HEAT TRANSFER ANALYSIS OF A HEAT EXCHANGER PLATE

Thesis for the Degree of M. Sc.
MICHIGAN STATE UNIVERSITY
Syed Abdullah Hassan
1962

HEAT TRANSFER ANALYSIS OF A

HEAT EXCHANGER PLATE

By

Syed Abdullah Hassan

AN ABSTRACT

Submitted to the Colleges of Agriculture and Engineering
of Michigan State University of Agriculture and
Applied Science in partial fulfillment of
the requirements for the degree of

MASTER OF SCIENCE
IN
AGRICULTURAL ENGINEERING

Department of Agricultural Engineering

Approval Carl W. Hall, Feb 27, 1962

February 1962

ABSTRACT

Plate type heat exchangers are very widely used in the dairy and other food industries because they offer flexibility, high rates of heat transfer, compactness and sanitary operation.

The objective of this study was to investigate the variations of overall heat transfer coefficients and the temperatures over a single plate. Thirty-eight pairs of thermocouples were soldered onto the plate surface to obtain the data. Tests were conducted in the actual operating conditions, with the test plate in the regenerator section of a high-temperature short-time milk pasteurizer. The flow rate of the milk through the entire unit was kept nearly constant at an average value of 6920 lbs/hr.

Contours of the overall heat transfer coefficients and the temperatures were plotted over the plate area. The average local overall heat transfer coefficients were observed to vary between 300-800 BTU/hr ft^2 ^{o}F. The average value for the plate was about 550 BTU/hr ft^2 ^{o}F. The variations of both the overall heat transfer coefficient and the temperature were pronounced near the ports due to converging and diverging flows. These contours suggested

that the velocity profiles were nearly identical along the length of the plate.

The overall effectiveness of the regenerator unit was calculated and was found to be 79%. This value though slightly lower, is comparable to the reported value of 82%.

HEAT TRANSFER ANALYSIS OF A
HEAT EXCHANGER PLATE

By

Syed Abdullah Hassan

A THESIS

Submitted to the Colleges of Agriculture and Engineering
of Michigan State University of Agriculture and
Applied Science in partial fulfillment of
the requirements for the degree of

MASTER OF SCIENCE
IN
AGRICULTURAL ENGINEERING

Department of Agricultural Engineering

February 1962

ACKNOWLEDGMENTS

The author expresses his sincere gratitude to Dr. Carl W. Hall for his able guidance, timely advice and personal encouragement which made possible this study.

Grateful acknowledgement is due Dr. T. I, Hedrick for providing facilities of the Michigan State Dairy Plant, where this study was carried out. The advice and guidance of Dr. G. M. Trout, Food Science and Prof. D. J. Renwick of Mechanical Engineering is also appreciated. The author is indebted to Erland Kondrup, Paul Cooper and Peter Hansen of the dairy plant; James Cawood and his staff of the Agricultural Engineering Research Laboratory for their help and assistance in making these tests.

The writer is also thankful to Dr. Arthur W. Farrall, Head, Agricultural Engineering Department for providing funds and use of facilities of the department.

Sincere thanks are due Dr. and Mrs. E. O. Anderson for their moral support and interest extended to the author during their stay in Pakistan and here in Connecticut.

Personal appreciation is extended to wonderful friends like Ram Misra, Harris Gitlin, and Gad Hetsroni for valuable help at various stages.

Last but not least, deepest gratitude is expressed to my father, Syed Mohsin Bokhari, whose inspiration for knowledge influenced the writer in coming to this country to pursue higher education.

TABLE OF CONTENTS

LIST OF FIGURES

SUMMARY

One of the gasketed plates from the regenerator section of a milk pasteurizer was used as a test plate. Seventy-six thermocouples, forming thirty-eight sets, were soldered to the plate in a pre-determined configuration. Local temperature differences across the plate wall were measured by these sets. Moreover, four thermocouples, one in each port, were placed to measure inlet and outlet temperatures of the hot and cold milk streams.

The local values of the overall heat transfer coefficients were calculated for each of the twenty tests performed. The averaged local values were found to occur in the range 300-800 BTU/hr ft^2 ^oF. The average value for the plate was about 550 BTU/hr ft^2 ^oF.

Contours of the overall heat transfer coefficients and the wall temperatures were plotted over the plate area. From these contours it was observed that the wall temperatures remained fairly constant along the width of the plate. The wall temperature along the length of the plate varied in a straight line. Effect of local conduction of heat from the main hot milk stream to the plate was noticed near the hot milk inlet port. This resulted in higher temperatures in the vicinity of the port.

The overall heat transfer coefficients were observed to be higher in the middle of the plate than near the edge of the corrugations. Here again, effect of local conduction was observed. The overall heat transfer coefficients varied little along the length of the plate.

The contours of temperature and overall heat transfer coefficients suggested velocity profiles identical to those obtained by previous investigations.

INTRODUCTION

Since the beginning of time, mankind has been confronted with an inevitable need for heat energy, to keep warm and to process raw materials. In the early stages of life the needs were simple and few, allowing for the direct exposure of the heat source. As the need became varied and more intricate it became necessary to build a device which would allow the heat to pass from the source to the heated media by indirect methods, so that the undesirable gases of combustion could be avoided. This device is called a heat-exchanger.

There are numerous versions of heat-exchangers in use today. Most of these are connected, in one way or another, with the shell-tube type. Appreciable amount of research has been done with this type and a wealth of information is reported and available in any standard book on the subject.

In recent years the popularity of portable gas-turbine prime-movers enthused researchers to design a heat-exchanger which in addition to being light, should be compact and have high efficiency of performance. This was further backed by the standing demand of the dairy and other food industries for a flexible heat-exchange device, so that different heat-exchange surfaces could be obtained from the same equipment, depending upon the processing load.

Plate type heat-exchangers met the requirements for weight, compactness and flexibility. Despite their failure to qualify for use with the gas-turbine prime-movers, because of the limitation in standing high temperatures and pressures, they have been readily accepted in dairy and other food industries.

A plate heat-exchanger consists, essentially, of rectangular plates which are designed to form narrow rectangular flow passages when pressed together. Higher heat transfer efficiency is achieved by corrugations on the plate which cause turbulence at a markedly small Reynolds' number.

Although extensively used, surprisingly little work is reported with this type of exchanger. The main objective of the study was to obtain basic information on the overall heat transfer coefficients based on a single plate.

It is hoped that the findings of the investigation will prove helpful in more economical use of the plate type heat-exchangers.

Dr. Carl W. Hall

The author

Figure 1

OBJECTIVES

The objective of this study was to determine the local overall heat transfer coefficients on a single plate, and to plot their values along the entire heat-exchange surface. Underlying interest was twofold:

1. To investigate the effectiveness of the heat exchange area with the present design. This involved determining U-values and temperatures over the area of the plate.
2. To compare the results with the fluid flow analysis of previous investigators.

REVIEW OF LITERATURE

The earliest known patent on a device similar to the plate heat exchanger was obtained in Germany in 1878 (9).* However, the credit for the conventional design accepted and in use today is generally given to an Englishman, Dr. Richard Saligman, for his invention in 1923 (17).

A plate type heat-exchange unit consists of rectangular metallic plates assembled face to face. The plates are usually pressed out of stainless steel sheets, though materials such as Hastelloy C, titanium and cupronickel are also in use (9, 14).

The plates have corrugations called turbulence promoters formed on the surface, with the finer design details varying from one manufacturer to the other. These corrugations, in addition to creating turbulence at Reynolds' numbers as low as 180-200, add considerably to the plate strength (7, 9, 14). The high degree of turbulence gives a higher rate of heat transfer. Overall heat transfer coefficients in the range 600-750 are quite common (9, 11). The thickness of plate range between 0.05 inch to 0.125 inch, depending on the size of the plate.

*Number in parentheses refers to the references given at the end.

Alternate plates are provided with rubber gaskets which run all along the periphery of the plate. The gasket serves two purposes: first, it separates the adjoining plates, thus forming a flow passage having a predetermined width, usually 3-5 mm and, secondly, it prevents leakage of the flowing fluid. The rubber used for the gasket is treated to make the surface polar (4), which helps make a stronger bondage with stainless steel plate. The gaskets so far used cannot stand temperatures and pressures higher than 300^{o}F and 150 psig (4, 9, 14), which accounts for one of the major shortcomings of plate exchangers. Use of Telfon and some of the silicones should lead to improved gaskets that can be exposed to temperatures and pressures higher than the present maximum (14).

The plates are mounted on a metallic frame having guide bars on top and bottom. These are pressed together in position by a screw-plunger or hydraulic press. On the same frame two or more heat-exchange units can be mounted. In that case, a spacer-block or terminal is placed between the two units. In the dairy industry, for pasteurizing milk, it is customary to use three heat-exchange units assembled on one frame. The units are called the regenerator, heater and cooler.

Each plate has four holes (or ports), two at the top and two at the bottom. The gasket is designed to allow opening of only two holes, one on either end, on each side of the plate. These form inlet and outlet ports for the flowing fluids. The flow pattern of the flowing fluid depends entirely on the design of the gasket on each plate and the relative position of these plates in the unit. The fluids may have a single pass, a multiple pass or a divided pass through the unit. A combination of the last two is a more common practice.

The plate heat exchanger has gained tremendous popularity in many industries in this decade, because it provides:

1. High rate of heat transfer at relatively small velocities.
2. Flexibility in obtainable heat-exchange area.
3. Ease in cleaning, sterilization and inspection.
4. Lesser possibility of contamination, the unit being totally enclosed.
5. Adaptability of the unit to more than one operation at the same time.
6. Saving in space.

E. L. Watson _et_ _al_. (16) made extensive studies of velocity flow patterns and pressure drop for various flow rates using three sets of plates. Each set had a different design of corrugations.

Velocity profiles were determined by two methods: motion pictures, and electrical conductivity tests. Motion pictures were taken at flow rates of 1000, 1700 and 2500 lbs/hr per plate, of a dye solution as it displaced water in the space between the plates. These pictures were analyzed to obtain velocity profiles and locate air-pockets. The electrical conductivity tests were made with 0.1% sodium-chloride solution. Fastest and slowest particles, defined in relative terms, were determined as the conducting salt solution displaced tap water in the flow channels. Conductivity changes were recorded on a recording potentiometer. The results from these two methods were found to be in fair agreement.

Air pockets were found to exist in inverse ratio to the flow rates. The existence of air pockets was found to reduce useful heat-transfer area and cause greater pressure drop, It also indicated a greater possibility of cook-on or milkstone deposits which impair heat transfer and increase cleaning difficulty, a fact supported by V. Mennicke (12).

The pressure drop studies by the same authors showed that for the same overall heat transfer coefficient the plate exchanger gave smaller pressure drop compared to a shell-tube unit. It was also found that divided flow, although it gave a smaller pressure drop as against multiple flow, resulted in lower heat transfer coefficients because of lower velocities and higher probability of air accumulation.

A. A. McKillop and W. L. Dunkley, continuing the above study, investigated the heat transfer characteristics. Heat transfer coefficients were calculated by two methods: mathematical analysis and empirical correlation by McAdams (10).

The mathematical analysis was based on the energy balance of the system. Each flow configuration yielded several simultaneous equations which were solved with a computer. Plots of overall heat transfer coefficient versus flow rates were made for each of the three units under study. These values were found to be higher compared to ones calculated with McAdam's correlation. This difference was attributed to the turbulence promoters, since McAdam's equation predicts coefficients for clear-channel flow.

V. Mennicke (12) investigated the performance of the plate heat exchangers as related to:

1. Flow arrangement,
2. Fluid flow ratio,
3. Fluid velocities, and
4. Heat exchange area.

He observed that:

1. The efficiency was very nearly the same for a combination of parallel and counter flow situations having fewer parallel flows, as of an all-counter flow arrangement.
2. The heat transfer increased in proportion to the increase in flow ratio, up to a ratio of four, beyond which the increase was significantly small.
3. The heat transfer improved linearly with increase of fluid velocities. Film coefficient "h" was related to velocity, V, as $h = C\ V^n$, where C is a constant. For the two flow configurations studied, the value of exponent n was found to be 0.814 for divided path of flow and 0.736 for continuous path of flow, respectively.
4. The heat transfer area versus efficiency curve assumed asymptotic trend beyond an area of three square meters and an efficiency of 85%.

To the best knowledge of the author, no attempt has been made to study the variation of the overall coefficient of heat transfer on a single plate, which in fact has been assumed constant (11). The sole objective of this study was to measure and evaluate these variations.

PRESENTATION OF THE KEY EQUATION

The following assumptions were made for the analysis:

1. Steady state exists in the unit.
2. Specific heats of the flowing liquids are constant.
3. Heat conduction along the plate surface is negligible.
4. Bulk temperature change of both hot and cold milk stream occurs in a straight line from inlet to outlet ports of the plate.
5. The velocities are nearly constant over a cross-section of the plate along the length.

The test plate, at position "17" in regenerator section, with hot and cold products flowing on either side is shown schematically in Figure 2a. The flow is a combination of counter and cross flows. Since the length of the plate is large compared to width (44 in to 13 in), a counter flow situation can be approximated. This assumption seems acceptable for a major portion of the heat exchange surface except near the ports.

With the above assumption the situation simplifies to the one shown (side view) in Figure 2b.

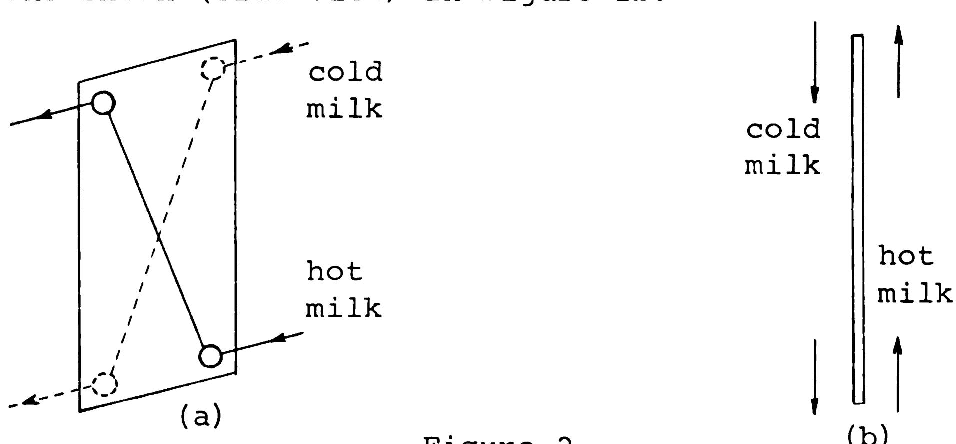

Figure 2

Let

t_{h_1} = entering temperature of hot milk ^{o}F

t_{h_2} = exit temperature of hot milk ^{o}F

t_{c_1} = inlet temperature of cold milk ^{o}F

t_{c_2} = outlet temperature of cold milk ^{o}F

X = thickness of the plate in ft (0.00365 ft as measured)

k = thermal conductivity of the plate material (18-8 stainless steel 302 series) 9.4 BTU/hr ft ^{o}F

Δt = temperature difference across the thickness of plate. ^{o}F

A_p = area of heat exchange perpendicular to the direction of flow (projected area) ft^2

U_p = overall coefficient of heat transfer based on projected area. BTU/hr ft^2 ^{o}F.

A_d = developed area of heat exchange. ft^2

ℓ = any location on the plate.

At any location on the plate, heat transfer Q_ℓ equals:

$$Q_\ell = k_\ell A_{p_\ell} \frac{\Delta t_\ell}{X_\ell} \qquad (1)$$

Since the plate has the same thickness throughout and is made of uniform material,

$k = k_{\ell_1} = k_{\ell_2} = \_\_\_\_$

$X = X_{\ell_1} = X_{\ell_2} = \_\_\_\_$ and

$$A_p = A_{p_{\ell 1}} = t_{p_{\ell 2}} = \_\_\_\_ \quad \text{therefore}$$

$$Q_{\ell} = k\ A_p \frac{\Delta t_{\ell}}{X} \quad \_\_\_\_ \tag{2}$$

For a situation where overall heat transfer coefficient U is constant, specific heats of the fluids are constant and the flow rate is constant, the heat transfer is given by

$$Q = U\ A_p\ \Delta t_m \tag{3}$$

Where Δt_m is the mean temperature difference or the difference of the bulk temperatures of the fluids flowing on the two sides. Since U value is not constant in our case, the above equation was modified for local conditions.

$$Q_{\ell} = U_{\ell}\ A_{p_{\ell}} (\Delta t_m)_{\ell} \tag{4}$$

$(\Delta t_m)_{\ell}$ being the difference of stream temperatures of the hot and cold products at location ℓ.

Equations (2) and (4) give $U_{\ell}\ A_{p_{\ell}} (\Delta t_m)_{\ell} = kA_p \frac{\Delta t_{\ell}}{X}$.

<u>U_{ℓ} is based on the projected area.</u>

The above equation can be rearranged to the form

$$U_{\ell} = \frac{k}{X} \frac{\Delta t_{\ell}}{(\Delta t_m)_{\ell}} = \frac{k}{X} \left(\frac{\Delta t}{\Delta t_m}\right)_{\ell}$$

Substituting proper values of k and X

$$U = 2575\left(\frac{\Delta t}{\Delta t_m}\right)_\ell$$

Values of Δt_ℓ were taken from the pairs of thermocouples connected across the plate thickness. $(\Delta t_m)_\ell$ were obtained from the local stream temperatures, calculated from the inlet and outlet temperatures of the flowing liquids.

Inlet and outlet temperatures of hot and cold milk flowing over the plate were measured by placing thermocouples in the center of the ports, Figure 5. At the hot milk outlet port, milk coming from the adjacent plate mixed with the local stream. The temperature of the mixture recorded by the thermocouple was higher than actual, since the mixing stream was at a higher temperature. This necessitated correction of t_{h_2}.

The corrugations on the two plates forming a flow channel are reversed in the assembly, thereby exposing the flowing stream to continually varying widths of the channel. This causes turbulence and mixing of the fluid at every location on the plate. The temperature of the fluid stream, therefore, remains approximately constant along the width of the plate and changes along the length only. At the same time, the temperature difference between the fluid and the wall surface is nearly constant at a cross-section along the width.

Temperature profiles, Figures 13 and 14 made from actual tests for the hot and cold sides of the plate showed that plate temperatures were nearly constant along the width of the plate but changed in a straight line along the length. The hottest and coldest plate temperatures were found to occur near the inlet and outlet ports, respectively. Since the plate temperature changes in a straight line along the length, the fluid temperature also changes in the same pattern but at a constant amount higher than the former. Therefore the outlet temperature of the hot milk, t_{h_2}, can be assumed to be the same amount higher than the coldest temperature of the plate wall as the inlet temperature is from the hottest temperature of the plate.

Values of t_{h_2}, corrected on this basis, are shown in Table 1 (appendix). To counter check this, an additional thermocouple was placed in the hot milk outlet port (Figure 6). It was placed at 1/8" from the port periphery. Tests were performed under similar conditions. For an average hot milk inlet temperature, t_{h_1}, of 117.791°F, the outlet temperature measured by the second thermocouple was 79.000 $^{\circ}$F. From Table 1 corrected values of t_{h_2} for t_{h_1} in the range 117°F - 118°F are shown.

Test no.	Hot milk inlet temperature t_{h_1}	Corrected hot milk outlet temperature t_{h_2}
18	117.530°F	79.541°F
6	117.875°F	80.301°F
15	118.022°F	79.917°F

The corrected values of t_{h_2} compare reasonably with the measured one.

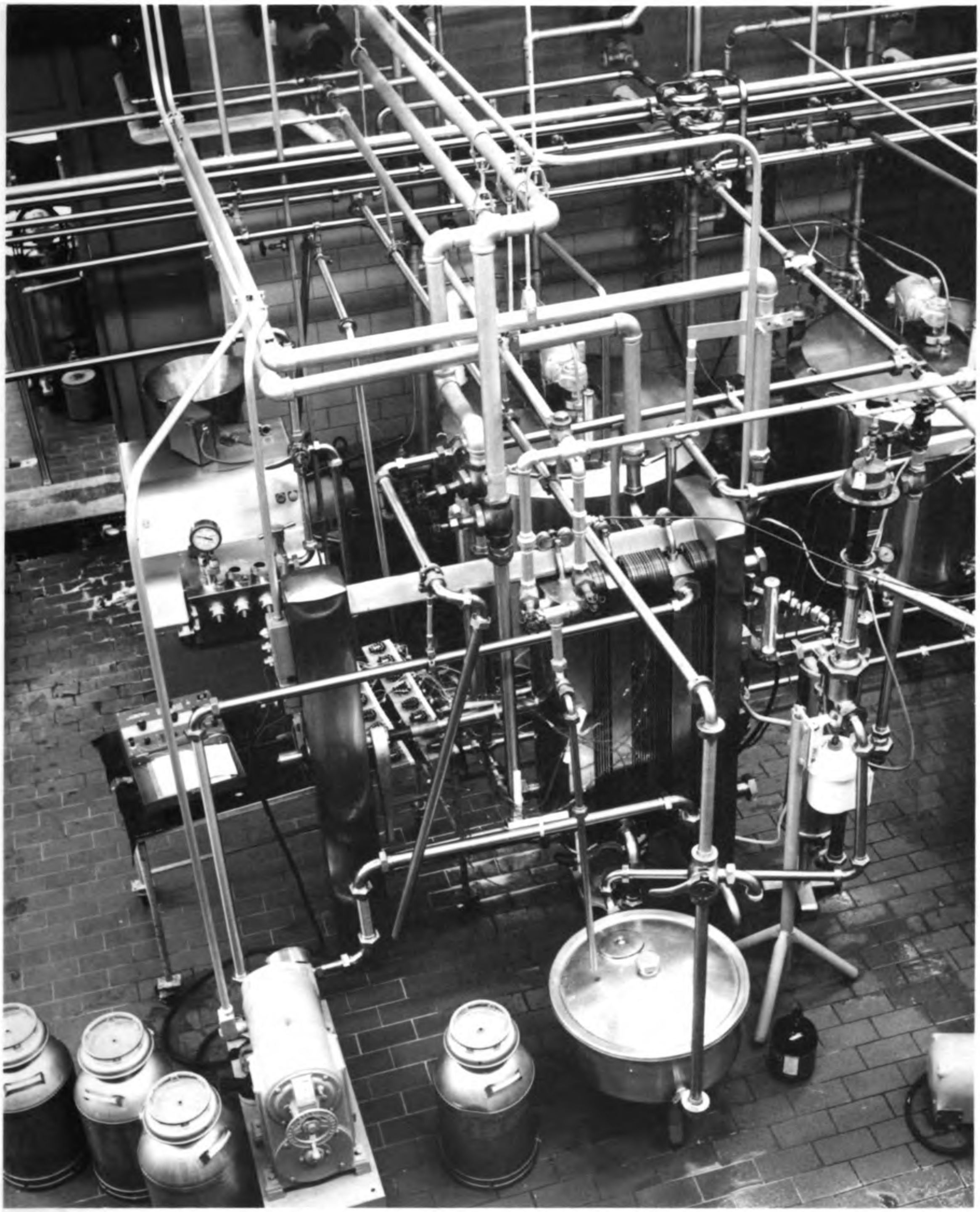

Figure 3. General view of the pasteurizing equipment

EQUIPMENT AND INSTRUMENTATION

Facilities of the Michigan State University Dairy Plant were used in carrying out this study. Tests were made with the high-temperature short-time (HT-ST) milk pasteurizing unit during actual operation.

The HT-ST pasteurizer is a plate type heat exchanger having three heat-exchange units, namely, heater, regenerator and cooler.

Raw milk is pumped by a positive displacement pump to regenerator section where it is heated by hot pasteurized milk. The regenerator section contains 29 plates, the heater 13 plates and the cooler 13 plates. Flow diagram for the regenerator section is shown in Figure 4.

The heated raw milk from regenerator flows into the "heater" unit. Here it is further heated by hot water, to pasteurizing temperature. This is then routed through a vacuum chamber and homogenizer before it enters regenerator section. Regeneration efficiency* of 80-82% is common under normal working conditions.

Cold water is used to cool the pasteurized milk in the "cooler" section.

*Defined on p. 35.

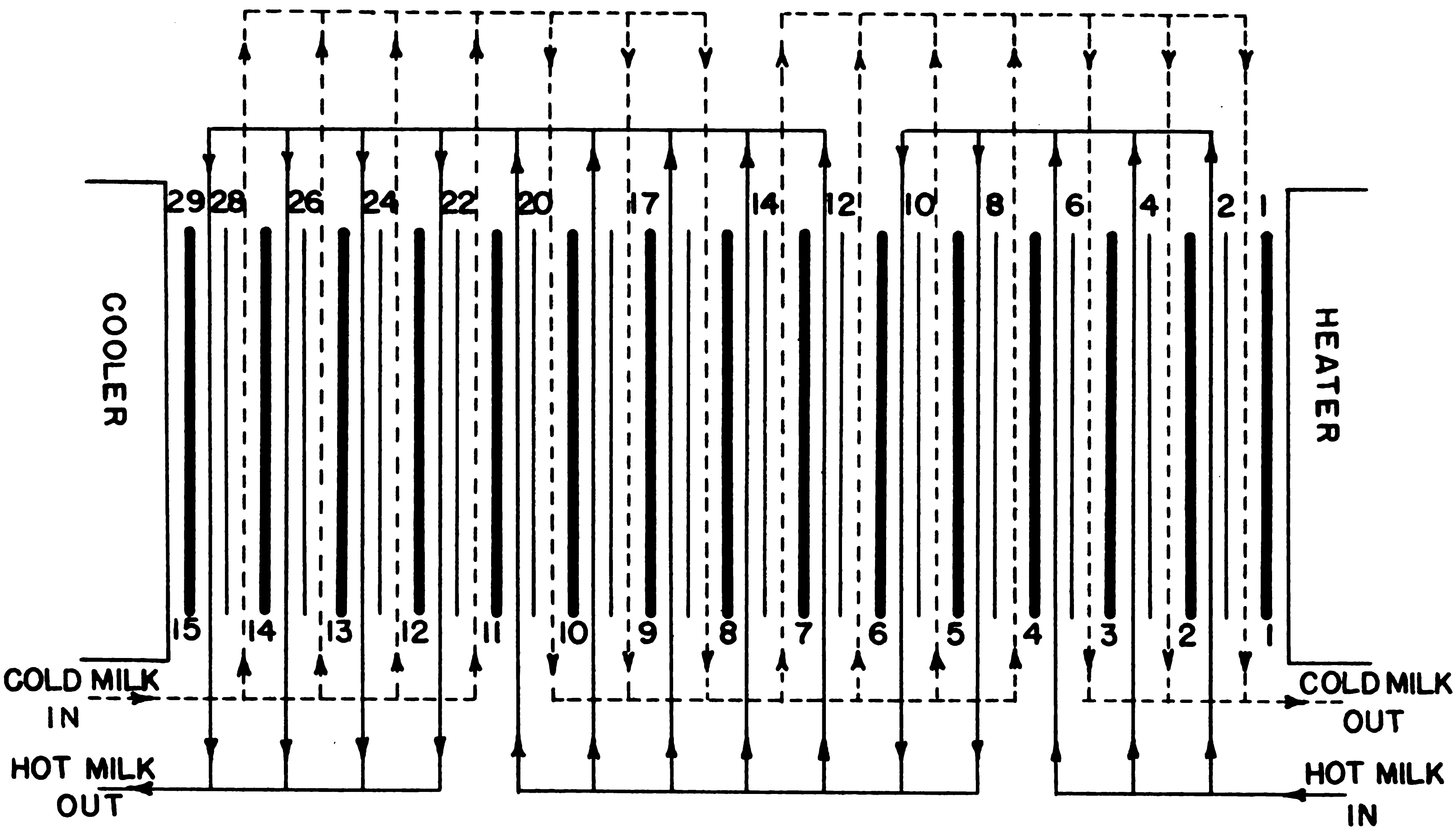

FIGURE 4. FLOW DIAGRAM OF THE REGENERATOR UNIT

This investigation was made with the test plate held at position "17" in regenerator section. At this position hot pasteurized milk flowed from bottom to top on one side and cold raw milk from top to bottom on the other side of the test plate.

Corrugations or the turbulence promotors on the plate were segmental in shape and curved upward when assembled in the unit. The plates immediately on either side of the test plate had corrugations curving downward for greater amounts of turbulence.

Instrumentation

A difficult part of instrumentation was placing thermocouple junctions on the test plate. Various bonding materials were tried for this purpose. Duco cement was tried first, as it had the advantage of quick setting. A 2-1/4" x 3-1/2" size sample piece was used for these trials. Material of this piece was the same as the test plate, i.e. 18-8 stainless steel of 302 series having no. 4 finish. The cement bond came off when heated to about 120° F in a water bath.

Epoxy glue was also used with type "E" hardener. The manufacturer's recommended procedure for initial setting

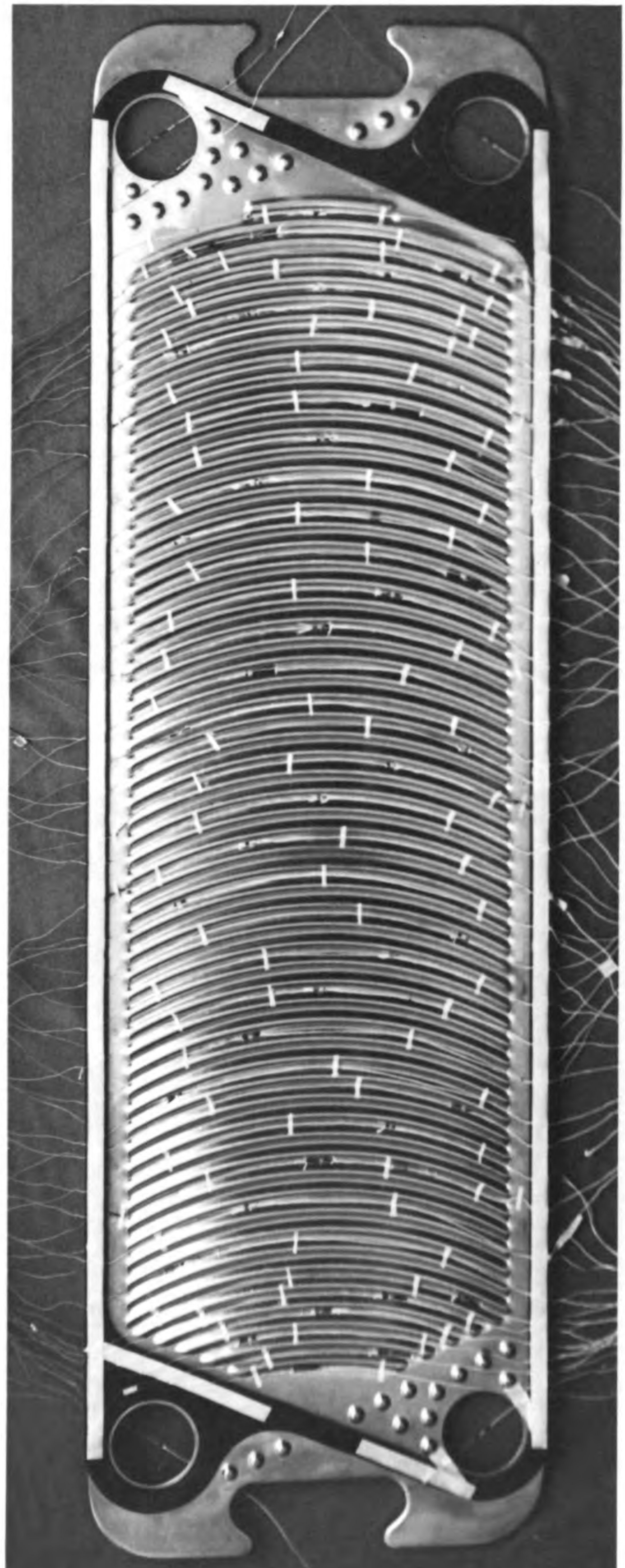

Figure 5. Test plate with thermocouples

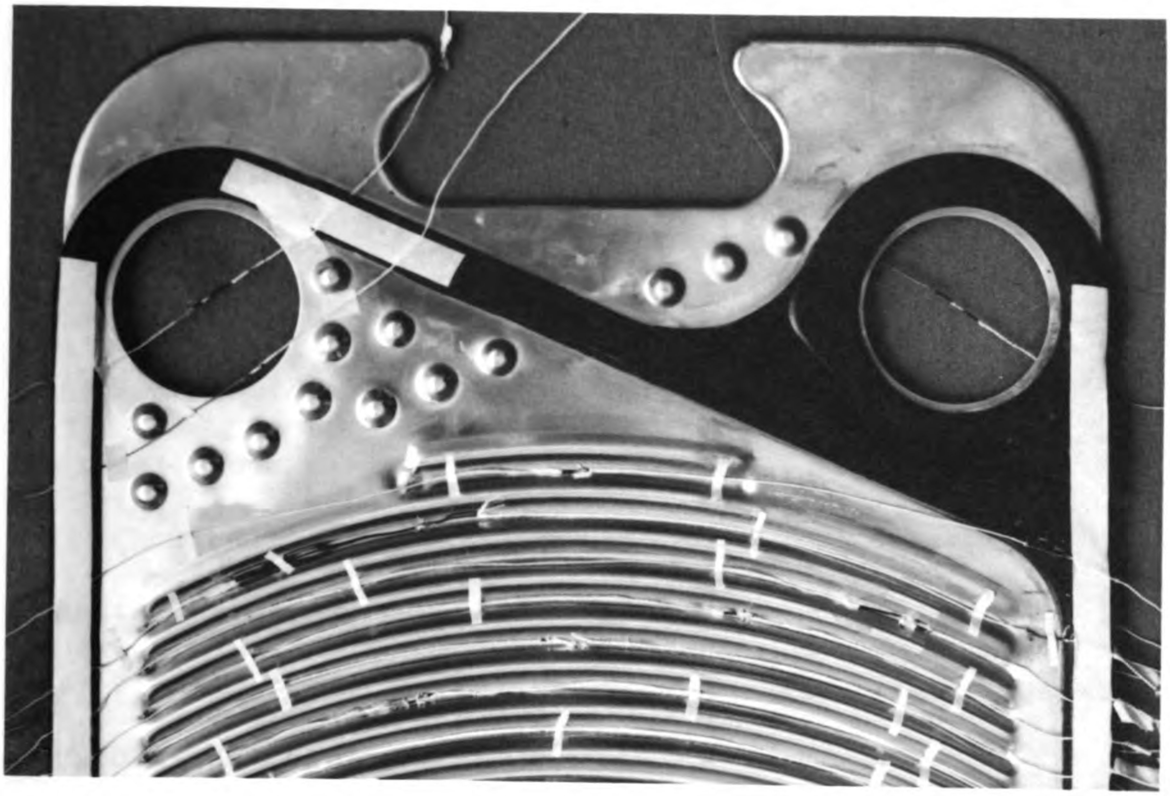

Figure 6. Detail of the test plate at hot milk outlet port

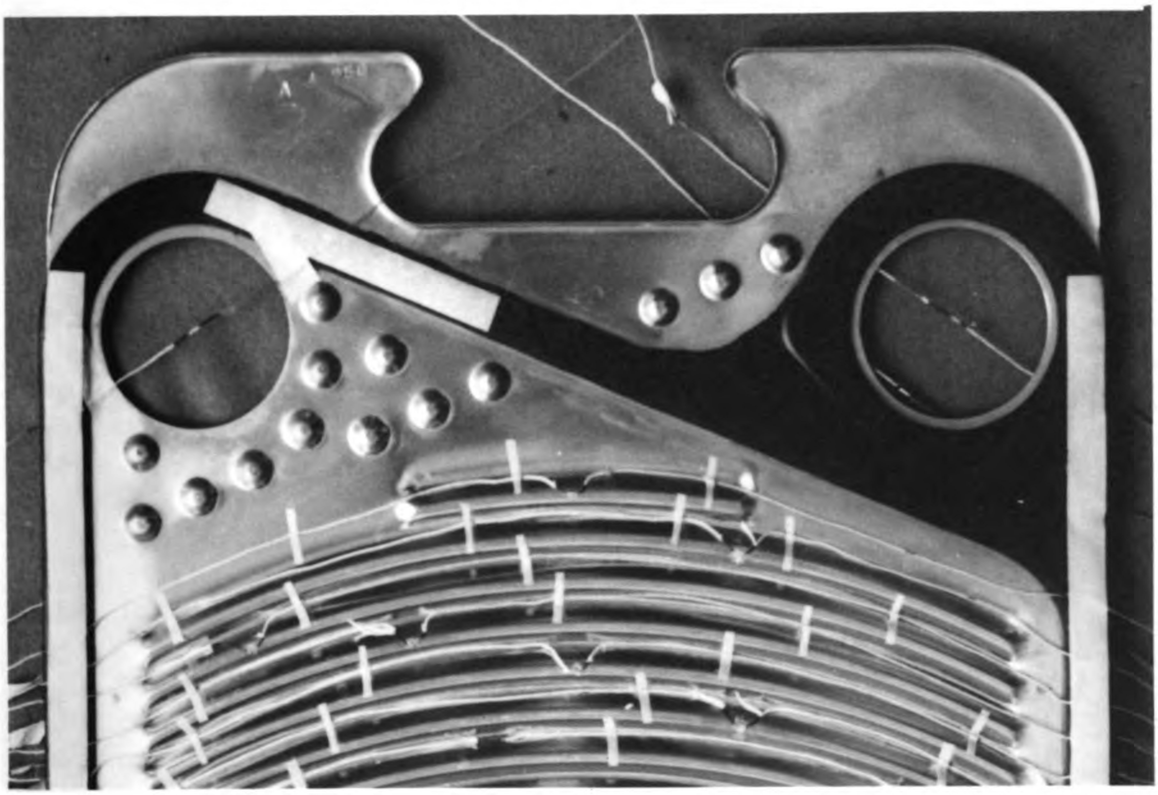

Figure 7. Detail of the test plate at the cold milk inlet port

was followed. The bond seemed to be very strong at room temperature. The test sample was heated to 165°F, the temperature the bond could be exposed to in the pasteurizing unit, and maintained at that temperature. In about fifteen minutes the glue started peeling off and therefore was rejected.

Chemical adhesives were not considered because of sanitary reasons.

Finally stainless steel solder with a proper flux was tried. The results were satisfactory.

Cu-constantan (1938 calibration; 24 B & S gage) thermocouples were used for measuring the temperatures. Thermocouple junctions were made by soldering the copper and constantan wires on the surface of the plate. The thinness of the plate prevented boring or indenting the plate at the junctions. The fact that two junctions had to be placed opposite to each other, across the thickness of the plate made boring or indentation even more difficult.

The soldering was done with extreme care so as to keep the junction thickness to a minimum.

Seventy-six thermocouples were soldered to the plate forming 38 pairs. The positions of these sets were arranged so as to get adequate information for plotting U-values.

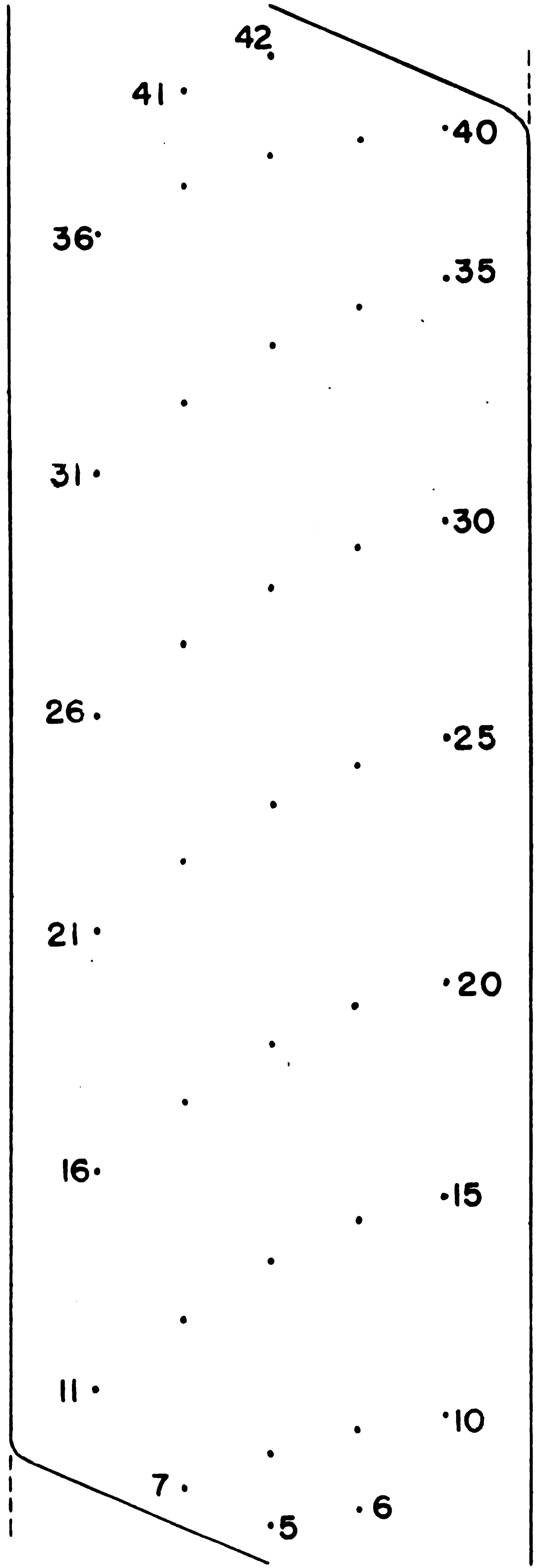

FIGURE 8. THERMOCOUPLE LOCATIONS ON TEST PLATE

The width of the plate was divided into six equal parts with five dividing lines. Thermocouple junctions were placed on these lines at the alternate ridges along the length of the plate. Near the entry and exit ports, however, junctions were placed on each ridge so as to better investigate these areas of diverging and converging flows. In addition, thermocouples 15-16 and 25-26 were also placed one ridge apart to allow the desired arrangement of junctions at the ports (see Figure 5).

Lead wires from a junction were run along the segmental shape of the corrugation that the junction was in, and taken out of the plate at the rubber gasket, nearest to the edge of the corrugation. Masking tape was used to keep them in place. All Cu leads were taken out at one end of the plate while all constantan leads were removed at the other end. This arrangement proved helpful in connecting leads to the outside switches. One hundred sixty leads thus obtained (seventy-six junctions on plate surface and four at inlet and outlet ports of hot and cold milk) were connected to adaptor switches, each taking eleven leads. The adaptor switches were mounted on wooden platforms to avoid contact with water.

Stream temperatures of incoming and outgoing milk both on hot and cold sides were measured by placing copper constantan thermocouples in the flow passage. The arrangement used is shown in Figure 6.

A reflecting galvanometer type Leeds and Northrup precision potentiometer (no. 8686) was used for measuring thermocouple outputs (Figure 9). A common ice bath reference junction was used with the potentiometer. As such, the manufacturer's indicated accuracy for the instrument was $\pm$ .05% of reading + 3 mv. The potentiometer could be read up to one thousandth of a millivolt or approximately 0.05°F.

The thermocouples were read manually. It took about 25 to 30 minutes to complete one cycle of readings.

The flow rate of the product through the heat exchanger was measured by timing flow of a measured quantity through the unit. This was repeated several times to check and confirm the rates, which were kept constant.

PROCEDURE

Test plate with the thermocouples was placed in the pasteurizing unit an evening before the working day. This was done twice a week, on Tuesdays and Thursdays, to conform with the production schedule of the Dairy Plant.

In the heat exchanger alternate plates were gasketed with rubber overlapping both sides of the edges. One of these gasketed plates was used as the test plate because of the ease in taking out thermocouple leads.

The position of test plate was seventeenth in regenerator section counted from the heater end (see Figure 4), or the ninth gasketed plate. This position was selected because of counter flow of milk and moderate temperatures.

General arrangement of the unit and the test equipment is shown in Figure 3.

Before starting the test the temperatures were stabilized at a constant value. This was done by measuring the temperature of the incoming hot milk and outgoing cold milk until two consecutive readings were coincident.

The potentiometer was standardized before, during, and at the end of each test.

Figure 9. Leeds & Northrup precision potentiometer

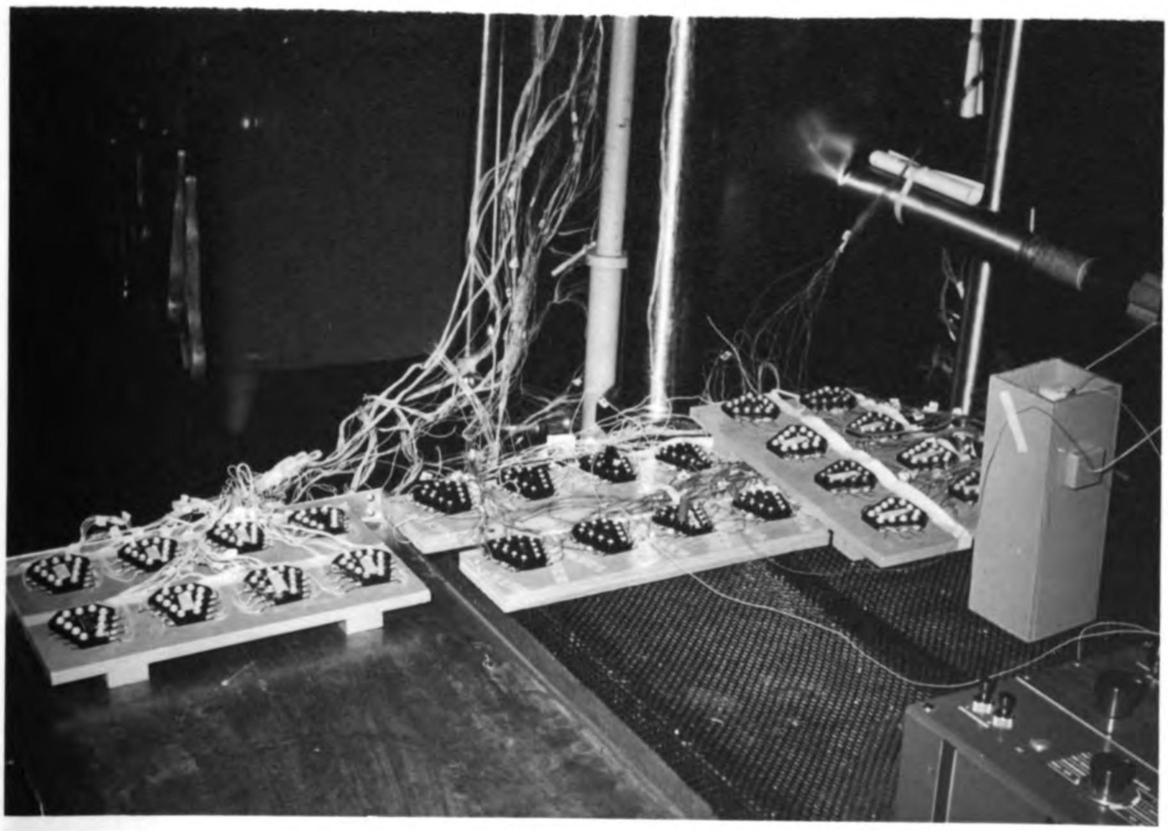

Figure 10. Arrangement of thermocouple leads and the switches

At steady-state the test was started by measuring the incoming and outgoing temperatures of hot and cold milk. Next the plate thermocouples were read. This was done in such a way as to read consecutively the thermocouples soldered opposite to each other on the plate. This insured better accuracy in measurement of the temperature difference because the time elapsed was at a minimum. Plate thermocouples were numbered from 5 to 42 on the hot side and from 43 to 80 on the cold side. Thus, the order in which thermocouples were read was 5, 43, 44, 6, 7, 45, 46, 8 . . . .

The potentiometer circuit is shown in Figure 11. The copper lead from ice bath was connected to the negative terminal of the potentiometer, while the constantan lead was soldered to an adapter plug which fits in the adapter switch connected to thermocouple leads. Positive terminal of the potentiometer was connected to another plug by a copper wire.

The two plugs thus obtained were moved simultaneously and plugged in similar numbered switches. The thermocouple output was measured at the potentiometer and values in millivolts were noted. At the same time the plugs were moved to the next position. It took two persons to do the job, one reading the potentiometer and the other changing

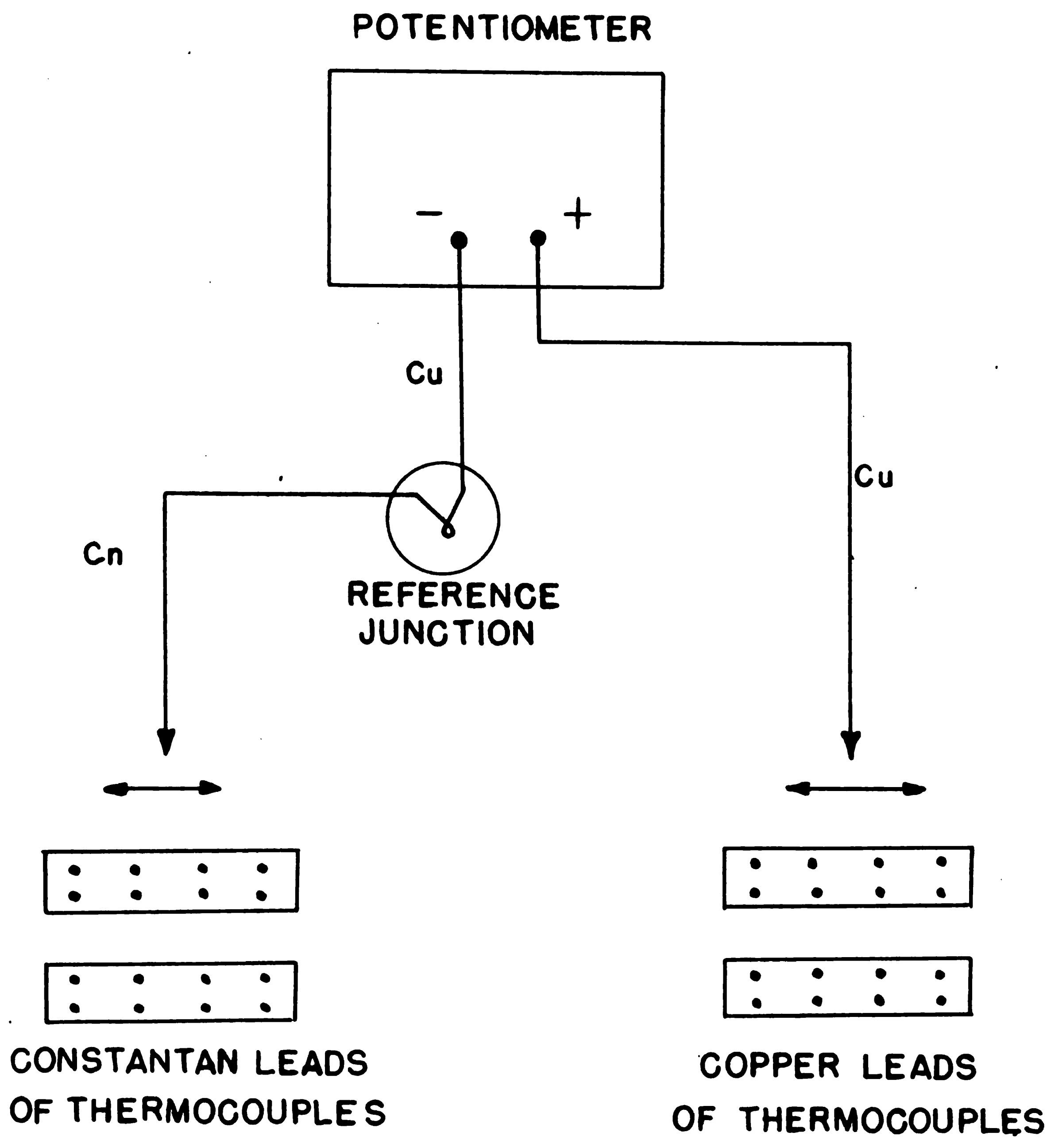

FIGURE II. POTENTIOMETER CIRCUIT

the plugs and noting the e.m.f. values. On the average, one cycle of readings was completed in about twenty-five minutes. During this time port temperatures were recorded five times so as to give a more realistic average.

The flow rate of the product was determined by timing the flow of a measured quantity through the heat exchanger. Product was pumped through the unit by a positive displacement pump running at a constant setting of speed. Flow rate checked at six occasions during the study showed a maximum variation of 1.5% from the average value of 6920 lbs/hr of the whole milk.

At the conclusion of each test the test plate was removed from the press before the unit was cleaned by CIP cleaning system. This was done to avoid damage to the thermocouple junctions and leads, since temperatures and velocities of cleaning solution were much higher than those of the product. Soon after the plate was taken out of the exchanger it was cleaned with alkali-base cleaning powder with light brushing and finally washed thoroughly with warm water. Before the plate was put back into the heat exchanger, the thermocouples were checked for any damage to the junction or insulation coating on the wires. The wires were repaired or replaced, as needed.

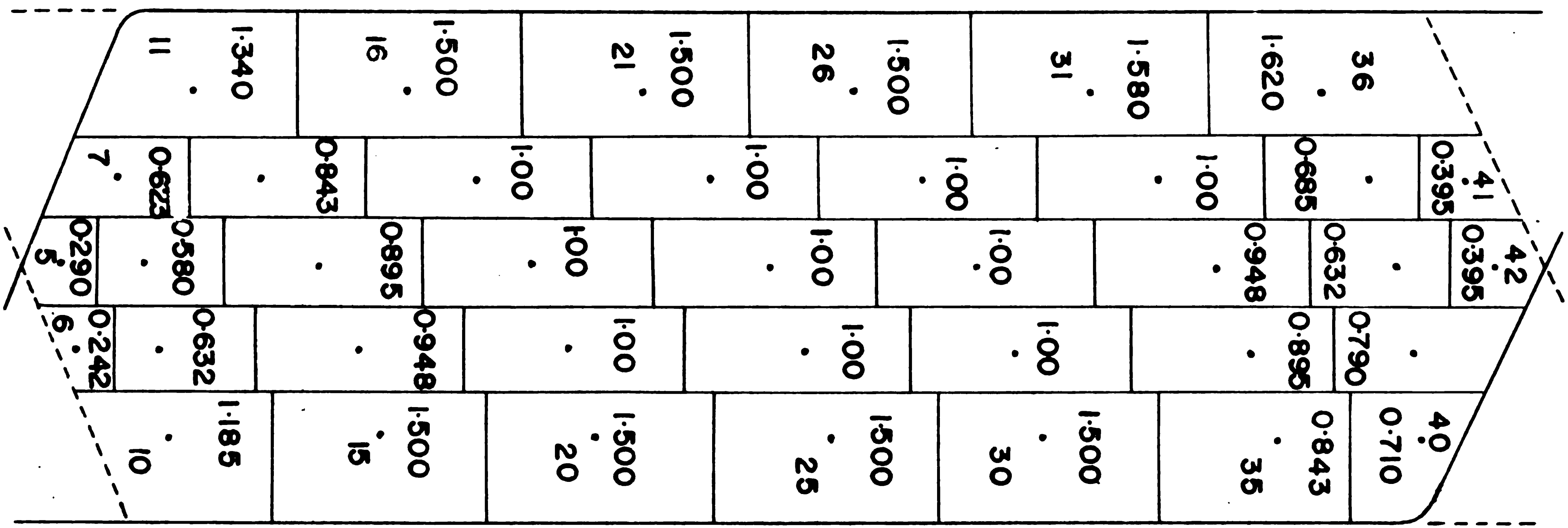

FIGURE 12. REPRESENTATIVE HEAT TRANSFER AREAS FOR THERMOCOUPLES

RESULTS AND DISCUSSION

Overall Effectiveness of the Regenerator Unit

The inlet and outlet temperatures of hot pasteurized milk and cold raw milk, measured for the entire regenerator unit, yielded an overall effectiveness of 79%. Since flow rate was equal for both hot and cold streams, effectiveness, E, of the regenerator was calculated using the formula

$$E = \frac{\Delta t'_c}{t'_{h_1} - t'_{c_1}}$$

where

t'_{h_1} = temperature of hot milk inlet, $^{\circ}F$

t'_{c_1} = temperature of cold milk inlet, $^{\circ}F$

and $\Delta t'_c$ = difference of inlet and outlet temperatures of cold milk stream, $^{\circ}F$.

This is lower than the average value of 82% commonly reported. This was attributed to the fact that the gasketed plate no. 14 (Figure 4) was assembled incorrectly by the operators, and allowed no flow on the two sides. This resulted in a lower effectiveness of that particular pass as well as a lower effectiveness for the entire unit. This was corrected after the tests were concluded.

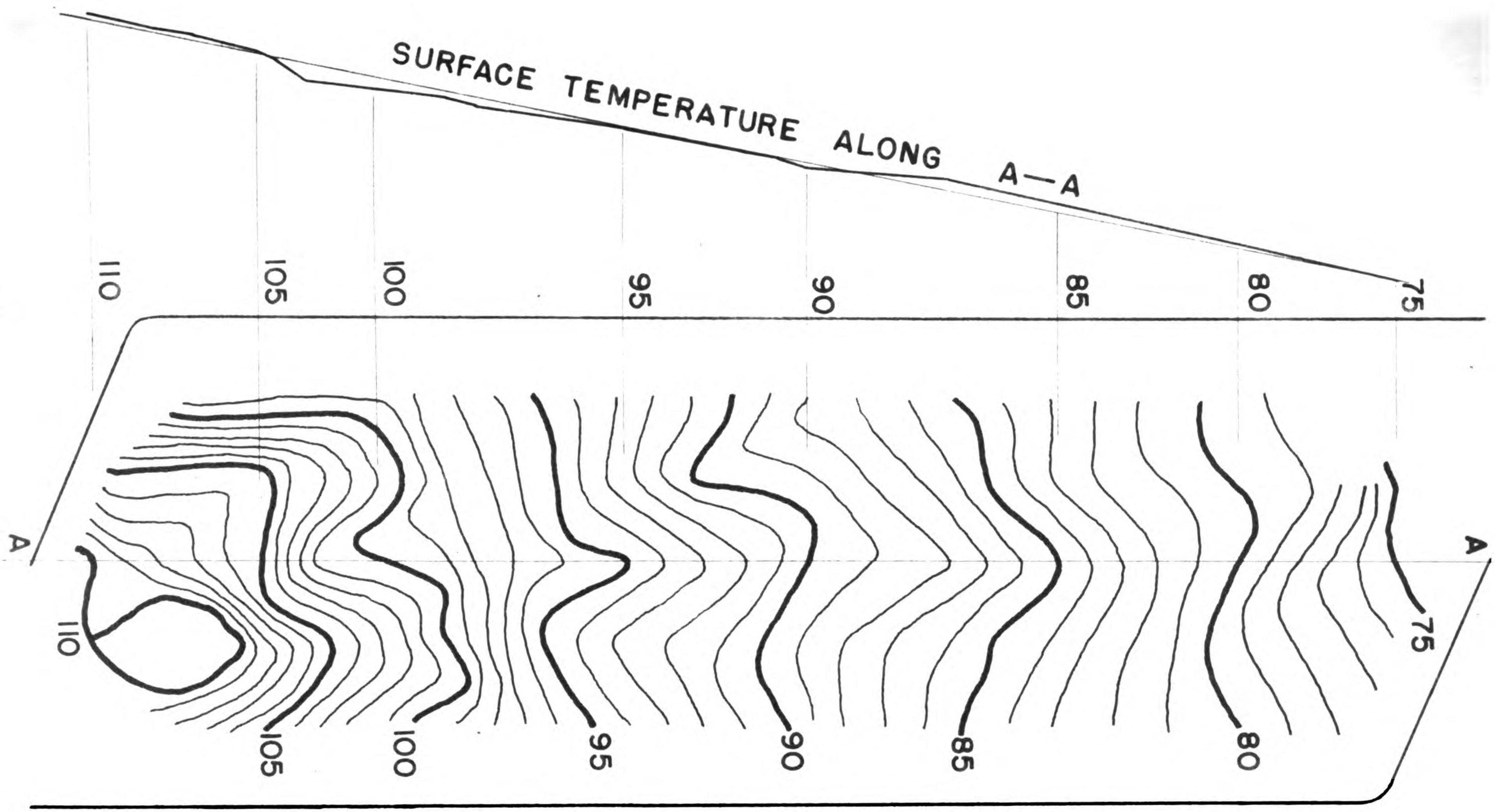

FIGURE 13. TEMPERATURE DISTRIBUTION ON PLATE SURFACE (HOT SIDE)

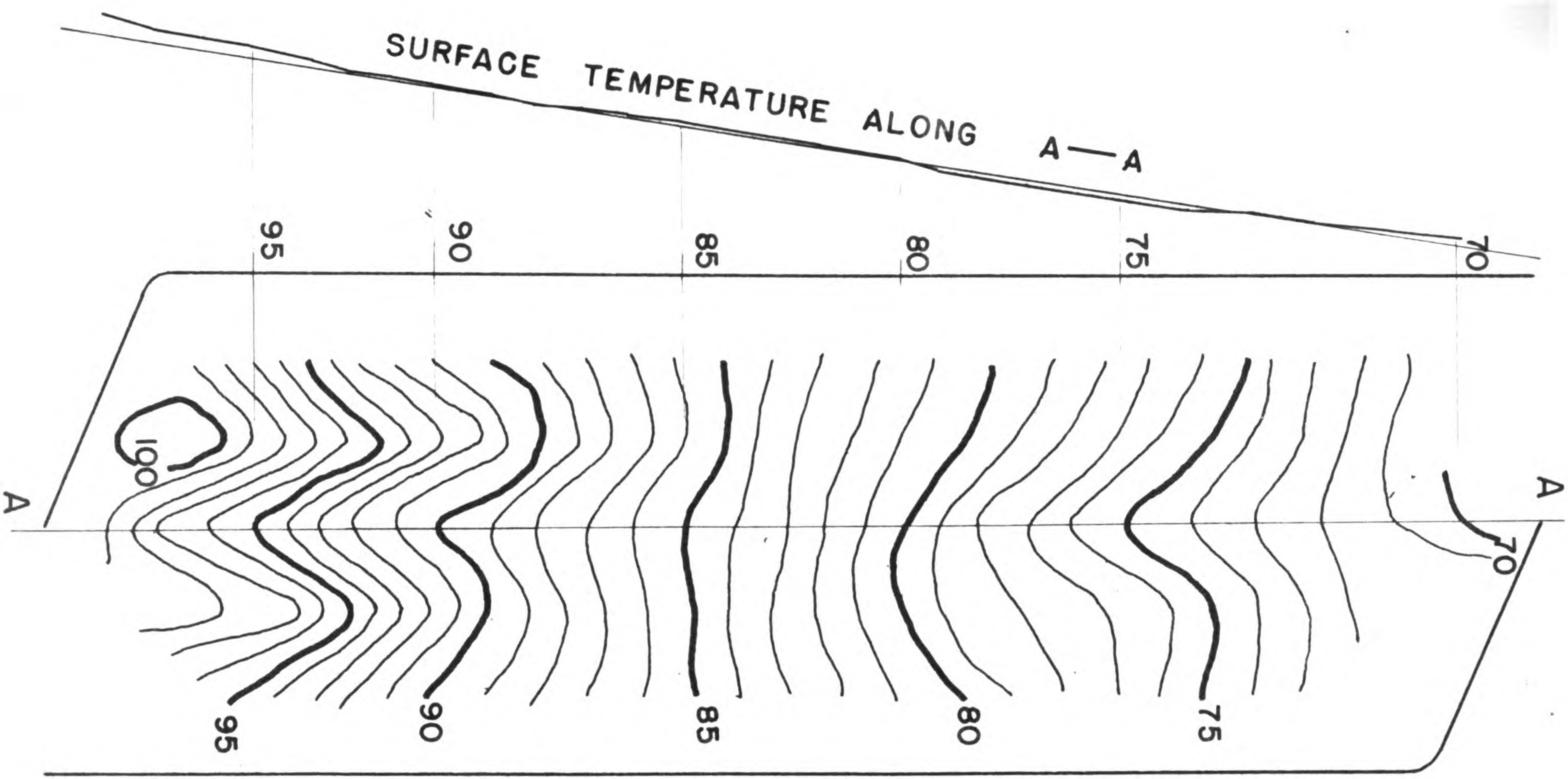

FIGURE 14. TEMPERATURE DISTRIBUTION ON PLATE SURFACE (COLD SIDE)

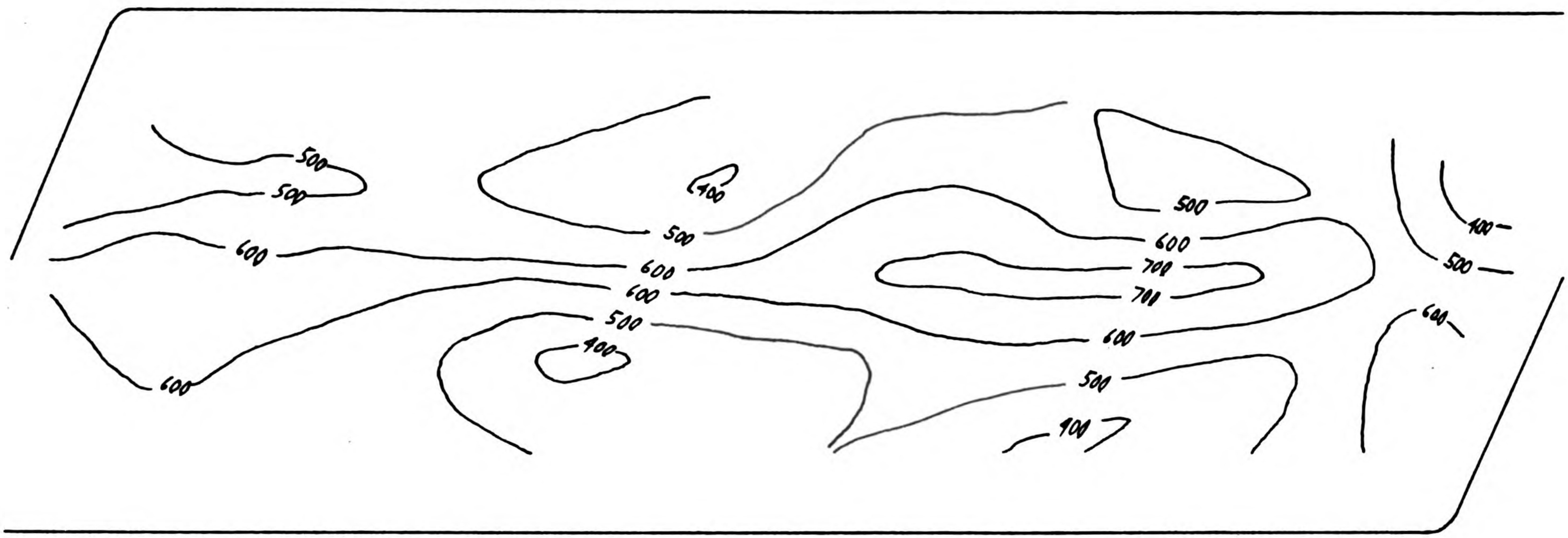

FIGURE 15. U-PLOT ON TEST PLATE FOR GROUP I.

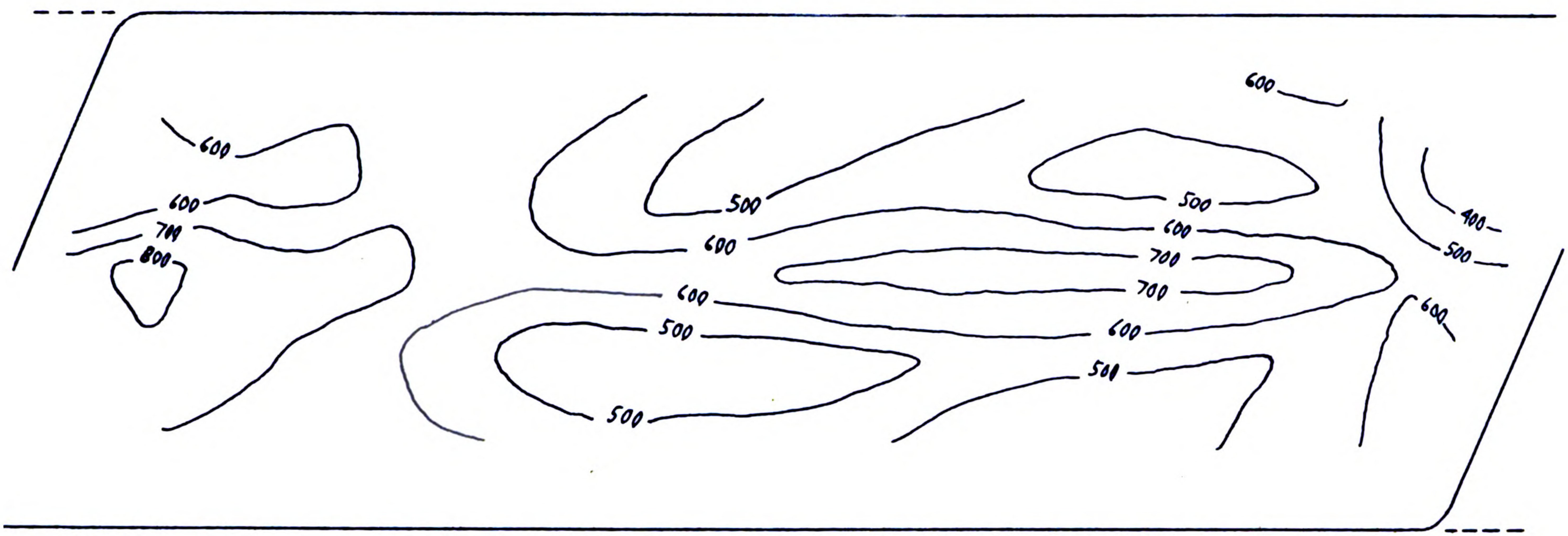

FIGURE 16. U-PLOT ON TEST PLATE FOR GROUP II

Temperature Measurements on the Plate

Plate temperatures were measured on both hot and cold sides as a part of the information required for calculating local overall heat transfer coefficients, U_ℓ.

Constant-temperature curves over the plate, i.e. isotherms, were plotted. It was observed that:

1. for main body of the heat exchange area, the temperatures, in general, were approximately constant along the width of the plate. Near the ports, however, the temperatures varied because of converging and diverging flows.
2. along the length of the plate the temperatures varied in a straight line.

Figures 13 and 14 show typical isotherm plots for hot and cold sides, based on data from test 3.

Overall Heat Transfer Coefficients

Local overall heat transfer coefficients were calculated from equation

$$U_\ell = 2575\left(\frac{\Delta t}{\Delta t_m}\right)_\ell$$

using notations given on page 14.

Values of $(\Delta t_m)_\ell$ for different locations were calculated from measured values of t_{h_1}, t_{c_1}, t_{c_2}; and corrected values of t_{h_2} as discussed earlier.

A total of twenty tests were performed under similar operating conditions, keeping the test plate at position "17" from heater end, in the regenerator section.

A close examination of the data showed that for all the tests the temperature differential $(t_{h_1} - t_{c_1})$ remained nearly constant (about 57°F). However, the data came into two clearly defined groups based on the temperature increase of the cold stream.

In group I average temperature increase of cold milk was observed to be about 19°F while it was 27°F for group II. This variation, though not clearly understood, could be attributed to one or a combination of the following:

1. Change in flow ratio: as is evident from Figure 4, at the test plate the theoretical flow ratio of hot and cold milk should be 3:5. Deviation from this value would be expected:

    a) if air was entrapped in one or both of the streams. More air accumulation has been reported by E. L. Watson (16), in a divided pass. Same reference observed that air pockets were formed on the down-flow side of the plate, in inverse ratio to the flow rate. The entrapped air and the air pockets would tend to affect the mass flow rate and the local velocities, thereby effecting the temperature rise of the flowing fluid.

b) if milkstone was formed on the test plate or the adjacent plates contained in that divided pass. This would block the flow channels and cause varying velocities and pressure differentials.

c) if the unit was assembled with different pressures, causing increase or decrease of the width of the flow channels.

2. Shift in heat balance at the plate.

It was assumed that the hot stream at the test plate lost the same amount of heat to the adjacent cold stream on right as the cold stream gained from the hot one on its left. This was approximated, on the assumption that the adjacent streams had identical temperatures. A deviation from this would mean a shift in heat balance resulting in more or less amount of heat gained by cold milk stream than lost by the hot stream at the test plate.

3. Change in degree of turbulence.

Varying degrees of turbulence caused by variations of milk deposits, air pockets, entrapped air, pressure differential, etc. would affect the amount of heat transferred and hence the fluid temperatures.

The situation is very complex with no direct and definite answer. Since no solid ground was available to discard either set of the data, both were treated equally.

Using values of local Δt_m (mean difference of local bulk temperatures), the local overall heat transfer coefficients (U_ℓ) were calculated. These are shown in Table 2 (Appendix).

Representative heat-exchange areas were allocated to each thermocouple junction, Figure 12. A weighted average overall heat transfer coefficient, U_{Av}, was calculated for each test. The mean and the deviations of U_{Av} were calculated separately for groups I and II, as well as for all the twenty tests. Results are summarized in Table 4 (Appendix). Means of U_{Av} for group I and group II were within 7% of each other.

These values were found to be within 15% of McKillop's results (11). However, significant difference was observed when compared to values predicted by McAdams' (10) or Peeples' (13) empirical correlations for clear channel flows.

The local overall coefficients were also averaged for each of groups I and II. Values are tabulated in Table 3 (Appendix). Comparison between the values for group I and II showed that these were within 10% of each other for an area extending from thermocouple 17 to the hot milk exit port. This was better than two-thirds of the entire plate area. Near the hot milk inlet port, however, variations were high. This is explained by the fact that local conduction of heat from main stream of hot milk to the plate is significant. This effect is clearly obvious from Figure 13, which shows a rather abrupt rise in plate temperature near the hot milk inlet port.

Figures 15 and 16 show contours of the overall heat transfer coefficients for groups I and II, based on the average local values.

A close study of these contours revealed several facts.

1. The average overall heat transfer coefficient varied from 300-800 BTU/hr ft^2 ^{o}F over the plate.
2. U-values were higher in the middle, at any cross-section along the width, and decreased towards the plate edge.
3. U-values were more uniform in the vicinity of the hot milk inlet port compared to any other location. This may be due to superimposed effect of the local conduction.
4. U-value variations were very small at cross-sections along the length compared to cross-sections along the width. This may be due to the fact that the fluid velocities at any cross-section along the length remain

nearly constant but vary, somewhat, along the width of the flow channel.

5. U-value variation is not large for the major part of the heat exchange area, beyond the port vicinities. For processes encountering very small temperature rise of the fluids, like in heater section of a milk pasteurizer, assumption of constant U for the plate is reasonably accurate.

The contours of temperature and overall heat transfer coefficients suggested that the flow velocity was uniform in the middle of the plate and was somewhat higher compared to the sides. Also it showed that the velocity at any cross-section along the length was constant. This meant that the velocity profile was similar all along the length of the plate, except near the ports. This observation agreed with the motion picture analysis of E. L. Watson *et al*. (16).

Sample Calculations

a) Overall effectiveness of the regenerator unit.

Since an equal amount of milk was flowing in hot and cold streams, the effectiveness is given by:

$$E = \frac{\Delta t'_c}{t'_{h_1} - t'_{c_1}}$$

Using the data from test 21,

t'_{h_1} = temperature of hot milk entering the regenerator = $170.880^{\circ}F$.

t'_{h_2} = temperature of hot milk leaving the regenerator = $64.318^{\circ}F$.

t'_{c_1} = temperature of cold milk entering the regenerator = $49.818^{\circ}F$.

t'_{c_2} = temperature of cold milk leaving the regenerator = $145.360^{\circ}F$.

$\therefore \quad \Delta t'_c = t'_{c_2} - t'_{c_1} = 145.360 - 49.818 = 95.542^{\circ}F.$

and $t'_{h_1} - t'_{c_1} = 170.880 - 49.818 = 121.062^{\circ}F.$

$$\therefore \qquad E = \frac{95.542}{121.062} = 79\%$$

b) The local overall heat transfer coefficients.

Using data obtained in test 3,

t_{h_1} = hot milk inlet temperature = $120.500^{\circ}F$

t_{c_1} = cold milk inlet temperature = $64.273^{\circ}F$

t_{c_2} = cold milk outlet temperature = $82.522^{\circ}F$

the hot milk outlet temperature t_{h_2} as measured in the test did not give proper values because of mixing of the adjacent hot milk streams. It was corrected as given below:

t_{w_h} = hottest temperature of test plate near the hot milk inlet port = $112.217^{\circ}F$

then

$$t_{h_1} - t_{w_h} = 120.500 - 112.217 = 8.283^{\circ}F$$

t_{w_c} = coldest temperature of the test plate near the hot milk outlet = $74.045^{\circ}F$.

According to the assumption that the bulk temperature of the hot stream at inlet and outlet ports is higher than the plate temperature, by the same magnitude, corrected value of t_{h_2} will be $t_{h_{2\ corrected}} = 74.045 + 8.283 = 82.328^{\circ}F$, knowing

$$t_{h_2},\ t_{h_1},\ t_{c_1},\ \text{and}\ t_{c_2}$$

$$\Delta t_1 = 120.500 - 82.522 = 37.978^{\circ}F$$

$$\Delta t_2 = 82.328 - 64.273 = 18.055^{\circ}F$$

$$\Delta t_1 - \Delta t_2 = 37.978 - 18.055 = 19.923^{\circ}F$$

Using the assumption that bulk temperatures of hot and cold milk change in a straight line, the plate length is divided into thirty-six equal parts. Each region represents an inverval of $0.560^{\circ}F$ which is a $\pm$ 1% change of average bulk temperature difference.

$$\frac{\Delta t_1 + \Delta t_2}{2} = \frac{37.978 + 18.055}{2} = 28.016^{\circ}F$$

Thus the value of $(\Delta t_m)_\ell$ in the first interval will be $18.055 + \frac{0.560}{2}$ or $18.335^{\circ}F$, for second interval it will be

18.335 + 0.560 or 18.895°F; for third interval 18.335 + 2 (0.560) or 19.455°F and so on.

The local overall heat transfer coefficients are calculated from equation

$$U_\ell = 2575\left(\frac{\Delta t}{\Delta t_m}\right)_\ell$$

Using proper values of $(\Delta t)_\ell$ and $(\Delta t_m)_\ell$, we find U_{18} (overall heat transfer coefficient for location containing thermocouple junction 18).

ℓ = 18 comes under interval 24 counted from the hot milk outlet end, $(\Delta t_m)_{18}$ will therefore be

$$(\Delta t_m)_{18} = 18.335 + 23\ (0.560) = 31.215^{\circ}F$$

Δt as measured by the pair of thermocouples soldered on either sides of the plate at 18 is 9.008°F.

$$\therefore \quad U_{18} = 2575\ \frac{(9.008)}{(31.215)} = 743 \text{ BTU/hr ft}^2\ {}^{\circ}F$$

This value has to be corrected for extension of thermocouple junctions into the flow.

Heat lost by milk on hot side equals

$$Q = W_h\ C_p\ \Delta t_h = \frac{6825}{5}\ (0.92)\,(38.172) = 48{,}000 \text{ BTU/hr}$$

also

$$Q = U_{Av}\ A_{eff}\ \Delta t_m$$

where

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\ln \frac{\Delta t_1}{\Delta t_2}} = \frac{37.978 - 18.055}{\ln \frac{37.978}{18.055}} = 26.74^{\circ}F$$

U_{Av} = average overall heat transfer coefficient for test 3 = 696 BTU/hr ft^2 $^{\circ}F$

and

A_{eff} = effective heat exchange area of plate = 3.273 ft^2

$\therefore$ Q = 696 (3.273)(26.74) = 60,900 BTU/hr

Thus heat transfer calculated from surface thermocouples is higher than actual.

Correction factor = $\frac{48,000}{60,900}$ = 0.79.

This factor will be constant for each test.

$\therefore$ Corrected U_{18} = 0.79 (743) = 587 BTU/hr ft^2 $^{\circ}F$.

CONCLUSIONS

1. The overall effectiveness of the regenerator unit was found to be 79%.
2. The plate-surface temperatures were seen to be fairly constant along the plate width, except near the ports.
3. The local conduction of heat from main stream of hot milk to the plate near the hot milk inlet port was observed.
4. The wall temperature changed linearly along the length of the plate.
5. The local overall heat transfer coefficients were found to be between 300-800 BTU/hr ft^2 ^oF. The average U-value of plate was about 550 BTU/hr ft^2 ^oF.
6. The contours of the overall heat transfer coefficient showed the values to be higher in the middle of the plate, decreasing gradually towards the plate edge.
7. The overall heat transfer coefficients were observed to be reasonably constant along the length of the plate.
8. The overall heat transfer coefficient for the plate can be assumed constant for processes encountering very little temperature change of the flowing fluids between inlet and outlet ports. This would apply to the heating section of a plate heat exchanger.

RECOMMENDATIONS FOR FUTURE STUDY

1. Determine the effect of different flow rates on the distribution of the overall heat transfer coefficients over the plate.

2. Investigate the effect of port design, i.e. size and center to center distance of the two ports on the same end of plates, on the heat transfer characteristics.

3. Study the effect of plate dimensions on the heat transfer.

REFERENCES

1. Anonymous
1955. Heat exchange conditions in plate type pasteurizer. Dairy Engineering 72 (11): 35.

2. Anonymous
1958. Improvements in and relating to plate type heat exchanger. Dairy Engineering 75 (2): 62.

3. Anonymous
1958. Constructional details of a plate heat exchanger. Dairy Engineering 75 (5): 137-139.

4. Anonymous
1954. Polarizing heat-exchanger gaskets. Dairy Man 71 (10): 312.

5. Coulson, J. M. and Richardson, J. F.
1956. Chemical Engineering. Vol. I., 384 pp. New York: McGraw-Hill Book Co.

6. Giese, H. and Bond, T. E.
1952. Plate heat exchanger design and construction. Agricultural Engineering 33 (10): 617-622.

7. Goodman, H. F.
1953. Some aspects of heat transfer in the plate heat exchanger. Proceedings International Dairy Congress 13 (3): 772-774. June.

8. Hansen, S. A., Wood, F. W. and Thornton, H. R.
1953. Some mechanical features of a plate-type HT-ST pasteurizer. Canadian Journal of Technology 31: 231-239.

9. Lawry, J. F.
1959. Plate type heat exchangers. Chemical Engineering 66 (13): 89-94, June.

10. McAdams, W. H.
1954. Heat Transmission. 3rd ed. New York: McGraw-Hill Book Co.

11. McKillop, A. A. and Dunkley, W. L.
1960. Plate heat exchangers: heat transfer. Industrial and Engineering Chemistry 52 (9): 740-744.

12. Mennicke, V. V.
1955. Wärmetechnische voraussetzungen und forderungen für den betricb von plattenerhitzern für milch. Fette Seifen Anstrichmitted. 57 (4): 256-262.

13. Peeples, M. L.
1960. Forced convection heat transfer characteristics of fluid milk products. Unpublished Ph.D. thesis, Ohio State University, January.

14. Troupe, R. A., Morgan, J. C. and Prifti, J.
1960. Plate heaters. Chemical Engineering Progress 56 (1): 124-126.

15. Trout, G. M. and Hall, C. W.
1962. Pasteurization of Milk (in process).

16. Watson, E. L., McKillop, A. A., Dunkley, W. L. and Perry, R. L.
1960. Plate heat exchangers: flow characteristics. Industrial and Engineering Chemistry. 52 (9): 733-740.

17. Whitaker, J. R.
1947. Plate heat exchangers and their applications. Australian Journal of Dairy Technology. 2 (3): 110-118, July-September.

APPENDIX

REYNOLDS' NUMBER CALCULATIONS

The corrugation design and the assembly of the plate unit was such that it was not possible to measure accurately the width of the flow channel. The difficulty was overcome by filling, with water, the flow passage of chilled water in the cooler unit. The cooler unit had plates exactly identical to the ones in the regenerator section. The water was drained and measured. Volume of this water equalled that of the six flow channels and the approximately 2-3/4" diameter x 9-7/8" long fluid passage. Since dimensions of plate area were known, the average width of the flow channel was calculated.

Volume of water collected	= 8,475 cc
	= 517.5 cubic in
Volume of 2-3/4" dia. x 9-7/8" long flow passage	$= \pi/4(2.75)^2(9.875) = 58.7$ cu in
Volume of six flow channels	= 517.5 - 58.7 = 458.8 cu in
volume/flow channel	$= \frac{458.8}{6} = 76.5$ cu in
Average heat exchange length of plate	= 38.5 in (measured)
∴ Average area of cross-section of the flow channel perpendicular to the direction of flow	$= \frac{76.5}{38.5} = 1.987$ in^2

Average width of plate = 13.0 in (measured)

Average width of the flow channel $= \frac{1.987}{13.0} = 0.153$ in

Average flow rate (measured) = 6,920 lbs/hr of whole milk

$$= \frac{6,920}{(8.6)(7.48)} = 107.5 \text{ cu ft/hr}$$

The total flow of 107.5 cu ft/hr flows into five channels on the hot side in the divided pass which contains position "17" where the test-plate was inserted. While on the cold side it flowed into three channels. Obviously Reynolds' number will be smaller for the hot side compared to the cold, since more quantity of the product is flowing through the latter. Considering the hot side only,

Flow rate/channel $= \frac{107.5}{5} = 21.5$ cu ft/hr

Velocity through one channel $= \frac{21.5}{1.987/144} = 1,560$ ft/hr

$= 0.433$ ft/sec

Reynolds' number (R_e) is given by: $R_e = \frac{D_e \rho V}{\mu}$

where

D_e = hydraulic diameter $= \frac{4(\text{area of X-section})}{\text{wetted parameter}} = 2.52 \times 10^{-2}$ ft

ρ = specific weight of whole milk = 64.3 lbs/cu ft

μ = viscosity of whole milk at 100^{o}F = 1.423 centipoise
$= 9.57 \times 10^{-4}$ lbs/ft sec

$$\text{Reynolds' number} = \frac{(2.52 \times 10^{-2})(64.3)(0.433)}{9.57 \times 10^{-4}} = 732$$

In plate heat exchangers turbulence is reported to occur at 180-200 (7, 9, 14). Therefore the flow was turbulent on both sides of the test plate for this investigation.

CALCULATION OF U FROM EMPIRICAL CORRELATION

Investigations of heat transfer in turbulent flow have resulted in many semi-empirical correlations based on the experimental results. Most of these are of the type

$$Nu = C(R_e)^m(Pr)^n \tag{1}$$

where

$Nu = \frac{hD}{k}$ = Nusselt number, dimensionless

$R_e = \frac{D\rho v}{\mu}$ = Reynolds' number, dimensionless

$Pr = \frac{\mu C_p}{k}$ = Prandtl number, dimensionless

C = constant

and m, n = exponents

Values of C, m and n vary with each investigator.

Peeples, M. L. (13) studied the flow of whole milk through a tubular heater and proposed the following equation

$$Nu = \frac{hD}{k} = 0.27\ R_e^{0.577}\ Pr^{0.4} \tag{2}$$

Modifying this for rectangular channel ($D=D_e$), film coefficients h_h and h_c (on hot and cold sides) can be obtained. Then overall heat transfer coefficients is calculated from

$$\frac{1}{U} = \frac{1}{h_h} + \frac{X}{k} + \frac{1}{h_c} \tag{3}$$

a) Hot side of the plate:

Average bulk temperature of hot milk $= \frac{119+81}{2} = 100.0^{\circ}F$

Average temperature of plate surface $= \frac{110+74}{2} = 92.0^{\circ}F$

$\therefore$ Average film temperature $= \frac{100+92}{2} = 96.0^{\circ}F$

ρ = specific weight of whole milk at $100^{\circ}F$ = 64.3 lbs/cu ft

C_p = heat capacity of whole milk at $100^{\circ}F$ = 0.92 BTU/lb

k = thermal conductivity of whole milk at $100^{\circ}F$ = 0.358 BTU/hr ft $^{\circ}F$

μ = viscosity of whole milk at $96^{\circ}F$ = 9.57×10^{-4} lbs/ft sec

V = velocity of milk through the channel = 0.433 ft/sec (calculated)

D_e = hydraulic diameter = 2.52×10^{-2} ft (calculated)

Substituting these values in equation (2)

$$h_h\left(\frac{2.52\text{x}10^{-2}}{0.358}\right) = 0.27\left(\frac{2.52\text{x}10^{-2}\text{x}64.3\text{x}0.433}{9.57\text{x}10^{-4}}\right)^{0.577}\left(\frac{9.57\text{x}10^{-4}\text{x}3600\text{x}0.92}{0.358}\right)^{0.4}$$

or

$$h_h = \frac{(0.27)(45)(2.39)}{7.04 \times 10^{-2}} = 413 \text{ BTU/hr ft}^2\ {}^{\circ}F$$

b) Cold side of the plate:

Average bulk temperature of cold milk $= \frac{61 + 83}{2} = 77.0^{o}F$

Average temperature of plate surface $= \frac{100 + 70}{2} = 85.0^{o}F$

$\therefore$ Average film temperature $= \frac{77 + 85}{2} = 81.0^{o}F$

ρ = at $77^{o}F$ = 64.3 lbs/cu ft

C_p = at $77^{o}F$ = 0.93 BTU/lb

k = at $77^{o}F$ = 0.347 BTU/hr ft ^{o}F

μ = at $81^{o}F$ = 11.4×10^{-4} lbs/ft-sec

V = 5/3(0.433) = 0.722 ft/sec

assuming flow ratio (W_h:W_c) of 3:5 at the test plate

D_e = same as on hot side = 2.52×10^{-2} ft

$$h_c\left(\frac{2.52x10^{-2}}{0.347}\right) = 0.27\left(\frac{2.52x10^{-2}x0.722x64.3}{11.7 \times 10^{-4}}\right)^{0.577}\left(\frac{11.7x10^{-4}x3600x0.93}{0.347}\right)^{0.4}$$

or

$$h_c = \frac{0.27(53.8)(2.64)}{7.26 \times 10^{-2}} = 528 \text{ BTU/hr ft}^2 \ {}^{o}F$$

Therefore

$$\frac{1}{U} = \frac{1}{413} + \frac{0.00365}{9.4} + \frac{1}{528}$$

or

$$U = 215 \text{ BTU/hr ft}^2 \ {}^{o}F$$

This is significantly smaller than the mean U_{Av} (550 BTU/hr ft^2 ^{o}F) obtained in this study. This can be attributed to the fact that the Peeples equation predicts U values for clear channel flows only. Turbulence promoters provided on the test plate create turbulence at relatively small Reynolds' number (200 compared to 2100 for clear flow). Therefore higher heat transfer rates are obtained at comparatively smaller flow rates.

Table 1

Values of Hot Milk Outlet Temperature °F

Test no.	Inlet temp: of hot milk °F t_{h_1} (A)	Hottest plate temp: °F (B)	Coldest plate temp: °F (C)	Corrected hot milk outlet temp: °F t_{h_2} = C + (A-B)
8	115.875	107.708	68.909	77.076
7	115.875	104.957	69.261	80.179
9	116.304	106.130	68.409	78.583
11	116.565	105.833	66.227	76.959
12	117.083	105.000	66.136	78.219
10	117.125	106.043	66.955	78.037
4	117.167	109.458	72.818	80.527
18	117.530	109.250	71.261	79.541
6	117.875	109.791	72.217	80.301
15	118.022	107.583	69.478	79.917
14	118.168	108.522	71.826	81.472
13	118.581	109.208	71.261	80.634
17	118.751	108.957	70.773	80.570
5	118.870	109.167	73.304	83.007
16	119.562	110.435	72.136	81.263
1	119.750	110.609	73.739	82.880
3	120.500	112.217	74.045	82.328
2	121.333	114.333	77.818	84.818
20	123.610	114.625	73.779	82.764
19	123.625	114.792	73.739	82.572

TABLE 2

VALUES OF LOCAL OVERALL HEAT TRANSFER COEFFICIENTS

in BTU/hr ft^2 °F.

Thermo-couple number	Group I: Test Numbers												Group II: Test Numbers							
	1	2	3	4	5	6	13	14	17	18	19	20	7	8	9	10	11	12	15	16
5	657	608	615	564	550	579	635	614	658	668	643	596	667	688	690	790	821	798	743	707
6	622	602	572	452	451	482	678	603	665	629	529	526	751	746	750	781	684	677	670	656
7	388	381	361	300	291	293	462	399	381	362	361	349	358	357	356	428	403	397	447	426
8	686	684	645	642	640	661	720	704	733	717	664	672	740	810	821	898	889	814	841	804
9	676	672	622	605	604	628	632	592	634	585	626	641	799	882	876	797	832	749	713	617
10	573	577	524	585	557	566	587	564	494	553	434	521	618	674	630	660	653	615	817	786
11				585	592	617	555	512	592	572	605	621	622	675	655	784	760	720	656	615
12	435	470	476	477	470	493	508	477	477	526	546	463	485	519	536	617	573	556	645	537
13	575	452	527	619	656	658	719	681	653	661	708	696	718	780	760	798	781	734	692	639
14	576	554	554	523	524	508	611	572	585	592	600	644	619	639	645	682	685	631	603	683
15	632	619	629	568	514	574	668	646	617	592	600	552	612	640	602	709	686	673	656	725
16	537	562	520	538	476	501	623	557	524	531	557	540	579	584	621	690	686	614	602	651
17	498	413	511	472	508	516	562	493	569	527	574	508	657	693	685	757	713	659	539	611
18	602	600	583	592	589	603	617	548	643	677	656	642	609	647	640	707	642	615	650	645
19	415	392	370	399	332	373	399	353	409	391	367	310	388	395	376	444	439	398	419	447
20	485	453	538	545	515	568	463	380	440	440	369	288	503	537	525	527	562	495	537	536
21	528	541	489	512	488	559	544	477	511	571	554	534	507	580	602	608	636	656	594	615
22	387	396	418	396	386	421	369	347	423	467	354	294	387	465	446	407	430	417	410	466
23	620	623	664	634	618	666	696	618	737	706	691	716	597	720	685	724	719	696	742	728
24	543	487	486	442	446	447	497	449	478	448	438	557	391	475	501	447	454	438	426	480
25	518	558	554	470	436	459	514	527	521	469	489	502	543	634	643	522	506	507	487	488
26	460	448	463	392	404	402	522	469	474	440	512	427	377	370	342	512	488	470	439	391
27	667	642	672	601	569	586	594	589	561	510	636	583	512	565	557	601	558	519	516	579
28	815	820	867	717	675	743	731	699	701	729	643	656	722	788	851	804	808	710	700	764
29	561	582	551	556	518	542	596	543	503	609	619	604	537	584	499	578	616	516	491	565
30	376	478	397	272	318	309	371	346	394	382	435	391	412	409	415	340	366	348	338	444
31	467	483	483	457	508	523	479	470	526	567	550	578	493	527	533	551	535	519	538	556
32	338	422	443	430	389	433	407	418	471	462	462	434	376	397	484	427	450	406	433	438
33	671	800	742	684	695	741	792	648	800	731	776	732	708	816	813	766	794	743	695	753
34	441	469	516	453	422	442	472	473	461	420	511	515	452	499	477	524	510	447	504	523
35	472	425	539	529	448	516	572	530	587	606	517	552	507	548	585	532	534	480	587	547
36	620	626	609	641	564	590	655	581	613	638	594	551	581	628	638	646	659	599	512	597
37	656	536	694	556	534	590	497	444	364	435	559	562	406	468	484	459	432	398		
38	502	566	535	505	505	537	563	678	518	648	622	641	536	534	561	632	661	642	519	640
39	637	593	644	649	576	593	749	747	663	792	681	717	516	611	638	654	701	513	700	766
40	601	605	679	663	595	661	739	734	648	694	763	780	616	602	672	666	754	725	634	605
41	273	354	245	268	239	282	351	314	352	358	317	317	207	274	254	287	248	299	277	324
42	414	431	449	547	502	571	527	518	543	568	478	437	412	428	467	511	511	568	615	558

Table 3

Average Local Overall Heat Transfer Coefficients, BTU/hr ft^2 $^\circ$F, for Group I and II

Thermo-couple location	Average U for group I	Average U for group II	Thermo-couple location	Average U for group I	Average U for group II
5	615	738	24	476	451
6	567	714	25	501	541
7	360	396	26	451	423
8	680	827	27	600	550
9	626	785	28	733	768
10	544	681	29	565	548
11	583	685	30	372	384
12	485	558	31	507	531
13	633	737	32	425	426
14	570	648	33	734	761
15	600	662	34	466	492
16	538	628	35	524	547
17	512	664	36	607	607
18	612	644	37	535	529
19	376	413	38	568	590
20	457	527	39	670	637
21	525	600	40	680	659
22	388	428	41	306	271
23	665	701	42	507	509

Table 4

Weighted Average Overall Heat Transfer Coefficient for individual tests. BTU/hr ft^2 ^oF

Group I

Test	1	2	3	4	5	6	13	14	17	18	19	20
U_{Av}	537	538	547	522	501	533	568	527	545	552	549	543

$$\text{Mean} = \frac{6462}{12} = 538 \text{ BTU/hr ft}^2 \ ^oF$$

Deviation from mean U_{Av} of 538 BTU/hr ft^2 ^oF is $\pm$ 6.9%.

Group II

Test no.	7	8	9	10	11	12	15	16
U_{Av}	536	563	582	603	604	566	565	586

$$\text{Mean} = \frac{4605}{8} = 575 \text{ BTU/hr ft}^2 \ ^oF$$

Deviation from mean U_{Av} of 575 BTU/hr ft^2 ^oF is $\pm$ 6.8%

Mean U_{Av} of group II (575) is within 6.9% of mean of group I.