1969

A study of low-cost heat exchangers for desalination

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A STUDY OF LOW-COST HEAT EXCHANGERS FOR DESALINATION

BY 541

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1938

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THESIS

submitted to the faculty of

THE UNIVERSITY OF MISSOURI - ROLLA

171256

in partial fulfillment of the requirements for the

Degree of

MASTER OF SCIENCE IN CHEMICAL ENGINEERING

Rolla, Missouri

1969

Approved by

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ABSTRACT

This investigation was proposed to search for simple and inexpensive but effective heat exchangers for the recovery of heat in water desalination by evaporation through non-wettable porous membranes.

Aluminum was chosen as the material for constructing the heat exchangers. Various fluted arrangements and one plate type arrangement were chosen to increase heat transfer coefficients and conserve material. Equations correlating the heat transfer coefficients to temperatures and flow rates have been suggested for estimating the overall heat transfer coefficient at different conditions.

The results of economic calculations indicate that the cost per unit transfer area of the exchangers in this investigation may eventually be reduced to a range of \$ 0.60/ft^2 to \$ 0.36/ft^2 depending on the type of equipment.

The most promising results were obtained with aluminum plates separated by aluminum screens which gave overall heat transfer coefficients of up to 550 BTU/hr.ft^2 ^{o}F with a potential cost as low as \$ 0.36/ft^2.

ACKNOWLEDGEMENT

The author wishes to express his sincere appreciation to his research advisor, Dr. M. E. Findley, Associate Professor in Chemical Engineering, for his constant assistance and consultation throughout this investigation.

The author is also grateful to his wife for her encouragement and moral support.

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NOMENCLATURE

A	heat transfer area, ft^2
a, b	constants
C	capacity rate, BTU/hr. ^oF
c	constants
C_{oc}	price of screen, $\$/ft^2$
C_{op}	price of pipe, \$/ft.
C_{os}	price of metal sheet, $\$/ft^2$
C_p	specific heat, BTU/lb. ^oF
D	diameter, ft.
D_1, D_2, D_e	inside, outside, equivalent diameter, ft.
G	mass velocity, lb/hr.ft^2
G_r, N_{Gr}	Grashof number
h	film heat transfer coefficient, BTU/hr.ft^2 ^oF
K	Boltzmann's number
k	thermal conductivity, BTU/hr.ft. ^oF
L	length of heat exchanger, ft.
m, n	constants
N	number of channels
N_{Gz}	Graetz number
N_{Pr}	Prandtl number
N_{Re}	Reynolds number
N_{St}	Stanton number

N_u	Nusselt number
p	constant
Q	total heat transfer rate, BTU/hr.
q	heat flux, BTU/hr.ft^2
s	constant
T	temperature, ^oR
ΔT	temperature difference, ^oR
t	temperature, ^oF
Δt	temperature difference, ^oF
t_{h1}, t_{h2}	inlet and outlet temperature of hot fluid
t_{c1}, t_{c2}	inlet and outlet temperature of cold fluid
U	overall heat transfer coefficient, BTU/hr. ^oF ft^2
V	velocity, ft/hr.
X	thickness, ft.
W	flow rate, lb/hr.
π	3.14
μ	viscosity, lb/hr.ft.
ρ	density, lb/ft^3

Subscripts

a	based on arithmethic average
b	based on bulk temperature
c	at cold side
h	at hot side

i	inside channel
o	outside channel
w	based on wall temperature

I. INTRODUCTION

Heat exchangers are important to the economics of many processes and operations where thermal energy is involved. One type of process where heat exchangers are an important item is in thermal methods of water desalination, where heat consumption is a major cost of production, and large quantities of heat must be recovered with minimum loss of temperature. This recovery of heat requires a large area for heat transfer and causes the heat exchanger capital costs to increase rapidly as thermal driving forces and heat costs are reduced toward the minimums.

One potential method of water desalination is that of evaporation through non-wettable porous membranes from a hot saline solution through a vapor filled porous membrane with condensation onto a colder fresh water stream (1). This method includes a counter-current heat exchanger to recover as much heat from the fresh water as possible. This recovered heat is then used to heat the salt water feeding the evaporator. One advantage of this method is that both streams can be at nearly the same pressure and the strength requirements of both heat exchanger and evaporator may be minimal. If the low strength requirements permit the use of low cost but effective heat exchangers, then the capital costs can be reduced. Also, the system economics permit the use of lower driving forces and net heat consumption. Another economic factor involved in water desalination is the potential development of numerous large-scale desalination systems, which could permit low-cost mass production techniques on desalination equipment including heat exchangers.

This factor might justify the use of types of heat exchangers which are presently considered unconventional.

For the above reasons, as well as the fact that porous-membrane evaporators may be similar in construction and in some desired characteristics to heat exchangers, this investigation was undertaken to obtain information on two types of potential low-cost heat exchangers. The purpose of the investigation was to develop operating models of these possible inexpensive exchangers, obtain rough estimates of costs, and an approximate idea of their performance in terms of heat transfer coefficients.

II. THEORY REVIEW

Studies in heat transport and heat exchange have been made extensively since the 18th century. In this section some of the theories and literature concerned are reviewed and summarized briefly.

A. Heat Transfer Phenomena

When there exists a potential difference such as energy or concentration, etc., between or among neighboring regions, this difference acts as a driving force to reduce the difference by transporting the energy or mass from the region at higher potential to that at the lower potential. Thus a local temperature variation will cause the transport of thermal energy from a higher temperature region to a lower temperature region. This transport of thermal energy is accomplished either by one or all of the three modes, conduction, convection, and radiation. All of these three modes of heat transfer are important industrially and each finds application under different conditions. The work throughout this investigation has been carried out at moderate temperatures and therefore the effect of radiation will not be considered since radiation is significant in liquids only at high temperatures.

1. Conduction and Thermal Conductivity

In a solid body, the molecules are essentially confined to their positions in the solid structure. The transfer of heat, thus, depends upon the exchange of vibrational kinetic energy between the individual molecules. This process is called conduction. In liquids and gases, the molecules are no longer confined to a certain point but constantly change their position even if the substances are in a state of rest. Without bulk fluid currents, however, fluid heat transfer is still classified as conduction. In a dielectric solid such as solid metals, the transfer mechanisms will also include transport by the cloud of free electrons in addition to the lattice vibration (2).

Because each kind of transport is a result of the potential difference, the transfer rate is proportional to this difference, the driving force, and a rate equation is usually written in a manner as follows:

$$\text{Rate} = \frac{\text{Driving force}}{\text{Resistance}} \tag{2-1}$$

For unidirectional conduction the rate equation reads:

$$\frac{dQ}{dA} = -k\frac{dT}{dX} \tag{2-2}$$

Similarly, for multidirection,

$$\bar{q} = -k \nabla T \tag{2-3}$$

These two equations are the well known Fourier's Law (4) of heat conduction in the one dimensional form and three dimensional form. The constant k is called thermal conductivity and is dependent upon temperature and the properties of the substance.

Bridgman proposed the following equation to estimate the thermal conductivities of pure liquids (5).

$$k = 2.80 \left(\frac{N}{V}\right)^{2/3} K V_s \tag{2-4}$$

Where N is the Avogadro number

V is the molal volume

K is Boltzman constant

V_s is velocity of low frequency sound and is defined as

$$\left(\frac{C_p}{C_v}\left(\frac{\partial P}{\partial \rho}\right)\right)^{1/2}$$

The thermal conductivities of solids are much more complicated and have to be determined experimentally because they depend on many factors such as porosity, crystallite size, structure of the molecule and the metallic or nonmetallic nature.

For pure solid metals, the following equation is applicable

$$k = L k_e T$$

Where k_e = the electrical conductivity

T = absolute temperature

L = Lorenz number

$\simeq 2.45 \times 10^{-8}$ Volts^2 deg C^{-2}

This equation is the well-known Wiedermann-Franz and Lorenz equation (2).

2. Convection

Pure conduction in liquids and gases seldom occurs because of the probability of masses of the molecules moving under the influence of density difference and the resultant heat transfer is largely dominated by convection. In the process of heat convection, thermal energy is transported by the movement of streams carrying molecules of a higher energy content from regions of higher temperature to regions of lower temperature, and vice versa. This process is essentially restricted to fluid systems. The movement of the fluid streams are produced either by the temperature gradient (density difference) or by external forces. The former case is called natural convection and the latter forced convection.

Heat Transfer Between Solid and Flowing Fluid

Consider a fluid flowing in contact with a solid surface. The fluid particles in the vicinity of the solid surface are restrained by the viscous forces of the fluid and a thin film or boundary layer is thus formed over the solid surface. In a laminar sublayer, heat is conveyed by conduction and the effect of convection is insignificant. Sufficiently far from the solid surface the conductive heat flow is then swamped by the effect of convection. The thickness of the film depends on the properties of the fluid, including the temperature and the velocity. The surface character of the solid is also an important factor.

Film Heat Transfer Coefficient

Figure (2-1) shows the flow of heat from a phase boundary into a fluid sufficiently extensive for convective currents to exist. The two different transport processes operative give a fairly abrupt change in temperature across the film. The curve ABCD represent the temperature change in going from the surface to the bulk of fluid. AB represent the region where conduction alone is operative, the thermal resistance is high and the temperature gradient steep but constant. Farther from the surface, the effect of convection increases gradually and both the thermal resistance and temperature gradient become lower as shown by the region BC. There is no obvious point of demarcation which could be taken as the edge of the film and if we extend AB to cut the zero gradient line through D at E, then the distance from surface to E is defined as effective film thickness.

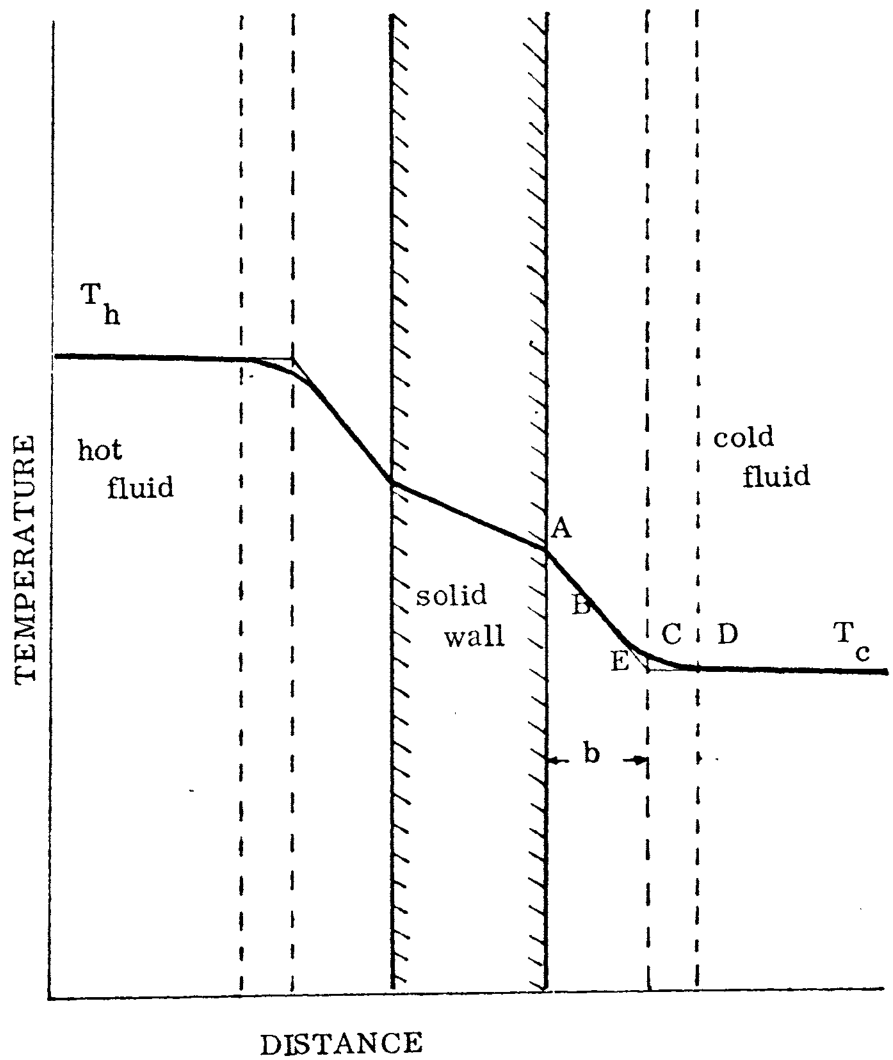


Figure (2-1) Heat Transfer Between Fluid and Solid

Fourier's equation is given below for the film.

$$\frac{Q}{A} = \frac{k}{b}(\Delta t) \tag{2-5}$$

Where k = effective conductivity of the fluid

b = effective film thickness

Δt = temperature change across the film

If we combine the two properties, k and b, to yield h the heat transfer film coefficient, or transmittance, then (2-5) becomes:

$$Q = h \Delta t A$$

The above equation is the Isaac Newton equation, for convective heat transfer.

The film coefficient is a function of the physical properties of the fluid flowing over the solid surface. Also the structure of the surface and the mechanisms of fluid flow have significant effects on the coefficient.

The film coefficient can be correlated with physical properties by stream parameters, either by measurements of the effective film thickness, theoretical analysis, or dimensional analysis of the variables in the system. Dimensional analysis is the usual method of obtaining correlations. It should be noted that the effects of solid surface character are usually neglected in correlation equations, this procedure can result in gross errors in some cases. For example, the convective heat transfer coefficient in a circular tube is a function of the following variables: tube diameter, distance from the entrance, viscosity, density, heat capacity, thermal conductivity, and the velocity of the fluid. Therefore, the film coefficient of heat transfer is usually correlated by equations including dimensionless groups of the above variables, such as Prandtl number, Nusselt number, Reynolds number, Stanton number, Graetz number, and others. These groups are defined as:

$$N_{Re} = \frac{D V \rho}{\mu} \qquad \text{Reynolds number}$$

$$N_{Pr} = \frac{Cp\,\mu}{k} \qquad \text{Prandtl number}$$

$$N_{u} = \frac{h D}{k} \qquad \text{Nusselt number}$$

$$N_{Gz} = \frac{Cp\,W}{k\ \ L} \qquad \text{Graetz number}$$

$$N_{St} = \frac{h}{Cp\,G} \qquad \text{Stanton number}$$

The most usual form of correlation is:

$$N_u = c\ (N_{Re})^m\ (N_{Pr})^n\ (\frac{D}{L})^a\ (\frac{\mu}{\mu_w})^b \tag{2-6}$$

Where $\frac{D}{L}$ is the ratio of diameter to the length of the pipe and $\frac{\mu}{\mu_w}$ is the ratio of viscosity of the fluid at bulk temperature to that at the solid surface temperature. For turbulent flow in a circular pipe the basic equation is:

$$N_u = c\ (N_{Re})^m\ (N_{Pr})^n \tag{2-7}$$

This equation is called Nusselt's equation. The constant c, m, and n have been determined by many previous experimenters and have been shown to vary slightly. W.H. McAdams (6) suggested

$$\frac{h D}{k} = 0.023\ (\frac{D V \rho}{\mu})^{0.8}\ (\frac{Cp\ \mu}{k})^{0.4} \tag{2-8}$$

for heating and cooling of nonviscous fluids where Reynolds numbers are above 10,000, Prandtl number is between 0.7 and 100, and $\frac{L}{D}$ is over 60 (8). Dittus-Boelter (3) proposed

$$\frac{h D}{k} = c\ (\frac{D V \rho}{\mu})^{0.8}\ (\frac{Cp\ \mu}{k})^{n} \tag{2-9}$$

Where c = 0.0243, n = 0.4 for heating

c = 0.0265, n = 0.3 for cooling

And A. P. Colburn modified equation (2-8) by changing 0.4 to 1/3 for heating.

$$N_u = 0.023\,(N_{Re})^{0.8}\,(N_{Pr})^{1/3} \tag{2-10}$$

This equation is generally applicable for ordinary liquids, if the $\frac{L}{D}$ is over 50 and the temperature drop is not excessive.

Another equation including the effect of viscosity was presented by McCabe and Smith (7) as

$$N_u = 0.023\,(N_{Re})^{0.8}\,(N_{Pr})^{1/3}\,\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{2-11}$$

For heat transfer in other than circular tubes, different correlations have been obtained for different shapes of channel. In smooth annuli, Wiegand (8) recommended the equation below,

$$\frac{h\,D}{k} = 0.023\,\left(\frac{D\;G}{\mu}\right)_b^{0.8}\,\left(\frac{Cp\,\mu}{k}\right)_b^{0.4}\,\left(\frac{D_2}{D_1}\right)^{0.45}$$

Where the subscript b indicates that the properties are evaluated at the bulk temperature.

Equation (2-9) has been indicated as applicable in noncircular ducts with moderate temperature differences (8). In laminar flow the forces convection sometimes is more complicated due to the fact that the intensity of the forced convection current is often not sufficient to damp out the natural convection current and both may occur simultaneously. Furthermore, the effects of $\frac{\mu}{\mu_w}$ and $\frac{D}{L}$ are also important. The equation

$$\frac{h_a\,D}{k} = 2\left(\frac{W\,Cp}{k\;L}\right)^{1/3}\,\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{2-12}$$

was derived by Sieder and Tate, and is used either inside or outside tubes (9).

The corresponding equation, called the Nusselt-type, is

$$\frac{h_a D}{k} = 1.86 \left(\frac{D G}{\mu} \frac{Cp \mu}{k} \frac{D}{L}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (2\text{-}13)$$

For fluids of moderate viscosity, the viscosity factor $\frac{\mu}{\mu_w}$ is neglected, and equation (2-12) becomes:

$$\frac{h_a D}{k} = 2 \left(\frac{Cp\ W}{k\ L}\right)^{1/3} \qquad (2\text{-}14)$$

Where h_a is based on the arithmetic average of Δt. Equation (2-12) is applied in the situation where the N_{Gz} is greater than 13 and the wall temperature is constant (15). If the temperature of the solid surface changes with length the situation becomes more complex. In general it is satisfactory to consider the wall temperature as constant. For Graetz number less than 13 the following equation is used (7).

$$\frac{h_a D}{k} = 2 \left(\frac{Cp\ W}{\pi k\ L}\right)^{1/3} \qquad (2\text{-}15)$$

Chen, Hawkins, and Solberg[10] presented a relation to predict the arithmetic mean heat transfer in annuli for laminar flow,

$$N_u = 1.02\, (N_{Re})^{0.45} (N_{Pr})^{0.5} \left(\frac{\mu_b}{\mu_1}\right)^{0.14} \left(\frac{De}{L}\right)^{0.4} \left(\frac{D_2}{D_1}\right)^{0.8} (Gr)^{0.05} \qquad (2\text{-}16)$$

where μ_b is based on the bulk temperature, and D_1 and D_2 are the diameters of inner and outer walls of the annuli.

The heat transfer in the region between laminar and turbulent flow is much different and it is difficult to establish a simple equation to predict the heat transfer coefficient in this region. The usual method is the graphical method based on plotting the Colburn factor, j_h, versus Reynolds number on a log-log plot.

The Colburn factors are defined as:

for turbulent flow $$j_h' = \frac{h}{Cp\,G}\left(\frac{Cp\,\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.023\left(\frac{D\,G}{\mu}\right)^{-0.2} \qquad (2\text{-}17)$$

for laminar flow $$j_h' = 1.85\left(\frac{D}{L}\right)^{1/3}(N_{Re})^{-2/3} \qquad (2\text{-}18)$$

The relationship of Colburn factors versus Reynolds number in the transition region are then represented by curves between the Reynolds of 2,000 and 10,000.

B. Heat Exchanger Analysis

A heat exchanger is the equipment necessary for the exchange of thermal energy between fluids at different temperatures. Heat exchangers usually can be summarized into two general types: direct transfer type and storage type. In the former type, the two fluids of either the same phase or different phases, exchanging the thermal energy are separated by a heat transfer medium and in the latter the thermal energy of a fluid at a higher temperature is periodically transferred and stored in a large heat capacity element, and then picked up by a fluid at lower temperature. In some cases, such as the cooling system of an automobile, cooling water picks up heat and then releases this heat to air. Such a system may be called a liquid coupled type although it essentially consists of two direct transfer type heat exchangers.

Numerous kinds of heat exchangers have been studied and designed according to the specific requirements of the situation. For example, small size and lightweight as well as efficiency are required for heat exchangers used in aircraft, special materials are required with corrosive fluids, and high heat flux rated heat exchangers are needed in gas turbine regenerators.

In industrial applications, the factors that determine the choice of material and the design of heat exchangers in most cases are the economic factors. The main purpose of this investigation is to try to find an inexpensive and simple type of heat exchanger and to determine its applicability for a specific process of water desalination.

The exchangers dealt with in this study are of the direct transfer type and theories are reviewed for this type only.

The process of heat transfer in a direct transfer type heat exchanger is as shown in Figure (2-1). There are three transfer steps; convectional transfer between hot fluid and solid wall, conduction in the solid wall, and convectional transfer between solid wall and cold fluid. The total driving force for the thermal energy transfer between the two fluids is the temperature difference between the two separate fluids.

$$
\begin{aligned}
q &= - h_i \Delta t_h \\
&= - \frac{k}{x} \Delta t_w \\
&= - h_o \Delta t_c \\
&= - U \Delta t
\end{aligned}
$$

where h_i, h_o = heat transfer coefficients the inside and outside of the wall.

Δt_h, Δt_c = temperature drop across the hot side film (inside) and cold side film (outside).

Δt = temperature difference between the two fluids.

U = overall heat transfer coefficient.

In the case where the solid wall is thin and has a high thermal conductivity, the thermal resistance existing in the solid wall, x/k is negligible in comparison with that existing in the films, $1/h_i$ and $1/h_o$, and the inside diameter and outside diameter can be considered to be the same and an equation relating the overall coefficient to the individual coeffients becomes:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \tag{2-19}$$

The two streams will reach the state of thermal equilibrium only when the streams are at the same temperature such that

$$t_h = t_c \tag{2-20}$$

and $\Delta t = 0$

By an energy balance, the heat transferred may be related to temperature as follows:

$$dq = -W_h C_{Ph} dt_h$$

$$dq = W_c C_{Pc} dt_c$$

where flows and differentials are taken in the direction of the hot fluid flow.

or

$$\frac{dt_h}{dt_c} = \frac{-W_c C_{Pc}}{W_h C_{Ph}} \tag{2-21}$$

Considering a counter flow heat exchanger, equation (2-20) represents the equilibrium line and equation (2-21) represents the operating line.

If Δt_e is temperature difference between the two fluids entering the heat exchanger and Δt_o represents the larger of the temperature rise of the cold stream or the temperature drop of the hot stream, then the ratio of Δt_o to Δt_e is defined as heat exchanger effectiveness.

$$
\begin{aligned}
E &= \frac{\Delta t_o}{\Delta t_e} \\
&= \frac{t_{h1} - t_{h2}}{t_{h1} - t_{c1}} \qquad \text{if } t_{c2} - t_{c1} < t_{h1} - t_{h2} \\
&= \frac{t_{c2} - t_{c1}}{t_{h1} - t_{c1}} \qquad \text{if } t_{c2} - t_{c1} > t_{h1} - t_{h2}
\end{aligned}
\tag{2-22}
$$

or,

$$
E = \frac{1 - \exp\left[-\frac{U\,A}{C_{min}} \left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \frac{C_{min}}{C_{max}} \exp\left[-\frac{U\,A}{C_{min}} \left(1 - \frac{C_{min}}{C_{max}}\right)\right]}
\tag{2-23}
$$

as shown in reference (11),

where $C = W\,C_p$ = is the capacity rate BTU/hr ^oF

C_{min}, C_{max} = the smaller capacity rate and larger capacity rate.

From the equation (2-23), it is obvious that we can increase the effectiveness of a heat exchanger by the following methods: 1) increase the transfer area A, 2) decrease the capacity rate ratio C_{min}/C_{max}, or 3) increase the overall transfer coefficient U. Method (2) is unfavorable because a decrease in the capacity ratio may relatively reduce the amount of heat transferred. The efforts that people have made to improve the effectiveness are either to increase the transfer area with minimum cost or to promote the transfer coefficient by increasing the Reynolds number, roughness of the surface, and modifying the flow pattern.

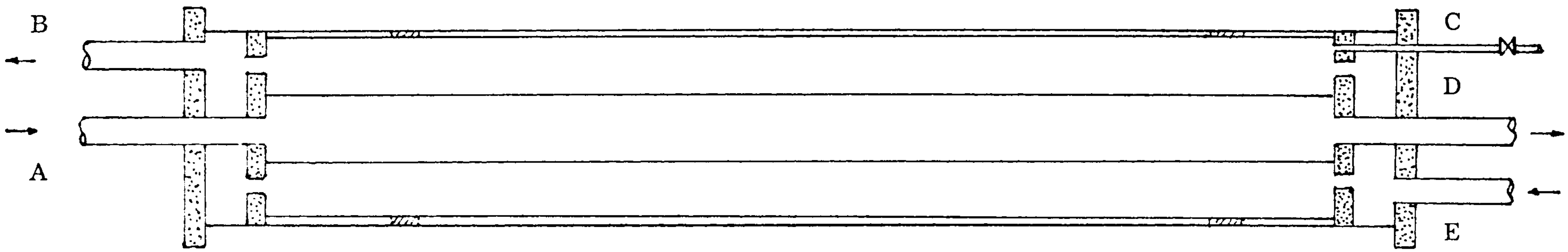
III. EXPERIMENTAL

A. Experimental Apparatus

Four simple heat exchangers have been tested in this investigation. Three of them were of the double pipe type with the inner circular tube replaced by a channel of a cross sectional shape as shown in Figure (3-1), Figure (3-2) and Figure (3-3). The fourth heat exchanger was a plate type with a screen spacer between plates and is shown in Figure (3-4). The design and construction of these exchangers were suggested in principle by the author's advisor. The dimensions of the four exchangers are given in Tables (2-1) and (2-2).

Heat Exchanger (E): Ease of construction and availability of materials were important factors considered in setting up the equipment in this investigation. Aluminum sheets with a thickness of 0.007 inches were used to construct the main part of the heat exchangers and served as the transfer medium. Plexiglas sheets 0.5 and 0.7 centimeter in thickness was applied to close both ends of the exchanger. The inlet and exit lines were made of glass tubes. The diameters of the tubes were 0.300 inches for inlet and 0.375 inches for exit. A small line is connected to the top part of the inner channel to vent any inert gases released from the flowing liquid. All these materials were held together with silicone glue which is water repellent and temperature resistant. A plugged tube was inserted in the second exchanger while three baffle disks were placed in the third. The details of these exchangers are illustrated in Figure (3-1) through Figure (3-4).

The shapes of the inner channel were chosen to provide more transfer area



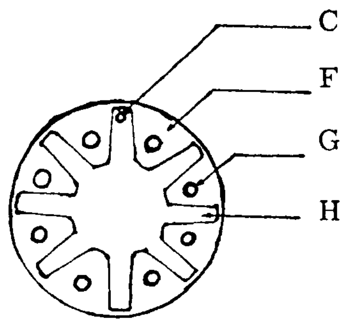
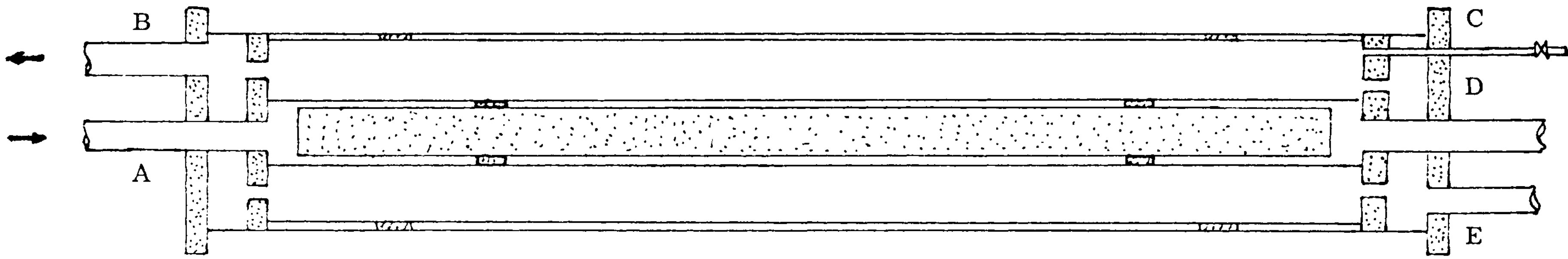


Figure (3-1) Heat Exchanger I

A. Hot Fluid Inlet Line
B. Cold Fluid Exit Line
C. Inert Gases Vent
D. Hot Fluid Exit Line
E. Cold Fluid Inlet Line
F. Cold Fluid Channel
G. Cold Fluid Distributing Holes
H. Hot Fluid Channel



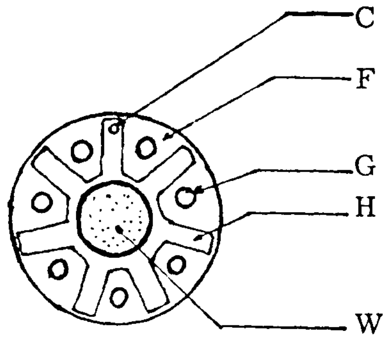


Figure (3-2) Heat Exchanger II

A. Hot Fluid Inlet
B. Cold Fluid Exit
C. Inert Gases Vent
D. Hot Fluid Exit
E. Cold Fluid Inlet
F. Cold Fluid Channel
G. Cold Fluid Distributing Holes
H. Hot Fluid Channel
W. Plugged Tube For Baffleing

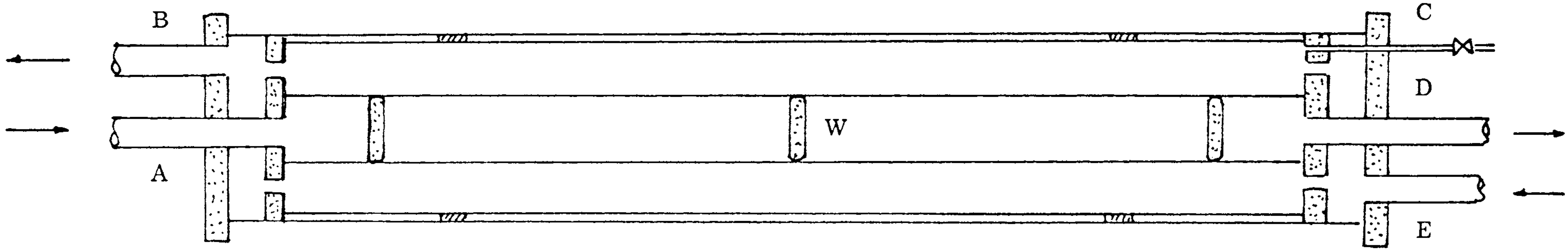
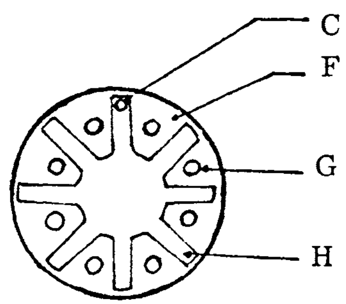


Figure (3-3) Heat Exchanger III



A. Hot Fluid Inlet
B. Cold Fluid Exit
C. Inert Gases Vent
D. Hot Fluid Exit
E. Cold Fluid Inlet
F. Cold Fluid Channel
G. Cold Fluid Distributing Holes
H. Hot Fluid Channel
W. Disk Baffler

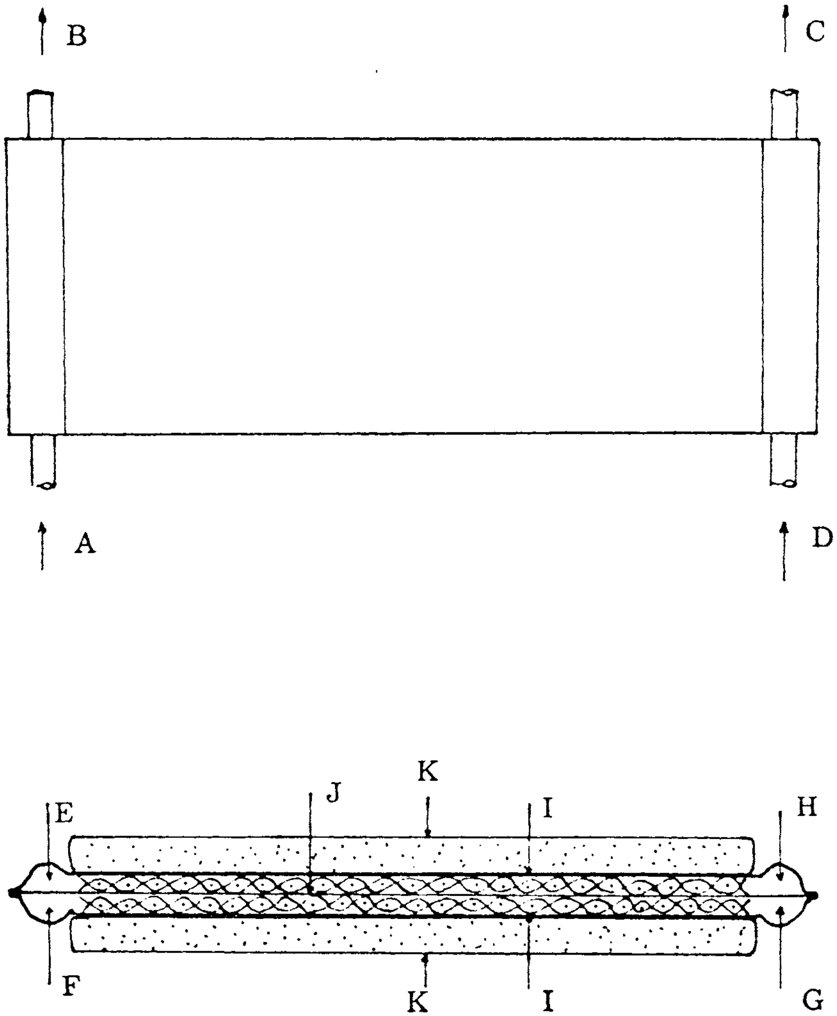


Figure (3-4) Heat Exchanger IV

A. Hot Fluid Inlet
B. Cold Fluid Exit
C. Hot Fluid Exit
D. Cold Fluid Inlet
E. Cold Fluid Exit
F. Hot Fluid Distributing Channel
G. Hot Fluid Exit
H. Cold Fluid Inlet Distributer
I. Screens
J. Heat Transfer Surface
K. Wooden Plates

with less material used and space occupied. In the fourth exchanger, two fine aluminum screens were placed between the plates primarily to separate the plates. The exchanger was then clamped between two wooden boards to keep the screen in contact with the aluminum sheets when pressure was applied to force the fluid through the channels. The sketch is shown in Figure (3-5).

Data obtained on these exchangers are shown in Table (2-1) and Table (2-2).

Pump (P): A centrifugal pump of 1/2 horse power was used to pump the hot fluid. A by-pass line was installed across the pump for adjusting the flow rate. The expected flow measurement accuracy was about 1 percent. The variations in flow were approximately 2 percent.

Thermocouples (TR 2, TR 3, TR 4, TR 5): Four thermocouples of copper constantan were inserted separately into the tubes of each inlet and exit of the heat exchanger. All thermocouples were precoated with silicone glue to prevent the thermocouple from directly contacting the passing fluid.

The thermocouples were calibrated at the boiling point of water and the deviations from the standard potentials were assumed to be a linear function of the difference from the reference temperature, the room temperature. The corrections were applied in the computer calculations on the raw data.

Heat Exchanger Fluid: The heat exchanger fluid was tap water in all cases. Inspection of exchangers after use indicated negligible scale formation.

TABLE 2-1

Description of Heat Exchanger I, II, III

Exchanger Number	1	2	3
Inner channel			
length feet	1.395	1.250	1.250
perimeter feet	0.665	0.725	0.665
cross section area sq. ft.	0.0081	0.0035	0.00585
equivalent diameter feet	0.049	0.0197	0.035
Outside channel			
length feet	1.441	1.355	1.415
perimeter feet	1.155	1.000	1.082
cross section area sq. ft.	0.0101	0.0081	0.008
equivalent diameter feet	0.035	0.0322	0.0296
Total area for heat transfer sq. ft.	0.930	0.729	0.847

Remark: The diameter of the plugged tube used in number 2 heat exchanger is 0.045 feet.

TABLE 2-2

Description of Heat Exchanger IV

Area for heat transfer, sq.ft.	1.07 x 0.291
Screen	
wire diameter inch	0.011
no. of wires/in. width screen	16
thickness of screen inch	0.024
cross sectional area of screen*	0.000298
Net cross section area sq.ft.*	0.000283
Wet perimeter ft.*	1.325
Equivalent diameter	0.00085

* Based on screen cross section of wires parallel to flow and one wire perpendicular to flow across the flow cross section.

Potentiometer (V): A potentiometer was used to measure the e.m.f. of the thermocouples produced by the temperature difference between the two ends of the thermocouple. The reference temperature was the room temperature which was read on a mercury thermometer. The readings were in millivolts and were converted to give the temperatures at the specific point measured. Thermocouple readings corrected by calibration were accurate to $1\,^{\circ}F$.

Container (R): A container was placed above the pump to furnish the head of the suction line of the pump and to make up hot fluid.

Heat Elements (H): An electric heater was used to supply the heat source. A burner was also used when the electric heat source was inadequate.

Insulation: Fiber glass was used for the purpose of insulation of the heat exchanger and the tubes. A wrapping with aluminum foil was then applied to cover the fiber glass. All parts of the equipment between temperature measuring points were insulated. Heat losses were from the cold fluid (outside) channel on the first 3 exchanger types and from both cold and hot fluids on the 4th type. Estimated heat loss in 1 channel was less than 40 BTU/hr,

Miscellaneous: A thermometer was used to measure the room temperature when the e.m.f. was read. A thermocouple selector was used for convenience. A stop watch and a 1,000 c.c. graduated cylinder were used to measure flow rates. A manometer was prepared and used with the fourth exchanger to measure the pressure drops across the heat exchanger, within approximately 0.05 in Hg.

B. Procedures

The flow sheet of the experiment is shown in Figure (3-5). With the by-pass opened, the pump was started to circulate the fluid and the heater was turned on for heating. The by-pass and both inlets to the exchanger were adjusted to obtain desired flow rates. When the system reached a steady state, the potentiometer was standardized and the four e.m.f. values were read. Room temperature was also recorded. Soon after that, both flow rates were measured by collecting the volume of the fluid flowing in a measured period of time.

These measurements of e.m.f. and flow rate were repeated once or twice at the same conditions. In the fourth exchanger, the plate heat exchanger, the pressure drops of both sides were read in inch of Hg. The burner was used to adjust the hot fluid near a specified temperature. The temperature of the hot fluid and the flow rates were alternatively changed to secure data at different conditions. The e.m.f. data were converted into temperatures and the overall heat transfer coefficients were then calculated.

C. Methods of Calculation and Assumption

1. Heat exchangers data

The total area for heat transfer was obtained by direct measurement of the aluminum sheet transfer medium separating the two fluids. The areas on both sides of the sheet were assumed to be the same because of the small thickness compared to curvatures.

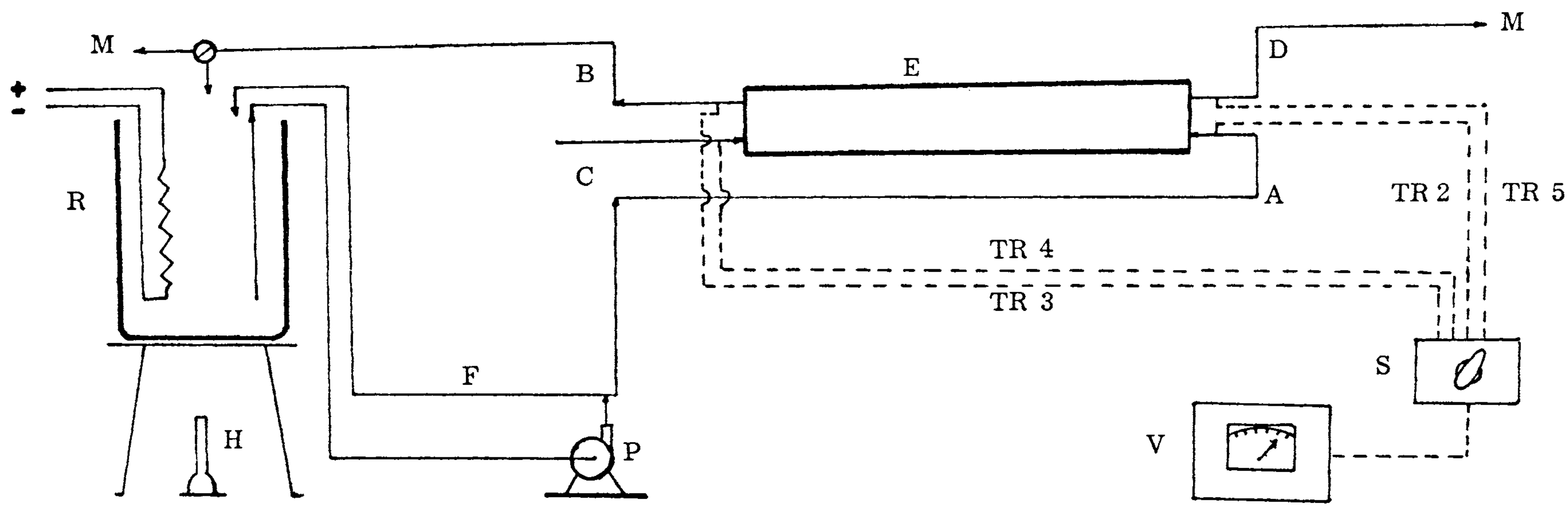


Figure (3-5)

FLOW SHEET

H. Heat Source
P. Pump
F. By-Pass
R. Container
M. To Measurement
A. Hot Flow In
B. Hot Flow Out
E. Heat Exchanger
C. Cold Flow In

D. Cold Flow Out
V. Potentiometer
S. Selector Switch
TR2. Thermocouple
TR3. Thermocouple
TR4. Thermocouple
TR5. Thermocouple
M. To Flow Rate Measurement

Each fin was made of 1 inch width metal sheet with the proportion of $\frac{3.5}{8}, \frac{1}{8}, \frac{3.5}{8}$, Owing to the technique, the fins were not uniform and it was difficult to obtain the cross sectional area of each channel by calculation. For this reason, the cross sectional area was obtained by filling each channel with water and dividing the net volume of water by the length of the channel. The equivalent diameter of each channel was obtained by the calculation of hydraulic radius. Or

$$D_e = \frac{4 \text{ x cross sectional area}}{\text{perimeter}}$$

2. Water Properties

For convenience in calculating the heat transfer coefficients by computer, the dependence of water properties on temperature have been approximated by the following equations using least square techniques. The details of the approximations are given in appendix A. The original data used for approximation were obtained from Compact Heat Exchangers by W. M. Kays and A. L. London, 1958[11].

Density lbs/ft^3

$$\rho = 64.81 - 0.02212\, t - 60.15/t$$

Viscosity lbs/hr-ft

$$\mu = -0.29 + 216.4/t - 2194/t^2$$

Conductivity BTU/ft-$^\circ$F

$$k = 0.425 - 7.37/t + 127/t^2$$

Specific heat BTU/lb-$^\circ$F

$$C_p = 0.979 + 0.00012\, t + 0.811/t$$

<u>Prandtl Number</u> $N_{Pr} = C_p \mu/k$

$$N_{Pr} = 1.15 + 624.5/t - 4658/t^2$$

where t is in ^oF and the range is from 32^oF to 240^oF.

3. <u>Calculation of Overall Heat Transfer Coefficient</u>

The energy losses due to pressure drops across the heat exchangers, were small and it was assumed they were negligible. In considering the process as an isobaric process, the heat transferred across the medium was obtained by calculating the enthalpy change of each fluid from inlet to outlet.

$$Q = W_h C_{ph} \Delta t_h = W_c C_{pc} \Delta t_c$$

and

$$U = \frac{Q}{A \Delta t_\ell} \tag{3-1}$$

where W is flow rate in lb/hr. Δt_h and Δt_c are the temperature change of both fluids and Δt_ℓ is the logarithmic mean of temperature difference between the two flowing fluids.

Theoretically, the heat released from hot fluid must be equal to the heat absorbed by cold fluid, but because of heat loss and error in measurement, there exists some differences between the two quantities. The equation

$$Q = W_h C_{ph} \Delta t_h$$

in the first 3 exchangers and

$$Q = W_c C_{pc} \Delta t_c$$

in the fourth exchanger were used to determine the heat transfer coefficient because the fewer possibilities for errors existed with these described equations. This was true because heat losses occurred from the outer cold channel in the first 3 types and from both channels in the fourth type. In exchanger 4, the heat losses were less from the cold fluid because of the lower Δt from the room. The maximum cold side temperature minus room temperature was about 70° F, and this existed at only 1 point in the exchanger.

D. Data and Results

All the data taken at different flow rates and temperatures and the overall heat transfer coefficients are listed in Table 1 through Table 4, Appendix B. The results and the calculation of individual heat transfer coefficients and cost estimates will be presented in the following section.

IV. DISCUSSION

As has been indicated previously, the object of this investigation was primarily to find simple, inexpensive but effective heat exchangers which can be used in the membrane-evaporation method for water desalination to obtain the best heat recovery and reduce the net heat consumption to the minimum. It is difficult to obtain accurate correlations of individual heat transfer coefficients for this investigation because other information is needed, such as wall temperatures, the end effects, and the temperature profiles of the wall and both fluids. Some of the characteristics were not varied and prevented the study of the effects of number and dimension of the longitudinal fins and the effect of wetted perimeter and the perimeter available for heat transfer. The work throughout this experiment emphasized the promotion of the heat transfer coefficient and the reduction of the costs of heat exchangers. The study of all the complex factors was beyond the scope and purposes of this investigation. For this reason, the heat transfer coefficients are correlated with flow rate and temperature only roughly by following the models used by previous investigators and the other effects were not considered.

A. Applicability of Aluminum for Water Desalination Process

The fact that copper and its alloys have served as superior heat transfer mediums for a long history may be changed by the development of aluminum and its alloys in heat transfer engineering. The corrosion due to salt water on many metals limits the applicability of these metals for the desalination of sea water. This reduces the possibilities of producing fresh water by desalination at a commercial price for daily living. In their report, M. W. Wei and E. T. Wanderer (12) indicated that the resistances of aluminum and its alloys to salt water corrosion are satisfactory and the use of aluminum for water desalination should be feasible. Other information from Alcoa (13) indicated that the cost of fresh water produced by a thermal desalination method will be greatly lowered by the utilization of an all-aluminum system. In light of these facts, the application of aluminum in membrane-evaporation desalination should improve the economics of this method.

B. Correlation of Heat Transfer Coefficients

1. Approximation of Individual Heat Transfer Coefficients

The overall heat transfer coefficients calculated from equation (3-1) were to be correlated as functions of the fluid conditions. The properties and dimensionless numbers of the fluids were evaluated at their average bulk temperature, the arithmatic average of the inlet and exit temperatures of each fluid.

One approximation was approached according to equation (2-7) and equation (2-19).

$$\frac{hD}{k} = c\,(N_{Re})^{m}\,(N_{Pr})^{n} \tag{2-7}$$

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \tag{2-19}$$

For rough approximation, with the first three exchangers, the parameters m and n were assumed to be the same for both channels but c was assumed different. This assumption was based on the fact that the heat transfer surfaces on both sides are similar in geometrical shape and that the Reynold's number is a function of the variables of flow rate and temperature and the Prandtl number is a function of temperature only. The difference in parameter c was based on the fact that the ratio of perimeter for heat transfer to the total perimeter was different in each of the channels. The first assumption might also be incorrect. However, the error in this assumption may be compensated more or less by the parameter c. Thus, to correlate the overall coefficient rather than the individual coefficients the effect of variations in m and n should be minor. In the fourth exchanger both geometrical construction and perimeter for heat transfer are similar on both sides; thus, the parameters are assumed to be the same on both sides of the transfer medium.

If equation (2-7) and equation (2-19) are combined,

$$\frac{1}{U} = \left(\frac{D}{c\,k\,(N_{Re})^{m}(N_{Pr})^{n}} \right)_i + \left(\frac{D}{c\,k\,(N_{Re})^{m}(N_{Pr})^{n}} \right)_o \qquad (4\text{-}1)$$

where i and o are subscripts for inside channel and outside channel. The parameters C_i and C_o, m, and n are then approximated by non-linear least squares fitting and the results are listed in Table (4-1). From these results, a value of 2/3 is taken for m and 1/3 for n. The rest of the parameters are chosen so that the equations fit the data best with the above values of m and n, and the modified equations become

For exchangers No. 1, No. 2, No. 3

$$N_u = c\,(N_{Re})^{2/3}\,(N_{Pr})^{1/3} \qquad (4\text{-}2)$$

For exchanger No. 4

$$N_u = c\,(N_{Re})^{0.4}\,(N_{Pr})^{-0.14} \tag{4-3}$$

The values of c for each channel of each heat exchanger and the standard deviation of the errors for each heat exchanger by using equation (4-2) and equation (4-3) are listed in Table (4-3).

Another similar approximation is based on equation (2-12) as follows:

$$\frac{hD}{k} = S\,(N_{Gz})^{p} \tag{4-4}$$

The values of S and p obtained by least square fitting are listed on Table (4-2). An average value of p is taken as 0.45 for the first three heat exchangers and 0.31 for that of the fourth. Then the values of S for each channel and each exchanger are again chosen and listed on Table (4-4) with the standard deviations of equation (4-5) for the first three exchangers and equation (4-6) for the fourth exchanger. For the first three exchangers

$$\frac{hD}{k} = S\,(N_{Gz})^{0.45} \tag{4-5}$$

For the fourth exchanger

$$\frac{hD}{k} = S\,(N_{Gz})^{0.31} \tag{4-6}$$

The outside channels of the first three exchangers give similar results in Tables (4-3) and (4-4) as expected. The unexpected negative exponent in Equation (4-3) is probably due to experimental variations. With this type exchanger Prandtl number apparently has little effect.

Table 4-1

Paramcters of Equation (4-1) by Least Square Fitting

Heat Exchanger Number	1	2	3	4
C_i	0.0115	0.056	0.076	0.344
C_o	0.304	0.079	0.081	0.344
m	0.641	0.743	0.709	0.391
n	0.404	0.357	0.453	-0.136

Table 4-2

Parameters of Equation (4-4) by Least Square Fitting

Heat Exchanger Number	1	2	3	4
S_i	1.239	0.975	0.738	0.370
S_o	2.789	1.204	0.452	0.370
P	0.399	0.409	0.534	0.310

Table 4-3

Values of c of Equation (4-2) and (4-3) and the Standard Deviation of Each Heat Exchanger

Exchanger Number	1	2	3	4
C_i	0.18	0.10	0.14	0.34
C_o	0.15	0.12	0.12	0.34
Standard Deviation of (U_M-U_E)	9.73	11.5	6.54	46.9

Table 4-4

Values of S of Equation (4-5) and (4-6) and the Standard Deviation of Each Heat Exchanger

Exchanger Number	1	2	3	4
S_i	1.40	0.80	1.10	0.37
S_o	1.00	0.90	1.00	0.37
Standard Deviation of (U_M-U_E)	9.56	10.9	11.4	52.4

2. Comparisons of Measured Valued with Estimated Values

The comparisons of the measured overall heat transfer coefficients U_M calculated using equation (3-1) with the estimated values U_E from equation (4-2), (4-3), (4-4), and (4-5) are illustrated in Figure (4-1) through Figure (4-4). The standard deviation of the differences from the estimated values of U, U_E, are given in Tables (4-3) and (4-4). The standard deviations indicate little difference in reliability between equations (4-2) and (4-5) or between (4-3) and (4-6).

3. Accuracy

The errors that could have occurred during the experiment include the flow rate measurements and e.m.f. measurements. A change in suction head to a centrifugal pump will cause a change of flow rate and thus temperature. Temperature errors arise during the measurements of e.m.f. when the system is not at steady state. Error may also exist in the conversion of e.m.f. to temperature by a plot which was used for calculations. Errors in temperature were less than 1^{o} F compared to temperature differences within streams as low as 3.1^{o} F.

The data in Appendix B show the difference between heat transfer based on cold side calculations and heat transfer based on hot side calculations. From these differences, it is estimated that the maximum error of overall heat transfer coefficients should be within 15 percent.

There are some other factors which do not affect the measurement but do induce variation in U_M or in U_E. These factors are those such as possible

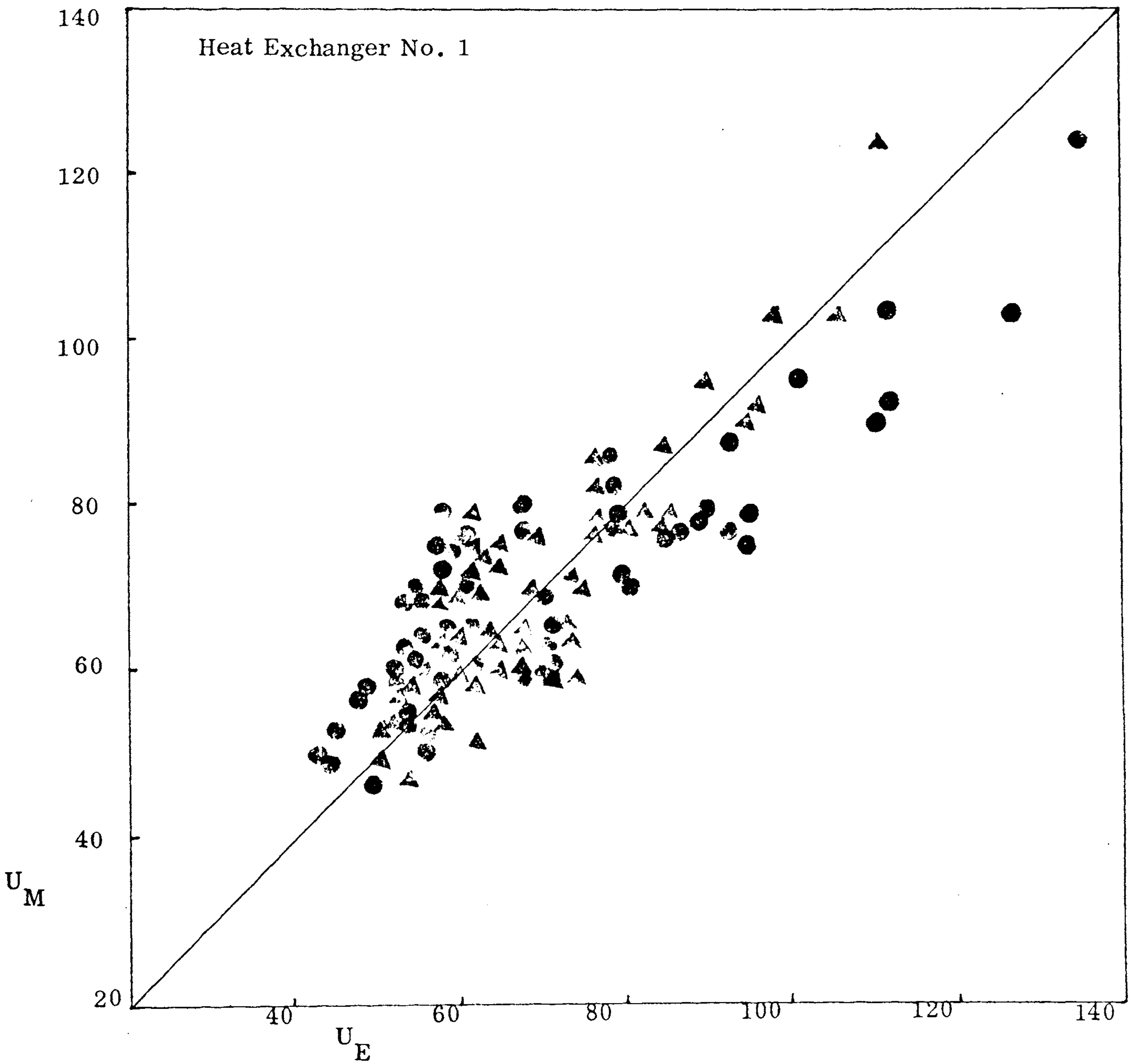


Figure (4-1) U Measured vs U Estimated

● $$U_E = \left(\frac{1}{0.18\,(N_{Re})_i^{2/3}(N_{Pr})_i^{1/3}} + \frac{1}{0.15\,(N_{Re})_o^{2/3}(N_{Pr})_o^{1/3}} \right)^{-1}$$

▲ $$U_E = \left(\frac{1}{1.4\,(N_{Gz})_i^{0.45}} + \frac{1}{1.0\,(N_{Gz})_o^{0.45}} \right)^{-1}$$

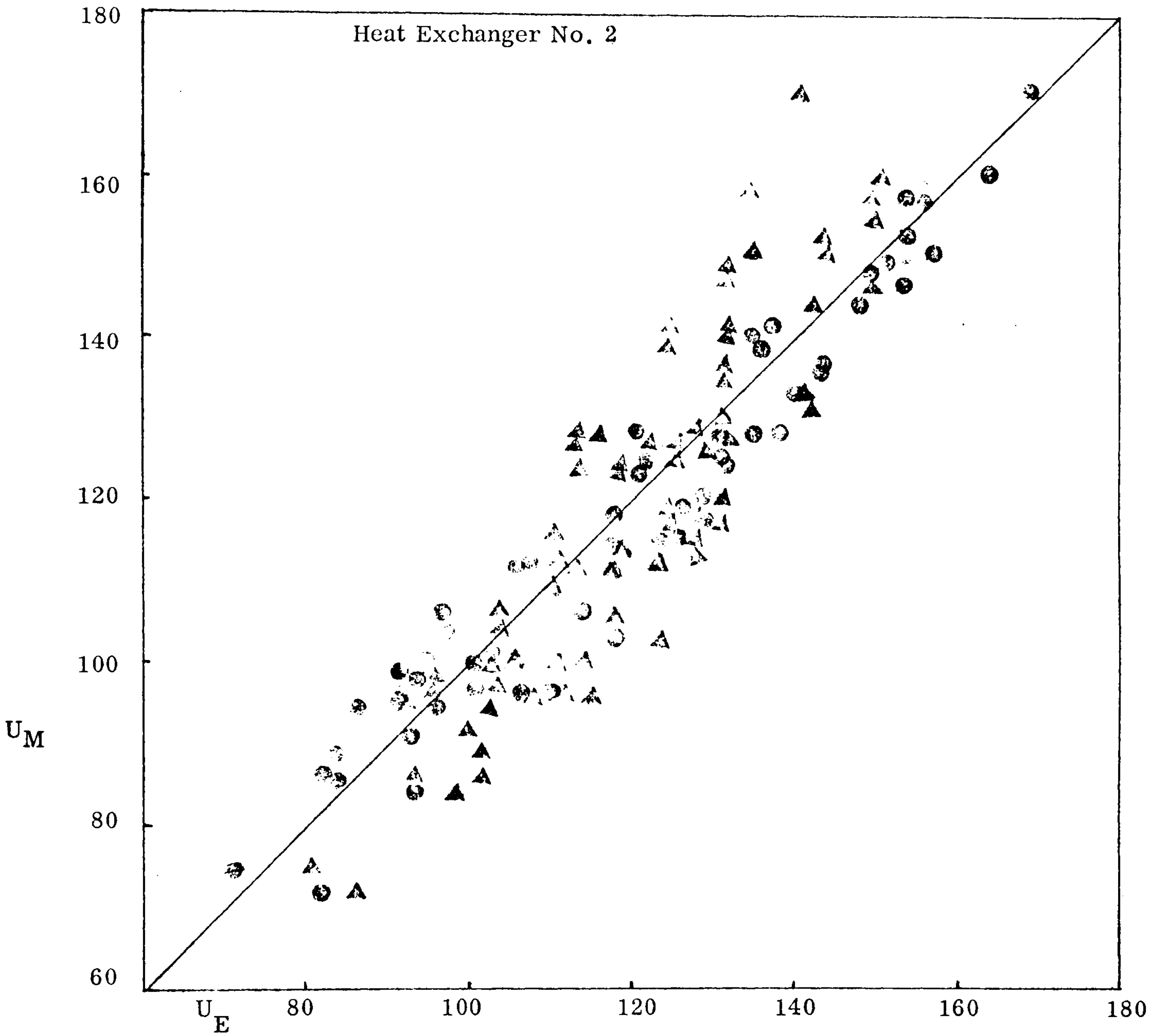


Figure (4-2) U Measured vs U Estimated

● $U_E = \left(\frac{1}{0.10\,(N_{Re})_i^{2/3}(N_{Pr})_i^{1/3}}\right) + \left(\frac{1}{0.12\,(N_{Re})_o^{2/3}(N_{Pr})_o^{1/3}}\right)^{-1}$

▲ $U_E = \left(\frac{1}{0.8\,(N_{Gz})_i^{0.45}} + \left(\frac{1}{0.9\,(N_{Gz})_o^{0.45}}\right)^{-1}\right.$

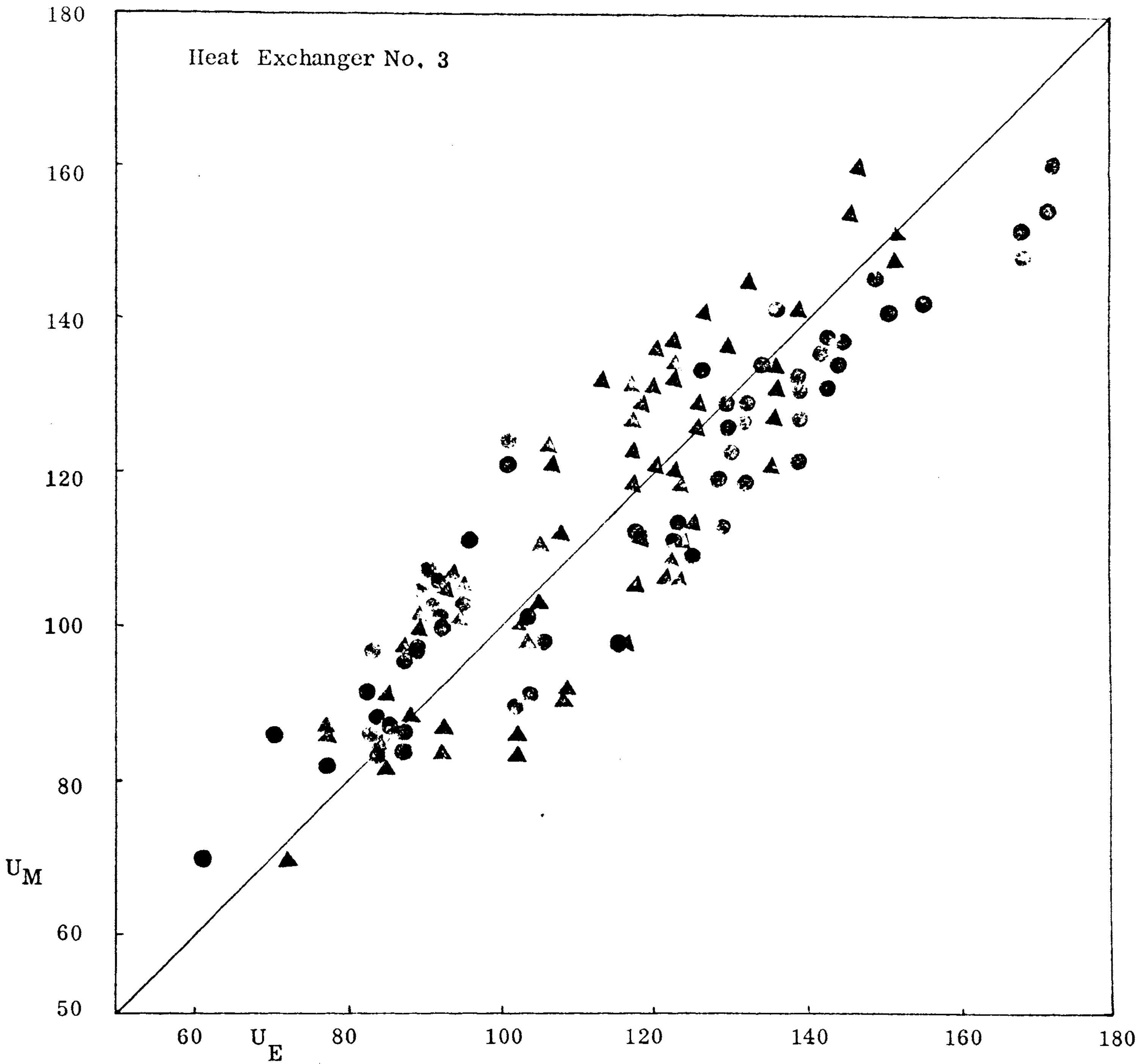


Figure (4-3) U Measured vs U Estimated

● $$U_E = \left(\frac{1}{0.14\,(N_{Re})_i^{2/3}(N_{Pr})_i^{1/3}}\right) + \left(\frac{1}{0.12\,(N_{Re})_o^{2/3}(N_{Pr})_o^{1/3}}\right)^{-1}$$

▲ $$U_E = \left(\frac{1}{1.1\,(N_{Gz})_i^{0.45}}\right) + \left(\frac{1}{1.0\,(N_{Gz})_o^{0.45}}\right)^{-1}$$

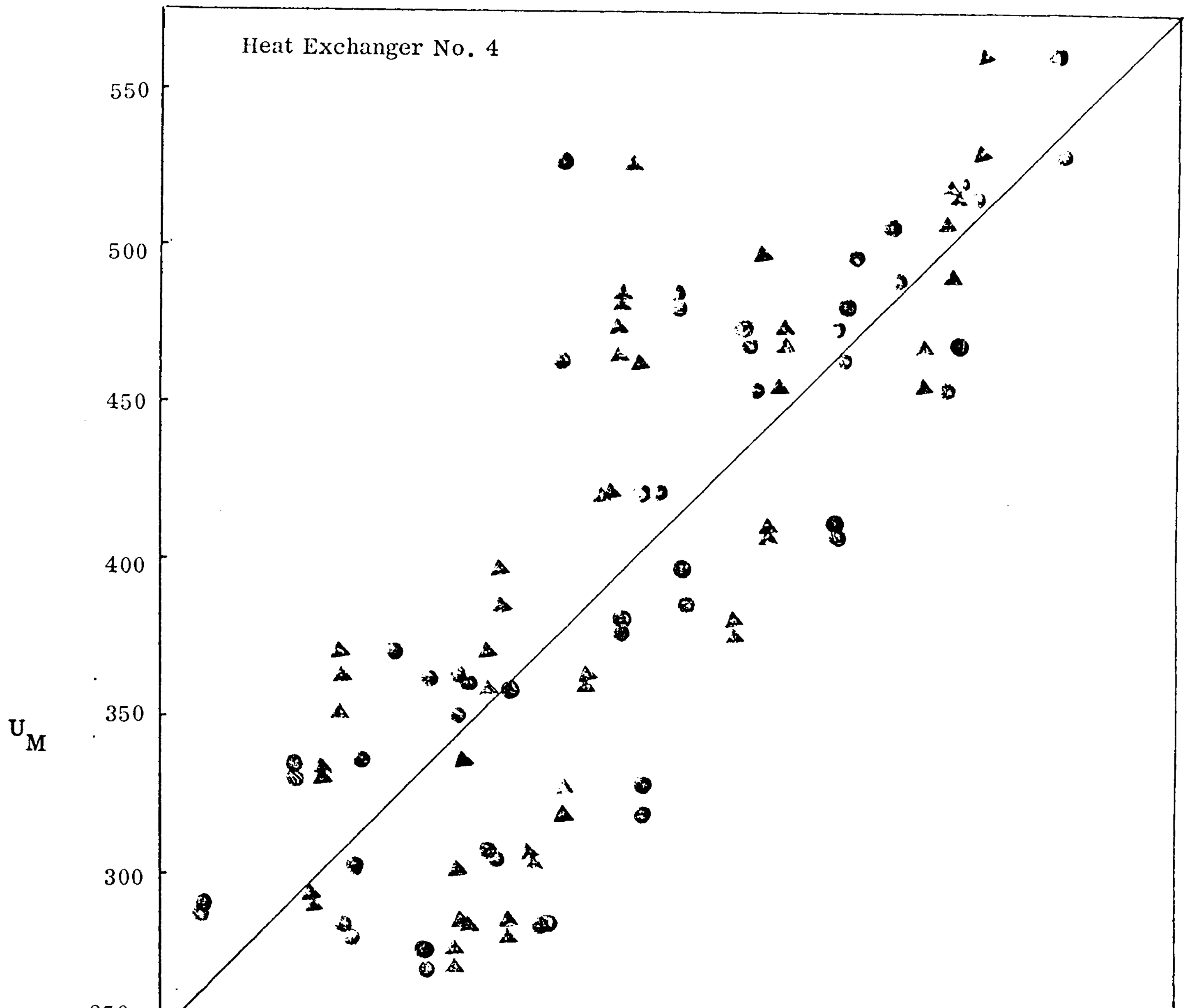


Figure (4-4) U Measured vs U Estimated

● $U_E = 0.344 \left(\frac{1}{(N_{Re})_i^{0.4} (N_{Pr})_i^{-0.14}} + \frac{1}{(N_{Re})_o^{0.4} (N_{Pr})_o^{-0.14}} \right)^{-1}$

▲ $U_E = 0.38 \left(\frac{1}{(N_{Gz})_i^{0.31}} + \frac{1}{(N_{Gz})_o^{0.31}} \right)^{-1}$

changes in cross sectional area and shape when the channels are not at the same pressure, possible air collection in channels, and poor flow distribution. The degree of these variations is difficult to estimate.

C. Cost Estimate

This estimate of the cost of these exchangers is based on equipment and conditions similar to those used in this investigation. Material choice is based on economics and known availability. The prices of materials were obtained from a current catalog (14). Labor and accessory costs were based on assumptions.

Type 1. For the type of number 1 through number 3 exchanger

Material Cost		
	Outside shell	$C_{op} \times L$
	Transfer medium	$C_{os} \times 2\pi D \times L$
	Accessory items	\$ 4.00
Labor Cost		\$ 5.00
Total Cost		$(C_{op} + 2\pi D \times C_{os}) \times L + \$\,9.00$
Total area		$2 \times \pi D \times L$

where C_{op} is price per foot of aluminum pipe of diameter D

C_{os} is price per square foot of aluminum sheet and

L is the length of exchanger.

The perimeter of inner channel is assumed to be twice the perimeter of the outside shell and thus the inner area will be twice the shell area.

Assume a pipe of 0.02 wall thickness and 6 inches in diameter is used as shell and a 0.019 thickness aluminum sheet is used as transfer medium. Then, if the length is 12 feet,

C_{op} = \$ 0.425 $\qquad$ C_{os} = \$ 0.22

Total cost = \$ 22.4

Total area = 38 feet^2

Cost/feet^2 = \$ 0.60

<u>Type 2. For type of number 4 exchanger</u>

In this type of exchanger, if many exchangers are installed together, then 1/3 of the material can be saved. Let C_{oc} be the price per square foot screen, N be the number of channels and L and W are the length and width of the exchanger.

Material Cost

Aluminum sheet	$C_{os} \times L \times W \times (N + 1)$
Aluminum screen	$C_{oc} \times L \times W \times N$
Accessory items	$1.0 \times N$
Labor Cost	$\$\,5.00 + 0.10 \times N$
Total Cost	$\$\,5.00 + 1.1N + C_{os} \times L \times W + (C_{os} + C_{oc}) \times W \times L \times N$
Total transfer area	$L \times W \times (N-1)$

C_{oc} of screen of mesh 16 x 18 is \$ 0.07. If an exchanger is 10 feet wide, 12 feet long and 10 channels are chosen, then

Total Cost = \$390.4

Total Area = \$1,080 ft^2

Cost/ft^2 = \$0.36

Increasing transfer area or N in each exchanger will reduce the first capital cost per unit transfer area.

If we used the same amount of material as used in type 1 for construction of a double pipe exchange, the total area available for heat transfer is 23.5 ft^2 and cost per unit area is about \$ 0.95. This first capital is much more than that of either type 1 or type 2.

D. Applicabilities of the Tested Heat Exchangers

In the first three tested heat exchangers, the heat transfer coefficients are about $1 \times (N_{Gz})^{0.12}$ times those obtained from circular tubes under the same conditions of temperature and mass velocity, and those of number 4 are $4 \times (N_{Gz})^{-0.02}$ times better than those of double pipe. These increases in heat transfer coefficients and the reduction of first capital cost are accompanied by the expenses of friction loss. It is difficult to conclude the advantages of these exchangers over double pipe heat exchangers without sufficient information concerning the pressure drop due to friction loss to compare the gain in recovery of heat and loss in power. But it is apparent that the use of aluminum for heat exchanger construction in membrane-evaporation method of water desalination could potentially reduce exchanger first capital from \$ 2.00/ft^2 (1) to \$ 0.60/ ft^2 in the first three tested exchangers and to \$ 0.36/ft^2 in the number 4 exchanger.

Since costs of pumping liquids are usually minor cost items, it seems likely that exchangers of the type of number 4 are the most promising due to both a low capital cost and a high heat transfer coefficient. The only disadvantage would be a higher pressure drop through the exchanger, and the use of shorter elements in parallel might reduce this disadvantage.

V. CONCLUSIONS

From the results of this study, the following conclusions are drawn.

For use in desalination by evaporation through porous membranes and possibly other uses, the first cost of heat exchangers may be reduced and heat transfer coefficients may be improved by using heat exchangers similar to those used in this investigation.

Two types of heat exchangers made from aluminum sheets 0.007 inch in thickness have been fabricated and successfully tested at essentially atmospheric pressure with water. The cost of such exchangers should be considerably less than conventional heat exchangers.

A plate type heat exchanger with flat aluminum sheets separated by aluminum screens appears to be the most promising type of exchanger studied with respect to both cost and heat transfer coefficient.

VI. RECOMMENDATIONS

(1). For desalination by evaporation through porous membranes, further studies are needed to determine the suitability of thin aluminum alloy sheets with respect to corrosion. The most promising alloys should also be determined.

(2). Studies should be made on the effect of screen character on heat transfer coefficient in the type 4 exchanger and on geometry effects in type 1, 2, and 3 exchangers.

(3). The possibilities of using apparatus similar to those of this investigation for membrane evaporator-condensers should be tried.

VII. APPENDICES

Appendix A

Water Properties

This appendix includes the computer program and results of the approximations.

1). Density

$$\rho = 64.81 - 0.02212\, t - 60.15/t,$$

Average $\pm$ percent deviation	0.17
Standard deviation	0.15

2). Viscosity

$$\mu = -0.29 + 216.4/t - 2194/t^2$$

Average $\pm$ percent deviation	0.44
Standard deviation	0.01

3). Conductivity

$$k = 0.425 - 7.37/t + 127/t^2$$

Average $\pm$ percent deviation	0.39
Standard deviation	0.002

4). Specific heat

$$C_p = 0.979 + 0.00012\, t + 0.811/t$$

Average $\pm$ percent deviation	0.09
Standard deviation	0.001

5). Prandtl number

$$N_{Pr} = -1.15 + 624.5/t - 4658/t^2$$

Average $\pm$ percent deviation	1.70
Standard deviation	0.07

```
C       CNCS490F        LIN,C.T.
C   APPROXIMATION OF WATER PROPERTIES AS FUNCTION OF TEMP.
C        1.SPECIFIC HEAT
C        2.VISCOSITY
C        3.CONDUCTIVITY
C        4.DENSITY
C        5.PRANDTL NUMBER
      DOUBLE PRECISION T(20),X(20),PP(20),DP(20),PDP(20)
      DOUBLE PRECISION A(20,20),W(20,20),P(20)
      DOUBLE PRECISION AAPD(25)
      DOUBLE PRECISION S1,S2,DABS,SD,DSQRT,STD,STDD
      READ(1,100) N,ND
  100 FORMAT(2I5)
      READ(1,102) (T(I),I=1,ND)
      NP=N+1
      DO 1 IA=1,5
      WRITE(3,152) IA
  152 FORMAT(////,'PROPERTY',I5)
      READ(1,102) (P(J),J=1,ND)
  102 FORMAT(8D10.4)
C       APPROXIMATION
C        1.P=A+BT+C/T
C        2.P=A+BT+C/T**2
C        3.P=A+B/T+C/T**2
      DO 1 JA=1,3
      WRITE(3,153) JA
  153 FORMAT(//,5X,'APPRO.',I5)
      DO 54 K=1,ND
      W(1,K)=1
      W(4,K)=P(K)
      GO TO (51,52,53),JA
   51 W(2,K)=T(K)
      W(3,K)=1/T(K)
      GO TO 54
   52 W(2,K)=T(K)
      W(3,K)=1/T(K)**2.
      GO TO 54
   53 W(2,K)=1/T(K)
      W(3,K)=1/T(K)**2.
   54 CONTINUE
      DO 4 LL=1,N
      DO 4 MM=1,NP
    4 A(LL,MM)=0
      DO 5 I=1,N
      DO 5 J=1,NP
      DO 5 L=1,ND
      A(I,J)=A(I,J)+W(J,L)*W(I,L)
    5 CONTINUE
      CALL EMILY(A,N,NP,X)
      WRITE(3,18) (X(I),I=1,3)
   18 FORMAT(//,5X,'A=',D14.5,'     B=',D14.5,'     C=',D14.5)
      WRITE(3,151)
  151 FORMAT(//15X,'TEMP',6X,'TRUE P',6X,'APP.P',6X,'% ERR')
      S1=0
      S2=0
```

```
      DO 6 L=1,ND
      PP(L)=X(1)+X(2)*W(2,L)+X(3)*W(3,L)
      DP(L)=P(L)-PP(L)
      S1=S1+DP(L)*DP(L)
      PDP(L)=DP(L)*100./P(L)
      S2=S2+DABS(PDP(L))
      WRITE(3,150) T(L),P(L),PP(L),PDP(L)
  150 FORMAT(10X,4D12.4)
    6 CONTINUE
      STDD=S1/(ND-NP)
      STD=DSQRT(STDD)
      AAPD(JA)=S2/ND
      WRITE(3,154) AAPD(JA),STD
  154 FORMAT(/20X,'AV ER=',D12.4,/20X,'ST DE=',D12.4)
    1 CONTINUE
      STOP
      END
      SUBROUTINE EMILY(A,N,NP,X)
      DOUBLE PRECISION A(20,20),X(100)
      NN=N-1
      DO 920 I=1,NN
      DO 920 M=I,NN
      J=M+1
      IF(A(I,I) .EQ. 0.) GO TO 920
      R=A(J,I)/A(I,I)
      DO 925 L=1,NP
      A(J,L)=A(J,L)-A(I,L)*R
  925 CONTINUE
  920 CONTINUE
      J=N-1
      DO 940 I=1,J
      M=N-I
      L=NP-I
      SUM=0
      DO 955 K=L,N
      X(N)=A(N,NP)/A(N,N)
  955 SUM=SUM+X(K)*A(M,K)
  940 X(M)=(A(M,NP)-SUM)/A(M,M)
      RETURN
      END
/DATA
```

```
INPUT:  N NUMBER OF CONSTANT
        ND NUMBER OF DATA
        T(I) TEMPERATURES
        P(J) WATER PROPERTIES
OUTPUT: X(I) CONSTANTS; A,B,C
        STD STANDARD DEVIATION
```

Appendix B

Data and Results

Data taken throughout the investigation and the calculated overall heat transfer coefficients are listed in Table 1 through Table 4. A computer program used to convert e.m.f. and to calculate overall heat transfer coefficients is also included.

```
C   CONVERSION OF EMF AND CALCULATION OF U
C       CNCH4ROE      LIN,C.T.
        DIMENSION EHR(160),ECR(160),EH(160),EC(160),EMF(5)
        DIMENSION TR(10),T(10),THA(100),TCA(100),DSH(100)
        DIMENSION DSC(100),VH(100),VC(100)
        DIMENSION CPH(100),CPC(100),U(100)
        READ(1,181) NOEX
  181   FORMAT(I5)
        WRITE(3,281) NOEX
  281   FORMAT(1H1,'****HEAT EXCHANGER NO.',I5)
        READ(1,11) ND,AT,ACH,ACC,DH,DC,HL
   11   FORMAT(I10,6F10.5)
        WRITE(3,500)
  500   FORMAT(//3X,'RT       EH         THI        THO        EC        TCI
     2    TCO       U',//)
        DO 1000 K=1,ND
        READ(1,12) EHR(K),ECR(K),(TR(I),I=1,5)
   12   FORMAT(7F10.5)
        DO 1001 J=2,5
        TRR=0.79+(TR(1)-20.)*0.04
        EMF(J)=TRR+TR(J)
        IF(EMF(J)-1.19) 101,101,102
  101   T(J)=EMF(J)*30./1.19
        GO TO 1011
  102   IF(EMF(J)-2.47) 111,111,112
  111   T(J)=30.+(EMF(J)-1.19)*30./1.28
        GO TO 1011
  112   T(J)=60.+(EMF(J)-2.47)*30./1.34
        GO TO 1011
 1011   CONTINUE
        T(J)=T(J)-2.*(T(J)-TR(1))/(100.-TR(1))
 1001   CONTINUE
        A1=1.002
        A2=1.04E-4
        A3=3.297E-6
        EH(K)=EHR(K)*(A1-A2*T(3)-A3*(T(3)**2.))*60./453.6
        EC(K)=ECR(K)*(A1-A2*T(5)-A3*(T(5)**2.))*60./453.6
C   TRANSFER C DEG TO F DEG
        DO 140 J=2,5
        T(J)=32.+T(J)*1.8
  140   CONTINUE
        THA(K)=(T(2)+T(3))*0.5
        TCA(K)=(T(4)+T(5))*0.5
C        THE DEPENDENCE OF WATER PROPERTIES ON TEMPERATURE
        DS(Z)=64.81-0.022115*Z-69.146/Z
        CP(Z)=0.97788+0.0001192*Z+0.8112/Z
        Z=THA(K)
        DSH(K)=DS(Z)
        CPH(K)=CP(Z)
        VH(K)=EH(K)/(DSH(K)*ACH)
        DTH=T(2)-T(3)
        Z=TCA(K)
        DSC(K)=DS(Z)
        CPC(K)=CP(Z)
        DTC=T(5)-T(4)
```

```
      VC(K)=FC(K)/(DSC(K)*ACC)
      BTUH=FH(K)*CPH(K)*DTH
      BTUC=FC(K)*CPC(K)*DTC
      DBTU=BTUH-BTUC
      DT1=T(2)-T(5)
      DT2=T(3)-T(4)
      DTLM=(DT1-DT2)/ALOG(DT1/DT2)
      U(K)=BTUH/(AT*DTLM)
      WRITE(3,50) TR(1),FH(K),T(2),T(3),FC(K),T(4),T(5),U(K)
   50 FORMAT(8F8.1)
 1000 CONTINUE
      STOP
      END
/DATA
```

INPUT: NOEX NUMBER OF HEAT EXCHANGER
ND NUMBER OF DATA
AT AREA FOR HEAT TRANSFER
ACH,DH CROSS SECT. AREA AND EQ. DIAMETER OF INNER CHANNEL SQFT,FT
ACC,DC CROSS SECT. AREA AND EQ. DIAMETER OF OUTSIDE CHANNEL SQFT,FT
HL LENGHT OF CHANNEL FT
FHR(K),FCR HOT FLOW RATE AND COLD FLOW RATE ML/MIN
TR(1) ROOM TEMPERATURE °C
TR(2),TR(3) E.M.F. INLET AND EXIT INNER CHANNEL mv
TR(4),TR(5) E.M.F. INLET AND EXIT OUTSIDE CHANNEL

OUTPUT: FH(K),FC(K) FLOW RATES LB/HR
T(K) TEMPERATURES °F
U(K) OVERALL HEAT TRANSFER COEFF. BTU/HR.°F.FT^2

Table 1

Heat Exchanger I

Room Temp ^{o}C $TR(1)$	Hot Flow Rate lb/hr W_h	Hot Temp in ^{o}F t_{h1}	Hot Temp out ^{o}F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in ^{o}F t_{c1}	Cold Temp out ^{o}F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr ^{o}F ft^2 U
25.0	119.5	124.3	102.3	2620.8	151.8	55.9	74.1	2621.8	59.0
25.0	115.[illegible]	124.2	103.9	2333.6	143.0	56.4	73.3	2406.5	51.0
25.5	137.2	120.0	103.6	2244.2	139.9	64.8	80.6	2206.5	61.7
25.5	138.3	118.2	103.2	2199.1	141.0	64.4	79.3	2099.4	60.1
25.5	138.2	119.7	102.4	2389.9	140.1	60.7	78.3	2462.8	61.8
25.5	138.2	115.6	99.4	2232.3	140.1	62.5	79.1	2320.4	65.4
24.5	134.5	121.4	102.9	2481.5	152.1	62.7	78.3	2373.0	64.1
24.[illegible]	134.4	122.8	104.3	2485.9	152.1	62.5	79.3	2540.6	62.6
24.6	144.[illegible]	124.0	108.6	2209.2	98.1	64.7	88.4	2315.1	60.0
25.0	177.2	107.3	96.7	1865.9	140.5	62.4	75.7	1857.3	61.0
25.0	177.2	110.0	93.1	2105.2	140.5	62.1	76.8	2073.6	65.4
25.0	177.2	110.4	99.1	1995.9	140.5	63.2	77.8	2050.4	62.6
25.0	203.3	108.6	98.3	2083.3	164.1	59.0	72.1	2134.0	59.1
24.6	204.4	109.6	98.7	2211.6	161.8	60.5	74.0	2189.9	64.5
24.8	85.0	169.0	131.7	3174.7	76.2	66.3	106.2	3036.8	53.3
24.8	85.0	168.2	133.4	2963.5	76.3	55.5	105.7	3360.4	48.9
24.9	84.7	170.1	133.5	3107.4	78.0	68.1	110.2	3276.5	53.4
24.5	85.3	169.9	130.9	3329.4	85.0	67.6	106.6	3306.1	56.6
24.5	83.4	167.2	123.9	3612.5	103.0	61.2	94.9	3464.0	57.6
24.5	83.4	165.0	123.3	3480.9	103.0	62.1	94.8	3360.6	57.1
24.5	100.2	162.1	121.2	4103.5	118.1	61.7	94.8	3896.2	69.7
24.5	206.2	112.3	101.8	2164.3	175.5	61.3	74.5	2313.6	59.5
24.5	205.0	110.2	99.5	2176.5	177.1	60.9	74.0	2326.9	62.6
24.5	204.4	108.7	97.6	2245.5	192.7	59.8	72.0	2354.0	64.9
24.3	162.3	135.2	116.7	2992.0	187.2	54.8	81.7	3143.6	61.0
24.5	163.7	135.5	115.9	3204.5	187.2	63.4	81.3	3333.5	64.6
24.5	163.8	132.3	114.0	2997.1	187.3	64.3	79.7	2872.2	53.0
25.0	74.0	153.4	119.9	2480.7	118.5	69.6	90.5	2478.6	47.3
24.8	125.4	150.6	126.0	3086.5	118.4	69.5	95.7	3092.5	59.5
24.8	90.6	164.2	128.1	3278.7	118.3	70.0	98.1	3314.4	56.9
24.8	90.9	162.9	128.0	3170.5	118.3	69.5	96.8	3217.9	54.7
24.8	138.2	160.1	132.2	3858.7	120.3	69.7	103.1	4013.6	69.5
24.8	140.9	157.7	132.4	3570.8	119.8	70.5	99.7	3485.1	64.1
25.0	211.1	154.1	135.0	4033.7	119.5	69.9	104.2	4085.7	75.9
25.0	174.0	155.5	135.1	3548.9	116.9	69.2	99.9	3579.7	63.0

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft² U
27.0	86.2	167.5	128.8	3336.0	114.8	73.8	103.0	3348.5	50.2
27.0	233.3	172.9	153.9	4445.1	110.1	73.0	113.5	4456.3	63.7
27.0	86.9	159.1	128.9	2635.3	69.6	71.2	103.5	2242.1	50.0
27.0	89.0	162.9	130.8	2854.1	75.6	70.6	108.1	2930.5	53.4
27.0	86.1	163.4	131.6	2745.3	76.5	72.9	109.2	2773.8	52.3
27.0	146.3	162.8	143.0	2888.1	75.3	75.1	114.0	2921.9	53.7
27.0	141.4	174.0	149.1	3525.6	75.8	76.8	124.3	3595.9	62.9
27.3	141.3	177.8	150.8	3823.9	75.7	78.6	127.9	3725.7	68.0
26.3	175.4	165.4	145.8	3437.0	75.8	77.5	124.3	3539.9	59.1
26.3	175.3	166.4	147.1	3380.9	75.2	77.3	121.6	3324.8	64.5
27.0	171.3	169.7	149.9	3397.2	74.5	77.5	121.7	3288.2	61.5
26.0	211.2	163.6	146.3	3658.2	75.8	75.2	123.2	3627.8	72.3
26.3	206.5	164.5	146.7	3620.2	77.1	76.7	125.4	3746.3	74.8
26.0	209.3	162.5	144.4	3804.6	77.1	77.0	123.8	3601.5	79.0
25.0	268.8	162.0	147.7	3843.8	76.8	74.3	122.1	3656.3	75.2
24.8	269.5	155.7	139.2	4433.0	150.3	70.2	100.4	4516.7	77.0
24.8	273.5	154.4	138.0	4502.4	149.8	69.2	99.3	4500.7	79.4
24.8	267.1	154.6	136.5	4833.3	208.0	67.8	91.2	4844.7	78.7
24.8	254.8	155.7	136.7	4829.9	198.8	67.3	90.8	4654.1	77.4
24.5	254.9	153.2	135.3	4630.9	197.5	66.2	90.1	4712.5	75.5
24.5	258.1	152.0	133.6	4867.9	245.1	65.1	85.2	4914.1	77.1
24.5	260.1	152.3	133.5	4885.4	250.9	66.3	86.4	5030.4	73.9
24.0	201.4	158.3	133.2	5045.3	250.9	66.4	86.2	4978.5	73.2
23.0	159.7	161.5	130.8	4632.3	251.1	65.3	83.5	4539.0	69.6
23.0	146.8	162.8	133.2	4794.0	251.0	64.3	83.9	4904.0	71.4
23.8	312.1	157.0	143.0	4659.2	133.8	70.8	106.9	4925.0	81.6
24.0	312.1	159.2	143.3	4977.5	133.8	70.5	116.6	4821.6	85.1
24.0	325.6	158.7	142.6	4922.5	133.8	71.1	106.6	4733.2	86.4
24.5	292.7	160.2	141.1	5599.7	210.5	68.1	94.3	5503.4	86.8
24.8	293.2	155.8	135.3	6625.0	266.6	65.2	88.0	6045.8	85.2
24.8	301.1	153.8	134.3	5869.5	346.7	65.3	82.5	5942.7	90.0
25.0	307.7	153.1	133.6	5993.3	352.0	64.6	81.5	5909.6	91.7
25.0	402.8	149.9	134.2	6333.7	289.8	65.4	86.6	6136.6	103.1
25.0	404.5	145.9	130.1	6385.9	408.8	64.0	79.1	6179.7	103.3
25.0	508.9	145.6	130.6	7622.4	408.7	62.8	81.1	7474.4	124.0
25.0	771.7	115.6	106.1	7307.9	685.3	60.5	71.1	7249.8	174.5
25.0	771.6	116.1	106.5	7370.7	705.0	60.7	71.4	7521.1	175.3

Table 2

Heat Exchanger II

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft^2 U
25.0	381.4	101.4	94.8	2516.2	402.7	64.3	70.7	2555.2	112.5
25.0	381.5	101.3	94.5	2594.5	402.8	62.5	68.9	2592.6	110.6
22.7	411.4	85.3	80.8	1850.2	574.9	60.6	63.7	1826.7	121.4
22.7	411.3	86.9	81.9	2065.5	574.9	60.3	63.9	2029.6	127.2
23.0	401.0	105.4	96.4	3534.5	574.7	60.6	66.7	3499.1	133.8
23.0	401.2	102.4	94.0	3339.6	574.7	60.3	66.0	3271.3	130.8
23.5	412.8	125.8	112.3	5540.0	574.5	60.1	69.8	5549.1	140.6
23.5	365.6	150.6	127.3	7132.2	574.1	60.8	73.6	7343.1	136.7
23.5	339.4	151.0	128.5	7624.1	574.1	60.9	74.4	7772.4	145.3
23.5	398.6	144.8	124.5	8097.6	760.5	60.1	70.4	7850.2	150.2
23.5	392.0	144.5	124.6	7777.7	760.6	59.8	70.2	7917.6	153.6
23.0	500.0	101.0	93.2	3894.8	753.3	59.1	64.4	4022.0	151.2
23.0	499.0	101.2	93.5	3833.2	753.3	59.1	64.4	4022.0	147.8
27.2	190.3	191.8	156.3	6429.4	219.5	75.8	103.7	6131.0	104.7
27.2	180.5	186.7	152.4	6200.3	219.7	73.6	100.9	5979.1	103.4
27.5	227.9	187.4	149.4	8697.1	406.7	67.7	87.9	8225.3	132.0
27.5	189.0	188.3	148.4	7576.5	406.9	66.0	84.3	7433.8	112.0
27.5	222.1	182.3	142.8	8793.1	589.6	62.4	77.8	9112.2	131.2
29.2	211.7	184.4	143.9	8893.0	589.5	63.6	78.8	8900.0	131.5
29.8	232.7	179.5	139.7	9279.0	781.4	63.1	75.5	9629.0	142.1
29.2	332.0	178.3	153.0	8456.6	383.9	68.0	89.7	8334.3	133.7
29.0	163.7	125.3	109.5	2580.2	238.0	69.6	80.8	2648.9	83.9
29.0	163.6	129.8	112.1	2894.2	238.0	69.0	81.1	2879.2	86.6
29.0	162.5	173.1	137.5	5795.6	237.6	67.5	90.8	5515.5	104.6
29.0	162.6	172.2	136.7	5783.3	237.5	67.9	91.8	5673.7	105.5
29.0	162.5	173.5	137.5	5859.5	237.5	67.8	91.8	5694.6	106.4
29.0	162.5	174.2	138.0	5887.2	237.5	67.7	91.9	5724.4	106.1
29.0	163.6	138.0	111.7	4302.5	422.5	61.1	71.5	4388.4	101.4
29.0	163.6	131.3	110.7	3364.2	422.1	59.2	77.6	3528.7	97.5
23.5	213.5	91.7	84.2	1598.6	422.9	62.0	65.8	1585.3	90.8
23.0	213.6	88.0	81.5	1375.4	422.8	62.0	65.3	1399.2	80.7
22.9	213.7	94.7	79.2	1385.0	621.2	60.0	63.1	1370.7	98.0

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft² U
22.7	213.7	82.1	77.7	1356.9	621.2	60.5	62.7	1370.8	95.7
22.7	213.7	84.0	77.6	1375.8	621.2	60.5	62.6	1316.0	98.3
22.7	212.9	114.3	98.3	3406.5	621.1	59.3	65.0	3563.0	105.2
22.7	212.9	116.2	99.2	3607.1	621.0	59.9	66.0	3754.1	111.1
22.7	212.3	136.5	112.3	5143.9	620.7	60.3	69.0	5368.0	119.8
22.7	212.2	140.7	114.3	5606.6	620.7	60.2	69.3	5696.4	123.4
22.7	211.9	153.0	121.4	6698.3	620.6	60.2	71.1	6762.5	129.3
23.0	209.2	154.3	122.5	6652.7	620.5	60.4	71.4	6816.1	125.8
23.0	209.3	155.0	12[illegible].3	7276.5	808.4	59.4	68.3	7158.5	135.6
23.0	209.4	153.5	119.2	7180.8	808.4	58.9	67.7	7133.8	136.2
23.2	209.8	138.2	110.1	5891.6	808.6	58.8	66.1	5851.0	132.2
23.2	210.5	110.1	93.4	3504.5	808.9	59.0	63.4	3497.9	119.5
23.2	210.5	109.3	92.8	3461.6	808.9	58.9	63.2	3425.6	119.8
23.2	211.0	87.4	79.8	1806.0	809.1	59.1	61.3	1785.2	108.9
23.2	211.1	84.9	77.3	1600.6	809.1	59.3	61.2	1571.0	106.1
22.0	78.9	119.4	97.8	1702.4	184.6	70.2	79.9	1783.4	70.3
22.3	78.9	120.4	98.4	1734.5	184.6	70.4	80.1	1775.0	70.5
22.3	78.9	120.6	98.4	1744.2	184.6	70.0	79.8	1807.6	69.9
23.5	97.9	170.9	127.7	4232.5	181.6	69.4	93.0	4270.7	85.8
23.7	98.0	168.3	125.7	4173.1	181.6	69.4	91.5	4019.1	86.7
23.7	95.6	168.8	117.5	4910.8	382.5	56.3	78.8	4755.2	97.9
23.7	95.6	167.1	116.9	4804.7	382.5	65.5	77.7	4671.7	96.0
24.0	87.9	147.6	105.8	3672.8	382.7	64.8	74.3	3611.0	90.7
24.0	87.9	147.4	105.6	3676.9	382.7	64.7	74.2	3628.0	90.8
24.0	88.2	114.6	91.1	2075.1	382.9	64.7	70.3	2161.3	82.4
24.0	88.2	117.8	92.2	2255.7	382.9	64.7	70.3	2144.4	84.7
23.5	86.9	116.8	88.6	2440.9	561.6	60.9	65.4	2526.9	87.3
23.5	86.9	115.0	89.2	2327.0	561.5	62.0	66.1	2328.3	87.7
23.5	86.9	114.0	87.6	2295.1	561.5	62.0	65.7	2080.8	87.9
23.8	99.3	141.2	101.2	3976.5	575.8	62.2	69.2	4012.3	101.4
23.6	97.2	141.0	100.3	3953.9	575.9	61.7	68.4	3860.8	100.7
23.5	97.0	165.4	111.3	5242.4	575.7	62.0	71.3	5331.5	103.8
23.5	86.6	164.3	107.3	4992.6	581.0	62.1	70.8	5022.2	100.6
23.5	86.6	166.4	109.1	4958.1	581.0	62.3	71.0	5073.2	99.7
23.5	382.2	85.3	82.9	909.9	198.0	67.2	72.0	951.8	86.4
23.5	382.2	86.6	84.0	969.1	198.0	67.2	71.8	916.9	84.3
24.0	380.8	110.0	103.5	2435.7	197.8	68.2	80.2	2380.2	102.9
24.0	380.0	108.5	102.3	2358.1	197.8	68.0	79.7	2293.2	102.7
24.5	379.2	132.7	122.3	3937.7	197.5	68.9	88.6	3874.7	111.1
24.8	376.9	161.5	145.1	6173.4	197.1	69.0	100.2	6125.3	123.7
24.8	376.8	162.4	146.1	6157.3	197.1	68.7	100.0	6145.3	121.3
24.8	377.9	153.9	135.0	7147.9	402.0	65.8	83.6	7133.3	140.6
25.0	377.9	154.1	134.9	7267.7	402.0	65.4	83.6	7261.8	142.4
25.0	379.7	122.2	116.6	4798.0	402.4	64.3	76.7	4983.1	125.5
25.0	379.7	120.3	116.3	4938.4	402.4	64.3	76.7	4983.0	129.5

Table 3

Heat Exchanger III

Room Temp °C	Hot Flow Rate lb/hr	Hot Temp in °F	Hot Temp out °F	Enthal Change hot BTU/hr	Cold Flow Rate lb/hr	Cold Temp in °F	Cold Temp out °F	Enthal Change cold BTU/hr	H.T. Coeff BTU/hr °F ft^2
TR(1)	W_h	t_{h1}	t_{h2}	Q_h	W_c	t_{c1}	t_{c2}	Q_c	U
22.5	130.5	162.2	129.8	4233.7	177.7	68.1	91.2	4086.3	75.5
23.2	139.8	123.7	105.3	3958.8	374.8	64.5	74.7	3834.6	94.7
23.2	201.7	136.9	114.6	4506.0	374.6	66.5	78.8	4608.0	100.4
23.[illegible]	202.4	112.4	97.6	2980.2	374.9	63.8	71.7	2975.9	95.1
23.[illegible]	202.5	111.8	97.2	2964.4	374.9	64.0	71.7	2894.1	95.8
21.7	154.0	106.7	91.2	2381.5	515.2	62.8	67.3	2319.3	83.8
21.7	154.1	104.5	88.7	2439.3	515.2	62.2	66.9	2410.5	90.8
21.7	137.9	134.1	103.3	4240.5	515.0	62.1	70.7	4431.6	97.1
22.5	137.8	137.6	105.5	4408.0	514.9	63.4	72.1	4497.3	98.3
23.2	204.2	152.9	117.4	7243.7	712.9	62.1	72.2	7200.4	127.2
23.2	204.3	148.6	114.8	6901.5	713.1	60.6	70.4	6982.7	124.5
23.5	204.6	139.1	108.4	6283.8	713.1	60.9	69.5	5195.6	128.2
23.5	204.6	139.3	108.7	6248.3	713.1	61.1	69.7	6164.9	127.4
23.5	192.1	110.9	91.9	3631.6	697.7	59.9	65.2	3693.5	111.6
23.5	192.1	110.9	92.7	3481.5	697.6	61.4	66.5	3569.4	109.7
23.3	192.6	84.4	76.8	1460.5	697.8	62.2	64.3	1477.6	97.8
19.8	350.6	86.9	82.9	1344.6	206.0	63.8	70.1	1291.7	83.6
19.8	350.8	82.4	79.6	1096.4	206.0	64.8	69.6	982.5	86.3
19.5	420.3	108.1	101.8	2640.7	208.4	65.7	78.3	2613.0	94.9
19.7	320.4	135.0	121.9	4188.9	208.1	68.5	88.4	4174.8	97.0
19.7	214.6	135.6	118.6	3644.2	208.1	68.5	86.0	3622.0	85.4
21.5	325.3	158.1	140.5	5729.8	227.7	71.6	98.8	5628.5	105.7
21.5	325.2	159.3	141.5	5792.7	227.7	70.8	98.6	5759.5	104.4
21.7	328.8	151.0	130.2	6848.6	384.9	66.4	84.5	6959.1	124.0
22.0	329.8	151.0	130.1	6888.6	384.8	65.6	83.6	6925.5	123.3
21.2	471.8	122.5	112.1	4902.2	385.1	66.9	79.0	4978.8	129.2
21.5	330.3	125.7	112.4	4382.7	385.1	66.9	78.7	4553.7	111.8
21.5	330.2	127.5	113.5	4624.1	395.7	66.8	78.6	4678.5	114.1
21.5	331.3	106.6	97.9	2854.2	395.9	66.7	74.1	2935.0	105.7
21.5	331.2	106.8	98.0	2895.0	396.0	66.4	73.7	2865.4	105.6
22.5	308.5	85.4	81.7	1129.1	396.1	68.3	71.1	1083.3	96.2
19.5	366.8	79.5	75.9	1295.4	558.2	63.3	65.5	1254.8	114.9
19.5	319.3	78.8	75.4	1099.5	558.2	63.5	65.4	1028.9	102.9
22.0	318.3	105.6	95.2	3305.3	567.9	63.7	69.4	3259.7	115.6
22.[illegible]	318.3	106.0	95.1	3449.1	567.9	63.3	69.1	3335.2	118.7
22.0	318.4	104.1	93.9	3255.7	567.9	62.9	68.5	3210.1	115.8
22.2	317.2	128.3	111.1	5448.3	567.6	63.3	73.0	5487.8	125.0
22.2	317.3	127.6	110.6	5396.6	567.6	63.3	72.9	5412.8	125.1
21.2	316.2	150.4	125.2	7960.2	567.2	64.2	77.9	7758.9	141.1

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft^2 U
21.2	316.3	149.[illegible]	124.5	7760.3	567.2	63.9	77.5	7684.4	139.1
21.2	317.4	125.9	109.2	5278.7	567.6	63.6	73.3	5510.7	127.3
21.2	317.4	125.0	108.7	5161.7	567.6	63.6	73.0	5360.7	125.6
21.5	316.5	148.2	121.5	8452.9	799.9	62.3	73.1	8578.1	149.3
21.5	315.7	144.6	119.2	8040.2	800.0	62.3	72.4	8084.8	147.7
21.5	317.8	118.7	102.6	5113.8	800.4	62.6	68.8	4980.9	134.7
21.5	317.9	118.1	102.1	5088.1	800.4	62.5	68.8	5016.3	135.5
21.5	319.6	98.7	88.8	3149.9	800.6	62.5	66.6	3286.5	127.8
21.5	319.5	74.8	71.6	1014.9	800.9	62.6	63.9	1025.3	12[illegible].5
21.5	319.5	74.7	71.6	986.7	800.9	62.6	63.9	989.9	117.4
21.5	608.9	84.5	82.7	1047.6	206.4	68.3	73.6	1001.9	100.2
21.7	521.7	106.6	101.8	2504.6	206.3	67.1	79.2	2475.1	95.6
23.2	516.7	158.7	146.4	6347.5	205.5	72.5	100.1	6072.6	112.1
23.5	515.7	160.7	148.3	6445.8	205.5	70.9	101.3	6216.5	111.9
23.6	517.8	149.4	134.1	7882.3	383.3	65.7	86.1	7795.7	141.4
23.6	517.9	148.6	133.4	7921.9	383.5	64.3	84.2	7667.6	140.1
23.6	52[illegible].3	124.8	114.9	5124.2	383.7	66.2	79.7	5160.1	12[illegible].1
23.5	522.1	103.7	98.0	2954.1	384.0	66.7	74.8	3098.6	115.0
23.6	522.1	104.7	98.4	3296.4	384.0	66.4	74.8	3200.2	125.8
23.3	524.2	76.3	74.5	924.4	384.4	65.0	67.4	898.6	118.5
23.3	524.2	76.7	75.0	855.0	384.4	65.8	68.1	847.7	113.4
23.3	524.1	78.8	76.2	1363.2	596.1	64.1	66.4	1393.6	131.0
22.8	524.1	78.5	75.7	1432.9	596.2	63.2	65.6	1446.7	133.1
22.5	522.3	104.6	96.4	4284.1	595.9	61.8	69.1	4338.2	144.2
22.5	522.4	103.6	95.6	4156.0	595.9	62.0	69.0	4206.8	143.9
22.5	520.6	124.8	112.1	6625.0	595.7	60.5	71.9	6754.3	149.6
22.5	520.6	125.0	112.2	6667.7	595.6	61.0	72.4	6753.5	151.8
21.6	522.6	78.2	74.2	2121.1	809.1	59.0	61.4	1929.1	156.5
21.6	522.7	76.6	73.1	1867.9	809.0	59.4	61.6	1750.4	153.5
21.6	522.7	76.2	72.8	1729.6	809.0	59.5	61.5	1607.5	145.2
22.6	521.9	102.7	93.5	4742.9	808.6	60.2	66.1	4745.7	160.2
24.2	819.4	112.4	103.2	7525.8	808.2	60.8	70.1	7519.3	229.9
24.0	147.7	190.0	154.2	5303.3	205.0	71.9	98.4	5425.9	72.1
26.0	253.2	188.7	144.[illegible]	9097.5	570.3	65.[illegible]	81.6	9420.5	115.3
26.5	253.4	189.3	14[illegible].7	9918.6	787.4	65.9	78.4	9814.4	127.7
26.7	387.4	170.[illegible]	139.4	11893.0	787.3	64.5	79.6	11857.6	170.3
27.2	386.4	178.5	149.5	11640.8	594.4	65.9	85.8	11226.5	157.8
27.4	386.1	188.4	151.[illegible]	11346.9	594.4	66.7	85.9	11382.6	143.9
27.5	353.0	190.5	161.6	10243.0	384.7	67.7	94.5	10284.0	127.3
28.0	341.9	192.5	163.3	10016.1	384.7	69.1	95.6	10147.3	123.8
28.0	351.1	192.9	170.6	7858.7	201.7	70.3	109.4	7871.5	101.3
28.[illegible]	311.7	197.5	172.3	7739.0	201.7	71.7	110.2	7742.1	97.2
28.3	311.7	196.7	172.3	7628.4	201.6	72.8	111.4	7766.5	97.6

Table 4

<u>Heat Exchanger IV</u>

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft^2 U	Press Drop hot in Hg DPH	Press Drop cold in Hg DPC
22.0	30.1	173.0	86.6	2299.9	28.8	76.0	154.9	2267.6	372.9	3.4	4.0
23.0	30.1	183.7	99.3	2542.7	28.7	77.2	162.8	2453.6	361.9	3.4	4.0
23.5	29.3	197.0	102.5	2768.3	28.3	77.7	173.1	2703.2	351.3	3.1	3.6
23.5	29.3	195.2	101.7	2743.1	28.3	77.7	172.5	2685.0	363.1	3.1	3.6
25.0	29.0	140.6	91.8	1408.7	28.7	77.5	127.5	1434.1	331.9	4.2	3.6
25.0	29.0	140.2	91.4	1411.6	28.7	77.2	127.2	1432.5	333.0	4.2	3.6
25.0	29.0	109.4	86.3	667.1	28.9	78.4	102.4	691.8	292.9	4.4	3.8
25.0	29.0	107.8	85.8	635.1	28.9	78.2	101.6	659.3	289.1	4.4	3.8
26.3	67.5	98.5	84.1	966.6	67.0	74.7	90.0	1026.4	361.5	6.1	6.3
26.3	67.5	99.7	84.5	1022.9	67.0	74.8	90.7	1061.1	362.1	6.1	6.3
26.3	67.5	100.3	84.6	1052.6	67.0	74.7	91.1	1091.6	362.2	6.1	6.3
26.5	66.4	142.7	98.4	2936.5	65.9	76.2	120.8	2947.6	421.3	5.7	6.1
26.5	66.4	145.0	99.2	3040.6	65.9	76.2	122.4	3037.6	420.9	5.7	6.1
28.0	67.1	191.3	112.8	5273.0	64.2	78.6	158.2	5104.9	479.9	5.2	5.6
28.0	67.1	192.3	112.3	5375.2	64.2	78.3	157.3	5065.3	464.5	5.2	5.6
28.0	67.1	192.1	112.4	5356.4	64.2	78.6	156.6	5132.1	473.8	5.2	5.6
23.8	158.3	83.4	75.9	1193.5	157.0	68.7	75.6	1073.0	455.0	8.2	8.6
23.8	158.3	81.8	75.2	1040.0	157.0	68.7	75.0	990.0	466.5	8.2	8.6
25.0	158.6	109.6	87.5	3502.0	157.2	66.8	87.8	3287.9	489.6	7.9	8.4
25.0	158.6	108.1	86.7	3391.9	157.3	66.0	87.2	3329.2	515.5	7.9	8.4
27.0	160.5	120.8	93.0	4457.6	153.6	67.1	94.8	4251.6	512.4	7.2	8.2
27.0	160.5	123.1	93.9	4687.3	153.6	67.6	95.9	4345.1	514.8	7.2	8.2
28.0	155.3	138.0	98.9	6220.1	157.6	67.0	101.6	5784.2	529.0	6.8	7.5
28.0	155.3	137.2	98.2	6044.1	157.6	66.1	102.1	5920.2	561.9	6.8	7.5
29.0	152.0	138.0	112.7	3976.8	65.3	68.8	127.3	3813.2	497.3	4.3	5.6
29.0	152.0	138.2	112.2	3948.9	65.3	68.5	126.7	3793.5	497.1	4.3	5.6
29.0	151.7	137.5	121.6	2412.2	36.2	68.8	135.2	2406.5	479.7	3.7	5.0
29.0	151.7	137.0	121.8	2381.2	35.7	68.9	134.9	2389.6	485.5	3.7	5.0
30.0	178.9	108.1	100.8	1310.1	37.4	68.8	107.3	1434.7	527.4	3.5	5.1
30.0	178.9	110.0	101.9	1456.3	37.4	68.8	108.5	1491.7	451.7	3.5	5.1
30.0	176.7	111.2	97.4	2421.1	68.3	68.5	103.6	2399.5	474.3	4.2	6.0

Room Temp °C TR(1)	Hot Flow Rate lb/hr W_h	Hot Temp in °F t_{h1}	Hot Temp out °F t_{h2}	Enthal Change hot BTU/hr Q_h	Cold Flow Rate lb/hr W_c	Cold Temp in °F t_{c1}	Cold Temp out °F t_{c2}	Enthal Change cold BTU/hr Q_c	H.T. Coeff BTU/hr °F ft^2 U	Press Drop hot in Hg DPH	Press Drop cold in Hg DPC
[illegible]	176.7	111.5	97.6	2457.2	68.3	68.4	103.3	2375.1	454.1	4.2	6.0
23.0	75.5	130.5	110.5	1503.2	31.3	82.6	126.9	1384.9	370.4	4.3	4.7
23.0	75.5	132.1	111.6	1549.4	31.3	81.8	128.1	1452.5	357.9	4.3	4.7
23.2	75.3	98.7	91.7	520.4	31.4	80.4	96.7	509.5	302.1	4.4	4.9
23.2	75.3	99.6	92.1	523.4	31.4	80.0	97.5	547.7	337.7	4.4	4.9
23.2	76.3	99.0	80.8	1395.8	135.2	73.5	83.2	1302.7	374.8	6.9	7.6
23.2	76.3	98.6	80.7	1367.7	135.2	73.6	83.1	1290.8	379.8	6.9	7.6
[illegible]	77.5	153.0	91.9	4728.9	136.0	72.7	103.3	4160.5	410.8	6.5	6.7
[illegible]	77.5	156.[illegible]	91.6	4551.1	136.1	72.7	102.3	4027.8	459.[illegible]	6.5	6.7
23.0	74.3	181.8	135.5	3448.8	31.2	74.4	174.3	3112.4	395.7	4.3	4.5
23.0	74.3	181.1	135.1	3420.0	31.2	73.9	174.2	3123.9	397.1	4.3	4.5
23.0	76.1	181.2	98.2	6314.9	139.3	74.6	117.5	5956.6	466.9	6.5	6.7
[illegible]	76.1	182.5	97.9	6292.3	139.3	74.0	115.9	5821.6	450.5	6.5	6.7
29.0	30.3	199.9	79.5	3654.5	147.2	74.6	97.3	3333.3	328.4	2.7	3.5
23.0	30.3	199.9	79.9	3639.8	147.2	74.7	97.1	3296.4	318.2	2.7	3.5
23.0	29.6	200.5	85.6	3407.9	74.2	75.3	117.6	3134.0	284.7	2.4	3.2
23.0	29.6	201.2	85.9	3422.5	74.2	75.8	117.9	3122.5	285.3	2.4	3.2
28.5	28.7	155.2	82.4	2087.8	73.5	75.6	102.3	1960.8	275.8	2.7	3.3
23.5	28.7	156.1	82.4	2116.3	73.6	75.4	102.0	1953.1	268.8	2.7	3.3
23.5	28.8	156.6	75.1	2339.5	141.2	71.0	86.9	2232.8	305.2	3.5	4.0
23.5	28.8	156.4	73.8	2375.8	141.2	69.7	85.7	2255.9	307.1	3.5	4.0
23.5	28.4	107.5	69.7	1069.6	138.6	67.5	74.9	1018.4	285.6	4.0	4.3
23.5	28.4	110.2	70.1	1137.3	138.6	67.8	75.5	1067.1	281.1	4.0	4.3

Appendix C

Computer Program for Least Squares Fitting

A computer program of non-linear method for least squares fitting in correlating heat transfer coefficients is listed in this appendix.

```
C      CNCH490F      LIN,C.T.
C   LEAST SQUARE FOR U CORRELATION FOR HEAT EXCHANGER
      DIMENSION TR(10),T(10),THA(160),TCA(160),DT(160)
      DIMENSION TW(100),VSC(100),PRH(100)
      DIMENSION DSH(160),DSC(160),CPH(160),CPC(160),VSH(160)
      DIMENSION CKH(160),CKC(160),U(160),REH(160),REC(160)
      DIMENSION PRC(160),HHH(160),HHC(160),HH(160),A(10,12)
      DIMENSION DTW(100),B(9),VH(100),VC(100)
      DIMENSION FHR(160),FCR(160),FH(160),FC(160),EMF(5)
      DIMENSION X(10),DTI(160),TWW(160),W(10,10)
      DIMENSION GZH(100),GZC(100)
      READ(1,181) NOEX
  181 FORMAT(I5)
      WRITE(3,281) NOEX
  281 FORMAT('****HEAT EXCHANGER NO.',I5)
      READ(1,11) ND,AT,ACH,ACC,DH,DC,HL
   11 FORMAT(I10,6F10.5)
      WRITE(3,500)
  500 FORMAT(/////3X,'FH        FC        VH        VC        THI
     5  THO       TCI       TCO       DTLM      QH        QC        DQ
     6     U',//)
      DO 1000 K=1,ND
      READ(1,12) FHR(K),FCR(K),(TR(I),I=1,5)
   12 FORMAT(7F10.5)
      DO 1001 J=2,5
      TRR=0.79+(TR(1)-20.)*0.04
      EMF(J)=TRR+TR(J)
      IF(EMF(J)-1.19) 101,101,102
  101 T(J)=EMF(J)*30./1.19
      GO TO 1011
  102 IF(EMF(J)-2.47) 111,111,112
  111 T(J)=30.+(EMF(J)-1.19)*30./1.28
      GO TO 1011
  112 T(J)=60.+(EMF(J)-2.47)*30./1.34
      GO TO 1011
 1011 CONTINUE
      T(J)=T(J)-2.*(T(J)-TR(1))/(100.-TR(1))
 1001 CONTINUE
      A1=1.002
      A2=1.04E-4
      A3=3.297E-6
      FC(K)=FCR(K)*(A1-A2*T(5)-A3*(T(5)**2.))*60./453.6
      FH(K)=FHR(K)*(A1-A2*T(3)-A3*(T(3)**2.))*60./453.6
C   TRANSFER C DEG TO F DEG
      DO 140 J=2,5
      T(J)=32.+T(J)*1.8
  140 CONTINUE
      THA(K)=(T(2)+T(3))*0.5
      TCA(K)=(T(4)+T(5))*0.5
      DT(K)=THA(K)-TCA(K)
      TW(K)=(THA(K)+TCA(K))*0.5
C      THE DEPENDENCE OF WATER PROPERTIES ON TEMPERATURE
      DS(Z)=64.81-0.022115*Z-60.146/Z
      CK(Z)=0.42529-7.3714/Z+127.33/Z**2
      CP(Z)=0.97788+0.0001192*Z+0.8112/Z
```

```
      VS(Z)=(-0.28776)+216.39/Z-2193.9/Z**2
      Z=THA(K)
      DSH(K)=DS(Z)
      CKH(K)=CK(Z)
      VSH(K)=VS(Z)
      CPH(K)=CP(Z)
      VH(K)=FH(K)/(DSH(K)*ACH)
      DTH=T(2)-T(3)
      Z=TCA(K)
      DSC(K)=DS(Z)
      CKC(K)=CK(Z)
      VSC(K)=VS(Z)
      CPC(K)=CP(Z)
      DTC=T(5)-T(4)
      VC(K)=FC(K)/(DSC(K)*ACC)
      BTUH=FH(K)*CPH(K)*DTH
      BTUC=FC(K)*CPC(K)*DTC
      DBTU=BTUH-BTUC
      DT1=T(2)-T(5)
      DT2=T(3)-T(4)
      DTLM=(DT1-DT2)/ALOG(DT1/DT2)
      U(K)=BTUH/(AT*DTLM)
      WRITE(3,502) FH(K),FC(K),VH(K),VC(K),(T(I),I=2,5),DTLM
     7,BTUH,BTUC,DBTU,U(K)
  502 FORMAT(13F8.1)
      PRH(K)=CPH(K)*VSH(K)/CKH(K)
      PRC(K)=CPC(K)*VSC(K)/CKC(K)
      REH(K)=DH*VH(K)*DSH(K)/VSH(K)
      REC(K)=DC*VC(K)*DSC(K)/VSC(K)
      GZH(K)=FH(K)*CPH(K)/(CKH(K)*HL)
      GZC(K)=FC(K)*CPC(K)/(CKC(K)*HL)
 1000 CONTINUE
      B(1)=2.0
      B(2)=2.
      B(3)=0.30
      N=3
      NP=N+1
      JD=1
      DO 60 JJ=1,20
      DO 1005 I=1,20
      DO 120 M=1,N
      DO 120 J=1,NP
  120 A(M,J)=0.
      DO 1111 L=1,ND
      Z=TW(L)
      VSW=VS(Z)
      WH=(VSH(L)/VSW)**0.14
      WC=(VSC(L)/VSW)**0.14
      RH=REH(L)
      RC=REC(L)
      PH=PRH(L)
      PC=PRC(L)
      HHC(L)=DC/(CKC(L)*WC*GZC(L)**B(3)*B(2))
      HHH(L)=DH/(CKH(L)*WH*GZH(L)**B(3)*B(1))
```

```
      HH(L)=HHH(L)+HHC(L)
      IF(JD .EQ. 30) GO TO 1111
      W(1,1)=-HHH(L)/B(1)
      W(1,2)=-HHC(L)/B(2)
      W(1,3)=-(HHH(L)*ALOG(GZH(L))+HHC(L)*ALOG(GZC(L)))
      W(1,4)=1/U(L)-HH(L)
      DO 1101 J=1,N
      DO 1101 K=1,NP
 1101 A(J,K)=W(1,J)*W(1,K)+A(J,K)
 1111 CONTINUE
      IF(JD .EQ. 30) GO TO 1004
      WRITE(3,490) (B(KK),KK=1,N)
  490 FORMAT(5X,'B(JJ)= ',3F10.4)
      CALL EMILY(A,N,NP,X)
      DO 1102 K=1,N
      DB=X(K)
      IF(K-2) 26,26,27
   26 IF(ABS(DB)-0.05) 1102,1102,1006
   27 IF(ABS(DB)-0.01) 1102,1102,1006
 1102 CONTINUE
      DO 88 KO=1,N
   88 B(KO)=B(KO)+X(KO)
      GO TO 1004
 1006 CONTINUE
      DO 1005 J=1,N
 1005 B(J)=B(J)+X(J)/2
      WRITE(3,131) (X(I),I=1,N)
  131 FORMAT(//,' FAIL TO CONVERAGE  ',/,' DELT B =',5F10.5)
      GO TO 3000
 1004 CONTINUE
      DO 1007 L=1,ND
      DTI(L)=DT(L)*HHH(L)/HH(L)
      TWW(L)=THA(L)-DTI(L)
      DTW(L)=TW(L)-TWW(L)
 1007 TW(L)=TWW(L)
      IF(JD .EQ. 30) GO TO 61
      DO 59 L=1,ND
      DWT=DTW(L)
      IF(ABS(DWT)-5.0) 59,59,60
   59 CONTINUE
      JD=30
   60 CONTINUE
      WRITE(3,132)
  132 FORMAT(//,' FAIL TO CONVERGE DUE TO DTW')
   61 CONTINUE
      WRITE(3,133) (B(I),I=1,N),(X(K),K=1,N)
  133 FORMAT(////,5X,'B =',3F10.5,/4X,'DB=',3F10.5)
      WRITE(3,134)
  134 FORMAT(/////,6X,'U           HI           HO           DU
     8  DTW        REH        REC       % DU',//)
      SPAE=0.
      SPE=0.
      DO 21 L=1,ND
      HI=1/HHH(L)
      HO=1/HHC(L)
```

```
      DU=U(L)-1/HH(L)
      PE=DU*100./U(L)
      SPAE=SPAE+ABS(PE)
      SPE=SPE+PE
      ZH=HI*DH/CKH(L)
      ZC=HO*DC/CKC(L)
      WRITE(3,135) U(L),HI,HO,DU,DTW(L),REH(L),REC(L),PE,GZH
     9(L),GZC(L),ZH,ZC
  135 FORMAT(8F10.2,4F10.1)
   21 CONTINUE
      APAE=SPAE/ND
      APE=SPE/ND
      WRITE(3,33) APE,APAE
   33 FORMAT(//5X,'AV % ERR',F10.2,/5X,'AV ABS % ER',F6.2)
 3000 CONTINUE
      STOP
      END
      SUBROUTINE EMILY(A,N,NP,X)
      DIMENSION A(10,12),X(10)
      NN=N-1
      DO 920 I=1,NN
      DO 920 M=I,NN
      J=M+1
      R=A(J,I)/A(I,I)
      DO 925 L=1,NP
      A(J,L)=A(J,L)-A(I,L)*R
  925 CONTINUE
  920 CONTINUE
      J=N-1
      DO 940 I=1,J
      M=N-I
      L=NP-I
      SUM=0.
      DO 955 K=L,N
      X(N)=A(N,NP)/A(N,N)
  955 SUM=SUM+X(K)*A(M,K)
  940 X(M)=(A(M,NP)-SUM)/A(M,M)
      RETURN
      END
/DATA
```

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