University of Louisville

ThinkIR: The University of Louisville's Institutional Repository

Electronic Theses and Dissertations

1936

Heat transfer coefficients.

R. Y. Browne

University of Louisville

Follow this and additional works at: <https://ir.library.louisville.edu/etd>

Part of the [Chemical Engineering Commons](#)

Recommended Citation

Browne, R. Y., "Heat transfer coefficients." (1936). *Electronic Theses and Dissertations*. Paper 1702.
<https://doi.org/10.18297/etd/1702>

This Master's Thesis is brought to you for free and open access by ThinkIR: The University of Louisville's Institutional Repository. It has been accepted for inclusion in Electronic Theses and Dissertations by an authorized administrator of ThinkIR: The University of Louisville's Institutional Repository. This title appears here courtesy of the author, who has retained all other copyrights. For more information, please contact thinkir@louisville.edu.

University of Louisville

Heat Transfer Coefficients

A Dissertation

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 Science

in

Chemical Engineering

Department of Chemical Engineering

by

R. Y. Browne

1936

TABLE OF CONTENTS

	page
List of Symbols	1
List of Illustrations	3
List of Tables	3
Acknowledgement	4
I. Introduction	5
II. Historical	8
III. Theoretical	15
IV. Apparatus	24
A. Heat Exchanger	25
B. Accessories	34
C. Operation	38
V. Procedure	41
VI. Data and Results	44
VII. Conclusions	51
VIII. Bibliography	54

LIST OF SYMBOLS

Symbol	Quantity	Units
A	Heat transfer surface	sq. ft.
C _p	Specific heat	B.t.u./ (lb.) (deg.F.)
D	Diameter of pipe	ft.
f	Friction factor	none
g	Acceleration due to gravity	ft./hr. ²
G	Mass velocity	lb./ (hr.) (sq.ft.)
h	Heat transfer coefficient	B.t.u./ (hr.) (sq.ft.) (deg.F.)
k	Thermal Conductivity	B.t.u./ (hr.) (sq.ft.) (deg.F./ft.)
L	Length of path of heat conduction	ft.
N	Length of test section	ft.
dN	Differential length of pipe	ft.
dP	Differential pressure drop due to friction	lb./sq.ft.
q	Quantity of heat transferred	B.t.u./hr.
t ₁	Fluid entrance temperature	deg. F.
t ₂	Fluid exit temperature	deg. F.
t ₃	Refrigerant entrance temperature	deg. F.
t ₄	Refrigerant exit temperature	deg. F.
V	Velocity of fluid in pipe	ft./hr.
w	Weight rate of flow	lb./hr.

LIST OF SYMBOLS (cont.)

Δt	Temperature difference	deg./F.
ρ	Density of fluid	lb./cu.ft.
μ	Viscosity of fluid	lb./{(hr.)(ft.)}

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

LIST OF ILLUSTRATIONS

	Page
Plate 1. Complete assembly of apparatus	26
Plate 2. Assembly of reservoir and pumping units	35
Plate 3. Mixing chamber construction and assembly	31
Plate 4. Split sleeves and inlet member	29
Plate 5. Dismantled view of outlet section	33
Plate 6. Thermocouple measuring instruments	29
Figure 1. Thermocouple switch board connections	39

LIST OF TABLES

	Page
Table I. Cooling Data	46
Table II. Physical Properties of Acetone	48

ACKNOWLEDGEMENT

The author of this thesis
wishes to acknowledge
the kind aid and helpful suggestions of
Dr. R. C. Ernst,
who directed this research.

I. INTRODUCTION

This investigation consisted of the construction, with modifications, of a heat transfer apparatus designed by Browne and Finger.⁽¹⁾ This heat exchanger was built in order to determine heat transfer coefficients under varying conditions of temperature and velocity and with different fluids and metals. The present work covers only the construction, calibration and a preliminary investigation to determine the operating characteristics. Almost one year was required for the construction of the equipment and the experimental work covers only the determination of coefficients under several conditions of velocity.

The object of this research was primarily to construct the equipment and to study the effect of temperature difference and velocity on the coefficients determined. It also included the correlation of data in such a manner as to make possible comparisons with the data of other investigators in heat transfer work. The correlation of coefficients is essential in order that the effect of changed conditions can be predicated with reasonable assurance.

The apparatus used was designed to obtain heat transfer data under the conditions of viscous and turbulent flow. Data was desired mainly for the upper regions of

viscous flow and the lower regions of turbulent flow in order to determine the abnormal effects in the critical region. Desirable data in the lower regions of viscous flow are not yet available for a complete correlation in this range. (12).

Film coefficients of heat transfer between liquids at low temperatures (32 to --40 deg.F.) have not been investigated to any noteworthy extent, and a thorough knowledge of coefficients at these temperatures would be useful in the design of certain types of industrial equipment utilizing low temperatures. Such design would necessitate the calculation of cooling area required in coolers, refrigerating units and similar apparatus, the minimum area depending to a great extent upon the magnitude of film coefficient.

The proper utilization of adequate heat transfer data would result in better and more efficient design and would tend to eliminate the unreliable methods which are commonly employed in the design of such equipment.

II. HISTORICAL

The determination of liquid film coefficients has received considerable attention by a great number of research workers interested in heat transfer. The majority of investigators have borne out the fact that a stagnant or stationary film is the controlling factor in the transfer of heat between liquids through a separating wall of high thermal conductivity. This stationary, or slowly moving film as considered by some investigators, varies in two ways; composition and magnitude.⁽¹²⁾ Its composition depends naturally on the liquid under test. Its magnitude is dependent upon the conditions of viscosity, temperature, diameter of pipe through which the liquid is flowing, density and mass velocity of the liquid. The effect of the film composition lies in its thermal conductivity, which is a property of the liquid itself, and varies only with temperature. The effect of the other factors on heat transfer through the film is widely varied since any one or combination of factors produces its own effect.

The affects of these various factors are most easily observed by studying their individual properties when equated in certain dimensionless groups. Such groups are the Reynolds number (DG/μ), the Prandtl number ($C\mu/k$), the Nusselt number (hD/k), and the Graetz number (wC/kN).

The comparison and plotting of certain of these dimensionless groups give an excellent method for correlating forced convection heat transfer data. Up to the present time, much work has been done on forced convection data and its correlation by this method.

Working with hydrocarbon oils in turbulent flow inside smooth pipes, Smith ⁽¹⁸⁾ found that the data could be correlated by plotting $(hD/k)/(\mu/k)^n$ against (DG/μ) , where $n = 0.4$ for heating runs and 0.3 for cooling runs. This is similar to the equation previously used by Morris and Whitman ⁽¹⁴⁾ in which (DG/μ) is plotted against $(hD/k)/(\mu/k)^{0.37}$ where μ is the viscosity taken at the average main body conditions. Their data was obtained from tests on oils and water inside pipes. The equation above coincides with that used by Sherwood and Petrie ⁽¹⁷⁾ who worked with various liquids inside pipes in turbulent and viscous flow. They used the Nusselt type of equation, $(hD/k) = .024(DG/\mu)^{.8}(C\mu/k)^{.4}$ for heating data and $(hD/k) = .026(DG/\mu)^{.8}(C\mu/k)^{.3}$ for cooling data, which was derived from the data of Morris and Whitman by Dittus and Boelter. The constants of the equation were varied slightly for heating and cooling in order to bring the data into better correlation.

McAdams and Frost (13) studied the data of several investigators on heating water inside tubes of varying tube length where (N/D) varied from 34 to 100. They found an effect produced and set up the equation $(hD/k) = .0272(1 + 50D/N)(DG/\mu)^{.8}$. In this case, μ is the viscosity at the mean film temperature which can be only estimated at best. Lawrence and Sherwood (10) reported data on heating water in four lengths of 0.593" I.D. copper pipes. The observed effect of the tube length was negligible, although (N/D) varied from 59 to 224.

In the region of viscous flow, sufficient dependable data are not yet available to warrant a general equation of all the factors which may be involved. However, for heating hydrocarbon oils at (DG/μ) below 2100, data may be roughly correlated by plotting (hD/k) against (w^C/kN) . The data of Sherwood, Kiley and Mangsen (16), who studied the effect of tube length on heating oils in viscous flow inside pipes, show no substantial effect of pipe length for values of (N/D) from 122 to 234. Drew, Hogan and McAdams (7) in correlating viscous flow data, plotted $(t_2 - t_1) / (t - t_1)$, (actual rise in temperature)/(temperature difference between wall and fluid) against both (w^C/kN) and (DG/μ) . A comparison of all data showed no correlation of rise divided by

temperature difference in terms of the Reynolds number which involves viscosity. This showed that for viscous flow, the use of the coordinate (w^C/kN) was preferable to (DG/μ) . However, Drew ⁽⁶⁾ analyzed the data of Rynalski and Huntington who worked with glycerine in viscous flow and plotted temperature rise divided by initial temperature difference against (w^C/kN) . He found that runs with an initial temperature difference of 65 deg. C. fell on a curve substantially higher than those with a temperature difference of 30 deg. C. This indicates that other factors may be necessary to bring the data into better correlation.

The relation of heat transfer data to pressure drop has been studied by Chilton and Genereaux ⁽²⁾ and Keevil and McAdams ⁽⁸⁾. A. P. Colburn ⁽³⁾ has carried out extensive work on the correlation of data by dimensionless groups. Keevil and McAdams worked with oil inside a straight pipe and found that friction factors were higher for heating runs and lower for cooling runs than those of isothermal runs. Previously, those men investigating friction losses made all their tests under isothermal conditions. This meant that the main body viscosity was also the film viscosity and it was not necessary to find a relationship between them. But since heat transfer tests cannot be made under isothermal conditions, it

is desirable that some correlation be made between film temperature and viscosity and main body temperature and viscosity. Keevil and McAdams found that the non-isothermal data could be brought into agreement with isothermal data by the following:

$$\text{for heating,} \quad t_{\text{corrected}} = t_{\text{ave.}} + 0.20 \Delta t$$

$$\text{for cooling,} \quad t_{\text{corrected}} = t_{\text{ave.}} + 0.32 \Delta t$$

where $t_{\text{ave.}}$ is the arithmetic mean between the inlet and outlet fluid temperature and Δt is the logarithmic temperature difference between fluid and pipe wall.

They first plotted their non-isothermal data as friction factor versus Reynolds number in which the viscosity was taken at the average fluid temperature. A series of curves were obtained for various temperature differences which were similar in shape to that for isothermal flow. However, since the friction factors are lower for heating and higher for cooling, it is obvious that the film viscosity is the controlling factor and this depends on the film temperature which is a function of the temperature difference between the pipe and fluid. This tends to explain why the curves obtained were normal when plotted for each temperature difference.

Work on the relation of coefficients to temperature

difference was carried out by Linden and Montillon (11).

They worked with water in an inclined tube evaporator and found that the coefficients increased with the temperature difference. Further study along these lines would be desirable, especially in the relationship between temperature difference, film temperature or viscosity, and friction.

III. THEORETICAL

In any heat transfer process, the heat may flow by three methods: conduction, convection, and radiation. Conduction involves the transfer of the kinetic energy of the molecules from one layer of molecules to another. The Kinetic energy transferred is that which is in excess of that of an adjacent layer of molecules. Convection is the flow of heat caused by the transference of the molecules themselves. In heat transfer by radiation, the molecules give rise to radiant energy, in an amount determined by their temperature, which is capable of passage to a distant receiver. In the problem under discussion the chief concern is the transfer of heat by conduction and convection.

In the transfer of heat from one liquid to another through a separating wall, the present day accepted theory is that the heat flows from the main body of one liquid to a liquid film on the pipe wall by convection of the liquid. The heat then flows through this film and the separating wall by conduction, through a liquid film on the other side of the wall by conduction and is then dissipated from the surface of the film by convection in the other liquid. This means that along both surfaces of the pipe there is a thin film of liquid which is relatively stationary with respect to the main body velocity.

The rate at which heat is transferred from one liquid to another depends therefore on several factors: the magnitude of the liquid convection, and the resistance of the liquid films and separating wall to the flow of heat. The Fourier equation for the flow of heat in the steady state, in which the temperature does not vary with the time is

$$q = \frac{k A \Delta t}{L} \quad (1)$$

in which (q) is the rate of heat flow, (k) the thermal conductivity of the material through which the heat is flowing, (A) the area at right angles to the direction of flow, (L) the distance and (Δt) the change of temperature over the distance. The factor ($\frac{k}{L}$) is called the coefficient of heat transfer. However, since the thermal conductivity and thickness of the two films and pipe wall will differ from each other, it is necessary to break this coefficient up into its several parts.

Comparing the resistance to heat flow with that to electrical flow in which

$$\text{rate of flow} = \frac{\text{driving force}}{\text{resistance}}$$

$\frac{L}{kA}$ corresponds to the resistance to heat flow. But this re-

sistance is made up of the resistance of the films and pipe wall, and it follows that

$$\frac{L}{kA} = \frac{L_1}{k_1 A_1} + \frac{L_2}{k_2 A_2} + \frac{L_3}{k_3 A_3} \quad (2)$$

Referring them all to the same area,

$$\frac{L}{k} = \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} \quad (3)$$

Substituting this in the original equation,

$$q = \frac{A \Delta t}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}} \quad (4)$$

Since the thickness of the films cannot be measured, the magnitude of the film resistance ($\frac{k}{L}$) can best be represented by a single value (h) which is called the film coefficient. Therefore the equation takes the form

$$q = \frac{A \Delta t}{\frac{1}{h_1} + \frac{L_2}{K_2} + \frac{1}{h_3}} \quad (5)$$

If the total resistance of the films and pipe wall are combined into an overall coefficient (U), then

$$q = U A \Delta t \quad (6)$$

Referring to equation (5), the thermal conductivity of most

metals is much greater than that for liquids, and the term $(\frac{L}{k})$ is so small in comparison with the others that it is usually neglected. It therefore follows that the resistance of the films is the controlling factor for the transfer of heat between two liquids, and any conditions which will reduce the resistance of these films will result in greater heat transfer.

Since $h = \frac{k}{L}$, (h) can be increased by an increase in (k) , the thermal conductivity, or a decrease in (L) , the thickness of the film. However, (k) is dependent upon the type of fluid and varies only slightly with temperature. Therefore, assuming (k) constant for a given fluid, (h) will depend on the thickness.

The factors which affect the thickness of the films are numerous, the foremost being the mass velocity of the fluid, which is the product of the linear velocity and the density. When a fluid is flowing through a pipe, there is a shearing force between the fluid and the stagnant or slowly moving film. The thickness of the film will depend upon the magnitude of this shearing force. The greater the mass velocity of the fluid, the greater will be the shearing force produced, and therefore the thickness of the film will be decreased.

Another factor which affects the thickness is the viscosity of the fluid. A fluid of high viscosity will result in the formation of a thick or very viscous film on the wall, whereas a less viscous fluid will form a much thinner film. Film viscosity depends on the film temperature. If the temperature of the film is different from the temperature of the fluid, then the viscosity will not be the same. Since there is always a temperature gradient between the pipe wall and fluid, it is therefore the viscosity of the film that is controlling and this viscosity must be taken at the film temperature.

Still another factor which affects the film thickness is the diameter of the pipe through which the fluid is flowing. When the diameter is large, the less effective will be the mass velocity of the fluid on the film because the ratio of surface to unit volume of fluid is decreased. However, when the pipe is small, this ratio is correspondingly increased. Therefore, it follows that the larger the diameter, the smaller will be the film coefficient.

In a term called the Reynolds number (DG/μ), all these factors are present. The type of flow of the fluid, viscous or turbulent depends on the value of the Reynolds number. Below a value of 2100 for (DG/μ), the flow is

viscous and above this value, the flow is turbulent. From the foregoing discussion, it is obvious that greater heat flow would result when the flow is turbulent instead of viscous. This has been found as such by the majority of investigators.

From Chapter II, it is noted that heat transfer data could be correlated by the use of certain dimensionless groups, of which the Reynolds number appears in nearly all correlations. However, the Reynolds number was previously confined to the study of the flow of fluids in which the friction of fluids inside pipes bore a definite relation to the Reynolds number. In the data of early investigators for isothermal flow, friction factors were plotted against (DG/μ) , the friction factor (f) being calculated from the Fanning equation-

$$-\frac{dP}{dN} = \frac{4 f \rho V^2}{2 g D}$$

With the advent of heat transfer data, it was later discovered that heat transfer had a marked affect on fluid friction. As brought out in Chapter II, the plotting of friction factor (f) against (DG/μ) resulted in curves which were similar to the isothermal curve but somewhat displaced, depending on the temperature difference between the fluid and pipe wall. The curves were similar only when the data for

any particular temperature difference were plotted, a separate curve being obtained from data for each temperature difference. As previously stated, the film viscosity is one of the controlling factors and depends on the film temperature. However, in the calculation of the friction factor, the viscosity is not included. Therefore a relationship between film viscosity and friction factor is evidently necessary to bring the isothermal and heat transfer data into agreement. However, the discrepancy may lie in the determination of the film temperature which can only be estimated at best. An accurate method of determining film temperatures would probably throw much light on the problem of friction and heat transfer.

Referring to equation (6), it is obvious that the film coefficient can be calculated from the overall coefficient (U) since

$$U = \frac{1}{\frac{1}{h_1} + \frac{L_2}{K_2} + \frac{1}{h_3}}$$

However, since the heat that passes through one film must also pass through the pipe wall and the other film, the coefficient can be calculated from the formula

$$q = h A \Delta t_m$$

where (A) is the surface area next to the film and (Δt_m) is the logarithmic mean temperature difference between the fluid inside the pipe and the pipe wall.

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}}$$

where Δt_1 = difference in temperature between the fluid and pipe at one end of the apparatus,

Δt_2 = difference in temperature between fluid and pipe at other end of apparatus

Δt_1 and Δt_2 are chosen so that Δt_1 is always the larger.

In the apparatus used, the temperature of pipe and inlet and outlet temperature t_1 and t_2 are measured. The area of the pipe is known and the heat transferred can be calculated from the temperature difference ($t_1 - t_2$), specific heat of the fluid, and the amount of fluid flowing.

IV. APPARATUS

In designing an apparatus for the determination of liquid film coefficients of heat transfer, it is necessary to provide for the determination of the quantity of heat transferred per unit time from one liquid to another through a separating wall of high thermal conductivity. Before determining the quantity of heat transferred the variable factors such as the entrance and exit temperatures of test liquid and refrigerant, the temperature of the pipe wall, and the velocity of flow must be measured. An apparatus for determining these quantities consists of the heat exchanger proper, which includes the test section, mixing chambers, calming sections, thermocouples, and manometers, and various accessories such as test liquid and refrigerant reservoirs and pumps, weighing tanks and thermocouple circuits. The apparatus used was that designed by Browne and Finger ⁽¹⁾ with a few modifications. The complete apparatus is shown in plate 1, page 27.

The various parts are described in the following discussion.

A. Heat Exchanger.

1. Test Section.

The test section consists of a standard one inch

Plate 1. Complete assembly of apparatus.

brass pipe enclosed in a cast iron shell. The brass pipe is a single length of 9 feet, 6 inches which includes a calming section of 23 inches, a test length of 6 feet and an exit section of 19 inches. The cast iron shell consist of the refrigerant inlet and outlet members C-2 and C-1, which are identical (Plate 4, Page 29). Also a number of split sleeves, D-1, D-2, F-1 and F-2, for making thermocouple connections to the pipe wall at each end and middle of the test section, and two 30 inch sleeves E-1 and E-2 (Plate 1, Page 27) make up the main portion of the shell. The split sleeve construction allows the inspection of the thermocouple connections without complete dismantling of the exchanger. This construction also permits the length of the test section to be varied by inserting different lengths of sleeves in place of those used.

The refrigerant inlet and outlet members include calming sections to prevent the existence of turbulent motion at the ends of the test section where the temperatures of the pipe wall were taken. Since the temperatures of the test liquid and refrigerant were taken at a point several feet from the end of the test section, an insulation space packed with rock wool was provided between the one inch brass pipe and $2\frac{1}{2}$ " steel pipe (see Plate 5, Page 33) to prevent the flow of

Plate 4. Split sleeves and inlet member

Plate 6. Thermocouple measuring instruments

heat from the test liquid to the refrigerant.

Conduction leaks between the brass pipe and cast iron shell were minimized by inserting thin asbestos strips between the parts where contact occurred. However, the area of contact was very small and occurred at a single position at each end of the test pipe and at a considerable distance from the test section.

2. Mixing Chambers.

The function of the mixing chambers was to thoroughly mix both the test liquid and refrigerant so that the temperatures of the liquids would be constant throughout their cross-section. The mixing was accomplished by bolting together a series of slotted copper plates and steel collars through which flowed the liquids before entering and leaving the test section. (See Plate 3, Page 31). Provisions were made in the chambers for introducing thermocouples into the mixed liquids.

3. Calming Section.

The calming section is located in the test pipe at the fluid entrance end of the exchanger. A calming cross was placed in this section to eliminate the rotary motion or turbulence imparted to the fluid by the mixing chamber at that

Plate 3. Mixing chamber construction and assembly.

point. The cross was constructed from sheet copper and consists of two interlocking plates.

4. Thermocouples.

The thermocouples were made from 28 B&S gage iron and constantan wire. This gage was found by Smith ⁽¹⁷⁾ to give best results in fluid streams. The wires were welded together at one end by an electric arc method. They were standardized in a vacuum bottle over a temperature range of 70 to - 40 deg.F. by a dry ice - acetone solution. The couples were calibrated to 0.2 deg F accuracy by a sensitive galvanometer and potentiometer circuit (Plate 6, Page 29).

The thermocouples in the mixing chambers were installed by inserting a 1/8" copper tube, in which the thermocouple wires were incased, through a 1/8" diameter hole drilled in a 1/2 inch bolt. The tube projected through the collar of the mixing chamber to the center of the liquid stream.

A group of four thermocouples, equally spaced around the test pipe were imbeded in the wall of the pipe at each end and the middle of the test section. (Plate 5, Page 33). A slot 1" long and 1/16" deep was made in the pipe for each thermocouple. The thermocouple junction was soldered to the pipe at one end of the slot and the two leads from

Plate 5. Dismantled view of outlet section

the junction were cemented in the slot by litharge and glycerine cement. The surface was then dressed down smooth with the pipe surface. This method of connection minimized a localized heating or cooling effect at the junction due to conduction along the wires from the test liquid.

5. Manometers.

The manometer M-1 (Plate 1, Page 27) was used to measure the pressure drop of the test liquid across the test pipe for the determination of fluid friction data. The manometer M-2 was used to measure the pressure drop across the mixing chamber M.C.-2. The amount of heat gained by the test liquid due to friction in the pipe and mixing chamber is in excess of that flowing through the tube wall, and must be deducted from the total amount of heat calculated from the temperature difference ($t_1 - t_2$). The manometers are mercury in glass and were inclined to give a large reading. They were connected to the mixing chambers by 1/8 inch copper tubing.

B. Accessories.

1. Reservoirs.

The reservoir T-1 for the test liquid is a horizontal cylindrical steel drum of 10 gallons capacity, with outlet at one end of the tank near the bottom and inlet at the top. A by-pass line around the tank was provided to regulate

Plate 2. Assembly of reservoir and pumping units.

the temperature of the test liquid to the exchanger.

The refrigerant reservoir T-2 is a vertical cylindrical drum of 20 gallons capacity. This tank has a removable top to permit the addition of dry ice, which was used as a cooling agent for the refrigerant. This tank was not provided with a by-pass line, but the temperature was regulated by a mechanical refrigerating unit inserted in the tank. The outlet connection was made at one side near the bottom. The outlets of both tanks were of sufficient distance from the bottom to prevent any sediment from being carried to the pumps.

2. Pumps.

The pumps P-1 and P-2 used to circulate the test liquid and refrigerant are identical in construction and capacity. They are bronze gear pumps with a capacity of 6 gallons per minute at 1,000 R.P.M. The two pumps were connected to a single motor by V belts. The pumps were run at a constant speed, and the velocity of both fluids was regulated by by-pass lines which recirculated a portion of the liquids through the pumps.

3. Weighing Tanks.

At a point in each line, a two way valve V-7 or

V-8 was inserted for directing the flow of liquid down to a weighing tank T-3. The object of the weighing tank was to give an absolute method of determining the quantity of the liquids flowing through the exchanger, from which the velocity may be calculated.

The weighing tanks consist merely of open vessels large enough to accommodate a reasonable quantity of liquid collected over a short interval of time.

4. Thermocouple Switch Board.

The thermocouple switch board S-1 was designed so that any one thermocouple could be easily connected to the potentiometer for measuring the E.M.F. produced. The iron and constantan wires of each thermocouple were soldered to copper leads at the cold junction of 0 deg. C. The copper leads from the iron wire and those from the constantan were run to the posts of dials 1 and 2 respectively (Figure 1, Page 39). Copper leads from the potentiometer were connected to each dial arm. The thermocouples were read by turning the two dial arms to the pair of posts corresponding to each thermocouple.

5. Insulation.

The heat exchanger and refrigerant reservoir must be covered with 85% magnesia covering to prevent heat transfer to

the apparatus from the atmosphere. The refrigerant pipes must also be covered with regular pipe lagging.

C. Operation of Apparatus.

The test liquid flows from the reservoir T-1 (see Plate 1, Page 27) through the pump P-1 and its velocity regulated by a by-pass valve V-2. It then flows into the mixing chamber M.C. 1 and its temperature is determined by thermocouple #1. The liquid then passes through the calming section and into the test section where the transfer of heat occurs.

The test liquid is again thoroughly mixed in mixing chamber M.C.2 and its temperature determined by thermocouple #2. It then flows back to the reservoir, or is bypassed directly to the pump by valve V-6 depending on the inlet temperature desired.

Likewise, the refrigerant flows from the reservoir T-2, through the refrigerant pump P-2 and the velocity is regulated by valve V-4. It then flows into the test liquid exit end of the exchanger through the mixing chamber M.C.3 where the temperature is taken by thermocouple #3. The refrigerant then passes through the annular space surrounding the test pipe, counter-current to the flow of the test liquid. It then flows out of the exchanger through mixing chamber

FIGURE 1 THERMOCOUPLE
SWITCH BOARD CONNECTIONS

M.C. 4 where the temperature is again taken by thermocouple #4. The refrigerant is then returned to the reservoir for recirculation.

V. PROCEDURE

In making test runs on the apparatus, a definite procedure was followed. The data recorded consisted of thermocouple readings and the amount of test liquid flowing through the exchanger.

In making a run, the rate of flow of the test liquid and refrigerant was regulated by valves V-2 and V-4. The setting of the valves was not changed and the rate of flow during a run was regarded as constant. The apparatus was then operated until equilibrium conditions were obtained. These conditions were reached when the entrance temperatures of test liquid and refrigerant became constant. The time required for reaching equilibrium varied from $1\frac{1}{2}$ to 3 hours depending on the velocity of the test liquid in the exchanger. All of the thermocouple readings were then recorded and the rate of flow determined. This was done by weighing an amount of liquid collected in the weighing tank over a short period of time. In making the runs it was found that the rate of flow could not be correctly determined by use of the three way valve V-7. A difference in head between the test section and valve V-7 prevented the flow from being the same at both points. This necessitated the collection of a portion of the liquid at a point after it had passed through the test section in determining the rate of flow.

After making the first few runs, it was also discovered that the manometers M-1 and M-2 could not be read. This was due to a corrosive action of the acetone on the mercury in the manometers. However, the pressure drops obtained in the first few runs were small enough to be neglected. This was probably due to the low viscosity of the acetone and the low velocities used.

VI. DATA AND RESULTS

A tabulation of the data and results is given in table I.

In calculating (h), the arithmetic mean temperature difference (Δt_m) of pipe and liquid was used. The accuracy of the arithmetic mean was well within that of the data. The arithmetic mean temperature difference is the difference between the average pipe temperature and the average liquid temperature. The average pipe temperature was taken as the average of the readings of all thermocouples connected to the pipe.

The diameter of the brass pipe is taken as the inside diameter. In making all calculations, the properties of the liquid were taken as constant. This was justified since a very small temperature range was used. The properties of the liquid are listed in table II. The acetone used was of technical grade and contained considerable methanol. The density was measured with a Westphal balance and the viscosity was determined with an Ostwald viscometer. The specific heat and thermal conductivity were estimated by taking into consideration the other properties and the composition of the liquid.

Table I

Run No.	Liquid Entrance T.C. 1	Liquid Exit T.C. 2	$t_1 - t_2$	Ave. Pipe Temp.	Mean Temp. Diff.
Units	deg. F.	Deg. F.	Deg. F.	Deg. F.	Deg. F.
1	91.6	86.4	5.2	79.3	8.37
2	92.8	80.8	12.0	78.3	8.45
3	97.0	88.3	8.7	81.9	12.04
4	97.7	90.4	7.3	83.1	12.04
5	93.7	91.4	2.3	79.5	6.93
6	92.8	91.5	1.3	87.0	5.05
7	100.0	91.7	8.3	85.1	10.80
8	97.6	90.6	7.0	84.0	10.07
9	100.5	88.5	12.0	85.5	9.06
10	101.0	93.4	7.6	87.2	10.50

Table I (Cont.)

Run No.	Liquid Velocity	Weight of Liquid Flowing	DG/ μ	h	$(hD/k)/(C\mu/k)^{0.3}$
Units	Ft./sec.	lb./min.	none	Btu/hr./sq. ft./deg.F.	none
1	0.223	4.19	3690	89.3	44.8
2	0.078	1.47	1292	71.5	35.9
3	0.113	2.32	1950	55.0	27.6
4	0.127	2.39	2100	49.3	24.8
5	0.453	3.85	7480	97.7	49.0
6	0.820	15.37	13530	131.0	65.8
7	0.173	3.25	2860	85.3	42.8
8	0.114	2.13	1380	50.8	25.5
9	0.080	1.50	1320	67.5	33.8
10	0.160	3.00	2640	73.7	37.0

Table II

Properties of Acetone at 36 deg. F.

Density	52.04 lb./cu.ft.
Viscosity	0.41 centipoises
Specific Heat	0.521 B.t.u./ (lb.) (deg.F.)
Thermal Conductivity	0.109 B.t.u./ (hr.) (sq.ft.) - (deg.F./ft.)

The amount of heat transferred (q) was calculated from the rate of flow (w), the temperature difference ($t_1 - t_2$) and the specific heat (C). The film coefficient (h) was calculated by dividing the amount of heat transferred by the inside surface area of the pipe and the mean temperature difference. The Reynolds number (DG/μ) and the velocity (V) were calculated directly from the rate of liquid flow.

An attempt was made to correlate the data by plotting (DG/μ) against $(hD/k)^3 / (C\mu/k)^3$. A plot of (WC/kN) against $(t_1 - t_2)/(t_1 - t_2)$ gave no correlation and is not included. The resulting curve is shown on page 49. The dip in the curve occurs at a Reynolds number of 2100. The lower curve is that from the data of Sherwood and Petrie for heating acetone. In the turbulent region, the curves are somewhat similar, but considerably displaced. This was due to their use of a smaller pipe, the diameter being 0.494 inches.

A decrease in D of $\frac{1}{2}$ gives a corresponding decrease in the ordinate $(hD/k)/(C\mu/k)^{0.3}$ for a given Reynolds number. The rise of runs 2 and 9 in the viscous region was unexpected and is contrary to theory. However, it is possible that these runs were in error. A more logical deduction can be obtained from the apparatus. The temperature of the refrigerant in the exchanger was always one or two degrees within room temperature, and the entrance temperature of the liquid about ten degrees above. The latter temperature was taken about two feet in advance of the test section and the exit temperature about $1\frac{1}{2}$ feet past this section. Since the exchanger was not insulated, it is possible that considerable cooling of the test liquid was caused by heat lost to the atmosphere in these sections. On this basis, the heat transferred between the two liquids calculated from the temperature difference $(t_1 - t_2)$ would be considerably in excess of that actually transferred in the exchanger. This would result in a high value for the coefficient and likewise for the ordinate $(hD/k)/(C\mu/k)^{0.3}$ on the curve. However, this error would be appreciable only in the case of low liquid velocities as is borne out by the curve.

VII. CONCLUSIONS

The results show that the present data is in agreement with other reported data in the turbulent region. The data in the viscous region is insufficient for accurate comparison. It is obvious that the exchanger must be sufficiently insulated before viscous data will be of any real significance.

Because of the small temperature difference obtained with the apparatus, the effect of the change in the liquid properties with temperature could not be shown. Any correlation which may be indicated by the results is therefore only concerned with the velocity of flow and the film coefficient. The properties of the liquid were included in the correlation since they are generally employed in heat transfer work of wider temperature ranges.

Although the primary purpose of doing low temperature work was not possible in this portion of the research, much was gained by the preliminary investigation and testing of the apparatus. Further investigation at the temperature level used would probably be of considerable value in preparation for work at lower temperatures.

The author is grateful
for the assistance of Mr. Hubert Donahew
in conducting this research.

VIII. BIBLIOGRAPHY

1. Browne, R. Y., and Finger, F. W. Jr.; Thesis in
Chemical Engineering, University of Louisville (1935).
2. Chilton, T. H., and Genereaux, R. P.; Trans. Amer.
Inst. Chem. Eng., 29, 161 (1933).
3. Colburn, A. P.; Trans. Amer. Inst. Chem. Eng., 29,
174 (1933).
4. Colburn, A. P., and Hougen, O. A.; Ind. Eng. Chem.,
22, 522 (1930).
5. Cryder, D. S., and Gilliland, E. R.; Ind. Eng. Chem.,
24, 1382 (1932).
6. Drew, T. B.; Ind. Eng. Chem., 24, 152 (1932).
7. Drew, T. B., Hogan, J. J., and McAdams, W. H.; Ind.
Eng. Chem., 23, 936 (1931).
8. Keevil, C. S., and McAdams, W. H.; Chem. and Met.,
36, 464 (1929).
9. Kirkbride, C. S., and McCabe, W. L.; Ind. Eng. Chem.,
23, 625 (1931).
10. Lawrence, A. E., and Sherwood, T. K.; Ind. Eng. Chem.,
23, 301 (1931).
11. Linden, C. M., and Montillon, G. H.; Ind. Eng. Chem.,
22, 708 (1930).
12. McAdams, W. H.; "Heat Transmission", McGraw Hill
Book Company (1933).

13. McAdams, W. H., and Frost; J. Amer. Soc. Refrigerating Eng., 10, 323 (1924).
14. Morris, F. H., and Whitman, W. G.; Ind. Eng. Chem., 20, 234, (1928).
15. Murphree, E. V.; Ind. Eng. Chem., 24, 726 (1932).
16. Sherwood, T. K., and Kiley, D. D., and Mangsen, G. E.; Ind. Eng. Chem., 24, 273 (1932).
17. Sherwood, T. K., and Petrie, J. M.; Ind. Eng. Chem., 24, 736, (1932).
18. Smith, J. F. D.; Trans. Amer. Inst. Chem. Eng.; 31, 83 (1935).