	51.9	34.4	18.0
66 ⁺	61.12	59.36	52.4	46.2	10.78	60.24	799.0	2720.	.0113	37.4	37.3	23.2	14.6
67	19.257	18.058	8.690	8.290	10.22	18.658	1365.0	2730.	.0113	44.0	43.9	37.9	14.8
68	17.304	16.282	7.03	7.180	9.72	16.793	1425.0	2766.	.0113	41.2	41.1	38.9	13.7
69 ⁺	63.75	60.08	44.60	33.44	22.63	61.92	795.0	2770.	.0113	37.0	36.9	22.7	14.6
70 [*]	80.381	49.240	46.123	41.484	5.84	49.811	930.0	2800.	.0112	51.9	51.8	26.5	19.4
71	15.794	14.840	6.159	6.293	8.88	15.217	1500.0	2827.	.0112	53.1	53.0	40.0	17.6
72 ⁺	62.11	60.08	52.03	46.2	11.70	61.10	851.0	2930.	.0111	42.6	42.5	23.0	16.6
73	24.896	23.984	16.480	16.016	8.192	24.430	1388.0	3040.	.0110	44.0	43.9	34.7	15.1
74 [*]	44.011	42.260	35.046	30.061	10.45	43.146	1118.0	3080.	.0110	52.6	52.3	28.1	19.2
75 ⁺	66.56	62.11	48.1	35.7	22.16	64.34	882.0	3120.	.0109	51.1	51.0	22.1	20.7

TABLE II HEAT EXCHANGER DATA

Run No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	ln Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{hD}{k} \left(\frac{c_p \mu}{k} \right)^{0.3}$
76 ⁺	55.88	54.91	47.2	41.8	11.26	55.90	979.0	3150.	.0109	49.2	49.1	24.4	18.9
77	21.078	19.701	12.064	11.887	8.36	20.390	1560.0	3205.	.0107	71.2	71.1	36.9	24.1
78	20.367	19.435	12.153	11.887	7.70	19.901	1575.0	3213.	.0107	82.6	82.6	37.1	17.8
79	49.239	47.167	39.615	33.407	11.71	48.203	1110.0	3273.	.0106	55.9	55.7	26.4	20.5
80	24.940	28.097	17.170	16.904	7.482	24.519	1800.0	3290.	.0106	47.1	47.0	34.8	16.2
81 ⁺	59.867	56.231	43.769	33.350	19.28	58.049	998.0	3305.	.0106	54.2	54.1	23.8	20.9
82	15.972	15.128	7.136	7.225	8.27	15.860	1785.0	3390.	.0106	80.1	80.0	39.8	16.6
83	19.868	18.813	12.109	12.298	6.95	19.191	1680.0	3392.	.0106	51.0	50.9	37.5	17.2
84	19.868	18.813	12.109	12.286	6.95	19.191	1680.0	3392.	.0106	50.5	50.4	37.5	17.0
85 ⁺	63.75	60.44	46.02	37.13	20.54	62.10	975.0	3400.	.0106	45.2	45.1	22.7	17.7
86	17.259	16.016	7.946	8.379	8.50	16.638	1770.0	3426.	.0106	71.1	71.0	39.0	23.7
87	12.464	11.709	3.984	4.561	7.72	12.087	1920.0	3430.	.0106	53.2	53.1	42.1	17.3
88	21.744	20.856	13.307	12.775	8.24	21.300	1865.0	3466.	.0106	49.8	49.7	36.5	16.6
89	13.130	12.242	4.161	4.739	8.16	12.686	1920.0	3480.	.0104	57.4	57.3	41.4	18.7
90	23.298	22.365	17.170	17.304	6.58	22.837	1680.0	3587.	.0103	77.8	77.7	35.6	26.3
91	21.566	20.456	12.908	13.041	7.94	21.011	1740.0	3610.	.0105	67.5	66.4	36.6	23.0
92 ⁺	44.046	41.900	34.873	22.931	13.51	42.973	1545.0	3690.	.0101	69.5	69.3	28.1	25.5
93	14.462	13.707	7.03	7.092	7.07	14.065	2205.0	4080.	.0100	66.5	66.4	40.7	21.8
94	18.236	17.570	11.576	11.487	6.28	17.903	2188.0	4280.	.0098	63.3	63.2	38.3	21.1
95	12.153	11.602	6.470	6.603	5.31	11.878	2490.0	4310.	.0098	71.0	70.9	42.2	23.1
96	21.610	20.634	14.107	12.974	6.97	21.122	2100.0	4350.	.0098	81.4	81.3	36.6	27.7
97	20.545	19.746	14.517	14.684	5.39	20.146	2188.0	4450.	.0097	88.5	88.4	37.0	30.0
98	21.168	20.367	15.217	14.684	5.77	20.767	2168.0	4490.	.0096	83.0	82.9	36.7	28.2
99 ⁺	54.55	54.16	45.7	37.1	11.62	53.21	1489.0	4660.	.0096	76.9	76.7	25.0	29.2
100	19.657	19.123	13.796	13.396	5.80	19.391	2363.0	4770.	.0095	60.0	59.9	37.5	20.2

TABLE II HEAT EXCHANGER DATA

Run No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	ln Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{\frac{hD}{k}}{(\frac{c_p \mu}{k})^3}$
101	25.651	24.674	17.881	17.926	7.260	25.163	2228.0	4940.	.0094	83.7	83.6	34.4	28.9
102+	54.673	52.631	42.489	34.181	15.14	53.652	1898.0	8030.	.0093	61.7	61.6	24.9	23.5
103	30.712	29.869	22.632	21.477	8.236	30.291	2160.0	5080.	.0093	61.5	61.4	32.6	21.6
104+	64.067	58.827	43.736	30.477	23.98	61.447	1615.0	5210.	.0092	96.0	94.8	23.0	37.0
105*	49.689	47.681	39.616	34.112	11.68	48.686	1755.0	5230.	.0092	85.7	85.5	26.1	32.1
106	21.388	20.678	16.194	15.261	5.30	21.033	2625.0	6450.	.0091	97.4	97.3	36.6	33.1
107+	52.215	50.888	42.004	33.904	13.20	51.401	1860.0	5720.	.0090	65.3	65.1	25.4	24.6
108+	50.315	48.407	40.170	34.712	11.86	49.459	1920.0	5790.	.0090	87.8	87.6	26.5	32.8
109+	57.997	55.019	46.366	35.670	15.17	56.508	1792.0	5820.	.0089	101.0	100.0	24.2	38.7
110	32.577	31.423	25.207	24.630	7.082	32.000	2520.0	6080.	.0088	114.8	114.5	31.9	40.6
111	32.488	31.512	25.873	25.251	6.438	32.000	2520.0	6080.	.0088	107.0	106.9	31.9	37.9
112	32.755	31.600	25.718	24.452	7.093	32.178	2565.0	6180.	.0088	117.0	116.8	31.9	41.4
113	30.313	29.336	22.766	21.655	7.615	30.825	2640.0	6190.	.0088	94.5	94.3	32.7	32.2
114	32.178	31.201	25.828	25.296	6.128	31.690	2640.0	6330.	.0087	117.5	117.3	31.9	41.6
115	23.697	22.898	18.147	17.348	5.550	23.298	2850.0	6390.	.0087	113.9	113.7	35.2	39.1
116	29.647	28.714	22.765	22.054	6.771	29.181	2855.0	6600.	.0086	109.2	109.0	32.8	38.2
117	29.558	28.670	23.253	22.498	6.239	29.114	2925.0	6800.	.0085	116.0	115.8	33.0	40.6
118	34.398	33.732	28.448	28.093	5.795	34.065	2790.0	6890.	.0085	90.0	89.6	31.2	32.0
119	31.772	30.969	25.917	25.207	5.817	31.379	2903.0	6940.	.0085	111.0	111.0	32.1	39.2
120	31.156	30.402	25.074	24.940	5.772	30.779	3015.0	7110.	.0084	109.5	109.2	32.5	38.6
121	31.689	31.068	25.873	24.940	5.972	31.379	3015.0	7200.	.0084	87.5	87.3	32.1	30.8
122	32.977	32.222	26.850	26.050	6.150	32.800	3105.0	7300.	.0084	107.0	106.9	31.8	37.9
123	30.357	29.802	24.230	23.076	6.327	29.980	3160.0	7410.	.0083	104.7	104.5	32.6	36.8
124	31.423	30.490	25.296	24.541	6.038	30.957	3195.0	7540.	.0082	138.2	138.0	32.4	48.6
125	29.425	28.561	23.520	23.475	5.508	29.003	3360.0	7810.	.0082	143.8	143.5	33.0	50.4

TABLE II HEAT EXCHANGER DATA

Run No.	Test Liquid Inlet	Test Liquid Outlet	Inlet Pipe Temp.	Outlet Pipe Temp.	ln Mean Δt	Av. Liquid Temp.	#/hr	$\frac{DG}{\mu}$	f	h	$\frac{hD}{k}$	$\frac{c_p \mu}{k}$	$\frac{\frac{hD}{k}}{\left(\frac{c_p \mu}{k}\right)^{.3}}$
126	29.913	29.114	23.919	23.742	5.683	29.514	3580.0	7630.	.0082	132.2	132.0	32.8	46.5
127	30.135	29.469	24.807	24.519	5.239	29.802	3580.0	7860.	.0082	119.1	118.9	32.7	41.8
128	29.958	29.247	24.141	23.897	5.684	29.803	3555.0	8280.	.0081	124.4	124.2	32.8	43.6
129	30.180	29.425	24.097	24.008	5.750	29.803	3735.0	8740.	.0080	136.8	136.6	32.7	48.0
130	31.201	30.446	25.029	24.763	5.928	30.824	3825.0	9030.	.0079	136.4	136.0	32.5	48.0
131	30.846	30.224	25.651	25.296	5.062	30.585	3915.0	9240.	.0079	134.5	134.3	32.4	47.5

RESULTS AND CONCLUSIONS

RESULTS and CONCLUSIONS

The data obtained in this investigation and that of Barnes⁽²⁾ and Groth⁽¹¹⁾ recalculated on a main-stream liquid temperature basis were plotted using the equation of Dittus and Boelter.

$$h = .023 \ k/D \ (c \mu / k)^{.3} \ (DG/\mu)^{.8}$$

Where the viscosity is at the film condition, μ_f , the exponent of the Prandtl group is modified to the .33 power.

The plot of the combined data, Fig. 6, is of the form:

$$hD/k / (c \mu / k)^{.3} \text{ vs. } DG/\mu$$

A fair degree of correlation in the upper critical and lower turbulent regions was obtained. In the lower critical and viscous regions the data did not follow the Dittus and Boelter correlation. The equation of the line best representing the data through the upper critical and lower turbulent region was:

$$h = .01 \ k/D \ (c \mu / k)^{.3} \ (DG/\mu)^{.93}$$

The slope of the line is consistent with the results of other experimenters who obtained slopes greater than 0.8 for kerosene. However, the values of the coefficients were definitely greater than those generally obtained. Barnes⁽²⁾, representing his data for the cooling of kerosene, proposed the equation:

$$h = .0387 \ k/D \ (c \mu_f / k)^{.3} \ (DG/\mu)^{.801}$$

Fig. 6 - Dittus and Boelter Correlation of Eocene Data.

evaluating the viscosity in the Prandtl group at the film temperature. These data were comparable with the data presented.

The line representing the data in the viscous region approaches a limiting horizontal position indicating only a slight change of the heat transfer coefficients with variation of the Reynolds number. The Nusselt-Prandtl ratio, therefore, is substantially independent of the Reynolds number in this region.

The Dittus and Boelter equation represents a correlation for turbulent flow conditions. It would not, therefore, be expected to correlate data in the viscous region. This has been pointed out by other investigators. A further substantiation is provided by the new data presented.

Further correlation was attempted plotting the Reynolds number vs. the Nusselt number. This correlation, Fig. 7, gave curves similar to those obtained using the Dittus and Boelter co-ordinates. The data plotted with hD/k as ordinate and Reynolds number as abscissae are slightly more scattered, but the slope of the curve was substantially the same.

A modified friction factor curve, Fig. 8, was used in the correlation of the friction factor and heat transfer coefficient. A plot of the isothermal friction factor vs. the

$$\frac{hD}{K}$$

100.

15.

150.

1,000.

$$\frac{DG}{\mu}$$

10,000.

Fig. 7 - Correlation of Reynolds Number vs Nusselt Number for Eocene Data.

Fig. 8 - Friction Factor vs Reynolds Number.

Nusselt group, Figs. 9-11, gave excellent correlations for both heating and cooling. In the turbulent region, Fig. 12, the heating and cooling coefficients were brought together. In the viscous region separate correlations for heating and cooling were obtained. The data through the critical region are well defined.

The correlation of Keevil and McAdams⁽¹⁶⁾ for the effect of heating and cooling upon the friction factor has been employed. The relation between the actual and isothermal friction factors is dependent upon the temperature gradient. The relation of the friction factors for heating is given by the equation:

$$F/f_h = 1 - .00317 (t_w - t)$$

and for cooling:

$$F/f_o = 1 + .00738 (t - t_w)$$

where $(t_w - t)$ is the temperature drop from tube wall to main-stream fluid conditions.

The data in the turbulent region fell in a well-defined straight line. Considering the fact that the value of the friction factor is obtained from the Reynolds number, the data are represented by a line equally well in the plot of hD/k vs. f or hD/k vs. DG/μ . If the data are considered as representing an area, the accuracy of the plots are comparable but slightly in favor of the friction factor-Nusselt co-ordinates.

Fig. 9 - Cooling Correlation of Friction Factor vs Nusselt Number for Eocene.

Fig. 10. - Cooling Correlation of Friction Factor vs Nusselt Number for Oils.

Fig. 11 - Heating Correlation of Friction Factor vs Nusselt Number for Oils.

Sherwood, Kiley and Mangsen
x Oil - Length 11.59' - Heating

Sherwood and Petrie
† Kerosene - Heating
J.D. Smith

o Oil - Heating
o Oil - Cooling

Morris and Whitman
□ Gas Oil - Heating
■ Gas Oil - Cooling
△ Straw Oil - Cooling

Barnes, Groth, and Scott
• Kerosene - Cooling

Fig. 12. - Correlation of Heating and Cooling Coefficients.

The character of the data in the viscous region would indicate that the Prandtl group which includes the properties of the liquid cannot be neglected. The Sieder and Tate correlation in the viscous region when the viscosity ratio is approximately unity becomes equivalent to the Dittus and Boelter correlation which has been shown to be inadequate. A plot of the friction factor or Reynolds number vs. the Nusselt number provides a fair correlation for similar types of liquids in the viscous region and is excellent in the turbulent region.

The study of this and other experimenters' data shows that for heating and cooling the correlation of Dittus and Boelter and Sieder and Tate is adequate in the turbulent and upper critical region.

A further study in the viscous region for both heating and cooling can well be continued.

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