

AUGMENTATION OF CONDENSATION HEAT TRANSFER
WITH IN-LINE STATIC MIXERS

by

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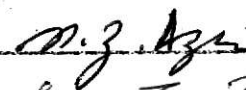
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**THIS BOOK
CONTAINS
NUMEROUS PAGES
WITH MULTIPLE
PENCIL AND/OR
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THROUGHOUT THE
TEXT.**

**THIS IS THE BEST
IMAGE AVAILABLE.**

CHAPTER I

INTRODUCTION

In the design and operation of a heat exchanger, it is often important to keep the ratio of the volume of the exchanger to the rate of heat transfer, as well as the ratio of the pumping power to the rate of heat transfer, as small as possible. The achievement of these two objectives is essential for certain systems, such as space vehicles. Their achievement improves the efficiency and reduces the size of the heat exchanger. It is well known that the ratio of the volume to the rate of heat transfer can be reduced by the use of extended surfaces or by means of displaced promoters in steady or transient operation. However it is generally believed that any of these methods invariably increases the pumping power.

Over the past fifty years, several investigators [1-9] presented analytical and experimental heat transfer and pressure drop results for fluids flowing through pipes with different types of twisted tapes designed to induce swirling in single phase as well as boiling flows. It was observed that the inside heat transfer coefficients in swirling flow could be, under favorable conditions, increased to at least four times the coefficients in purely axial flow for the same mass flow rate. At the same time the pumping power per unit rate of heat transfer could be reduced. The increase in heat transfer was found to depend on the degree of swirl and on the density or temperature gradient. Such observed phenomena were analyzed qualitatively by Frank Kreith and Davie Margolis [2], and it was shown that they were

primarily the result of a centrifugal force which induced a radial inward and outward motion of the fluid.

The recently developed Kenics STATIC MIXER^{*} system is characterized by flow phenomena quite different from those encountered in continuous twisted tape swirl generators. The static mixers combine all the effects of the extended surfaces, displaced promoters, and twisted tapes on the flow. The static mixers have no moving parts, are placed in-line, and give rise to the continuous radial mixing of the moving stream. When the stream is passed through the mixer, two unique mixing actions, namely, flow division and radial mixing, operate simultaneously, resulting in nearly plug flow characteristics. The overall effect is to cause the stream to be continuously and completely inverted radially so that particles entering at the center of the stream are forced to the outer wall and back again to the center on a continuous basis. Because of thorough mixing, the radial profiles of temperature, velocity and composition are minimized. The static mixers can be used in the mixing processing of single-phase flow and two or multi-phase flow to improve the mixing as well as the heat transfer coefficient. It has been claimed that the mixing action is accomplished with a relatively low increase in the pressure drop.

The heat transfer and pressure drop characteristics for single-phase fluid flowing inside tubes with static mixers have been studied theoretically and experimentally by Genetti and Priebe [10]. Their results showed that heat transfer coefficient were considerably enhanced for laminar flow with a static mixer element inserted in a circular tube. The literature survey

^{*}STATIC MIXER is a Registered Trademark of Kenics Corporation.
U.S. Patent No. 3286992.

revealed that no experimental data are available for the effect on heat transfer when static mixers or twisted tapes are used during condensation.

The objective of the present study is to investigate experimentally the effect of the static mixers on the steady state heat transfer and pressure drop characteristics during condensation inside a horizontal condenser.

CHAPTER II

LITERATURE SURVEY

The literature survey is divided into two parts, namely, a survey of literature on techniques to augment the heat transfer, and a survey of literature on condensation and two-phase flow.

Techniques to Augment the Heat Transfer

In recent years, a great deal of research effort has been devoted to developing apparatus and performing experiments to define the conditions under which augmentation techniques will improve heat transfer. The number of such techniques which have been full-scale industrial application is increasing. Some of these techniques, such as surface vibration [11,12], fluid vibration [13], and electrostatic fields [14], require external power and consequently, would reduce the ratio of volume to heat transfer rate at the expense of increasing power consumption. Other augmentation techniques, such as surface roughness [15,16,17] and displaced promoters [18], do not require external power, however, the percentage increase in the heat transfer rate resulting from these techniques is accompanied by any even higher percentage increase in the pressure drop which results in a decrease in the heat transfer rate per unit pumping power.

Other augmentation techniques that lead the way towards achieving the better of both worlds, namely, reducing the equipment size and the pumping power simultaneously, was the use of swirl flow generators. Generation of swirl flow has been accomplished by coiled wires and spiral fins [19],

propellers [20], coiled tubes [21], twisted tapes [1-9], and the newly developed static mixers [10,22]. Emphasis in this review will be placed on the use of twisted tapes and static mixers as swirl generators.

The heat transfer and pressure drop characteristics for single phase fluids flowing inside tubes with tape-generated swirl flow have been studied theoretically and experimentally by numerous investigators. In 1921 Royds [1] measured heat transfer coefficients while cooling high temperature air flow with and without swirl through a tube. He imparted the swirling motion by means of twisted steel strips. His results showed appreciable increase in heat transfer coefficients but this increase was accompanied by a comparable increase in pressure drop. Kreith and Margolis [2] used twisted tapes and twisted wire inserts as vortex promoters and observed a substantial increase in Nusselt number in both cases. They also found that the increase in pressure drop due to tape inserts was smaller than the increase in pressure drop due to wire inserts. Kreith and Margolis [2] emphasized the fact that the amount of power per unit heat transfer rate is an important factor in heat exchanger design. Smithberg and Landis [3] conducted analytical and experimental studies on vortex flow generated by twisted tapes inserted inside horizontal tubes. Their analysis succeeded in predicting the heat transfer coefficient and friction factor within engineering accuracy. Smithberg and Landis [3] recommended the use of the twisted tapes as an inexpensive and efficient heat transfer augmentation technique in tubular heat exchangers, on account of the markedly increase in the Nusselt number during vortex flow.

In an initial study of boiling burnout of water in vortex flow through test sections with inlet swirl generators, Gambill and

Green [4] obtained promising results which encouraged Gambill et al. [5] to conduct an extensive investigation of the effects of twisted-tape swirl generators. In their studies, they performed straight-flow burnout and swirl-flow burnout tests. They observed that the corresponding heat fluxes with swirl-flow were considerably higher than those obtained with straight-flow under similar conditions. They also reported that the ratio of vortex flow to axial flow burnout heat fluxes were approximately two for water flow in identical tubes, at equal pumping powers. An experimental investigation to determine the effects of twisted tapes inserted in tubes on the heat transfer and pressure drop of boiling Freon 11 and water were reported by Blatt et al. [6]. Three tube diameters, four twist-to-diameter ratios, and one pipe length were tested. Their results from Freon 11 experiments definitely established that higher heat fluxes, heat transfer coefficients, and exit qualities could be obtained by swirl flow than by axial flow. However, no apparent effect was observed with water. The pressure drop across the test section increased in all cases owing to the presence of the tape.

Bernstein et al. [7] reported experimental heat transfer and pressure drop data for boiling water while using twisted tapes of both plain and perforated types. Both types were effective in increasing the heat transfer coefficient. However, the pressure drop was very high, especially at small twist-to-diameter ratios. A study of heat transfer and pressure drop in a decaying swirl flow, where a tape insert was used only to impart a centrifugal component to the flow, but did not affect the friction by its presence in the test section, was conducted by Blackwelder and Kreith [8].

In their study heat was transferred to air flowing in a half inch electrically heated tube in which a twisted tape, upstream of the heat transfer test section, imparted a swirl motion to the fluid. It was found that the augmentation of the heat transfer coefficient per unit pumping power in a decaying swirl exceeded considerably that in a continuous swirl flow in the initial 20 to 30 diameters downstream from the end of the swirl inducer. The flow decayed exponentially downstream from the end of the tape, resulting in a continuous decrease in Nusselt number along the flow. Hence, Blackwelder and Kreith [8] suggested that for optimum design of heat exchangers, with tape induced swirl flow, one must consider a combination of continuous and decaying swirl flow. Recently Klepper [9] confirmed the findings of Blackwelder and Kreith [8], when he measured considerably smaller friction factors in the decaying swirl region than in the fully developed swirl flow region. Klepper's [9] results suggested that an empty tube region following a short twisted tape could provide heat transfer enhancement, at a relatively small pressure drop penalty. One drawback of the short twisted tape is the resulting nonuniformity of the heat flux. However, intermittent tapes are particularly useful for cases where non-uniform heat fluxes are involved.

An extensive survey of the methods of augmentation of the heat and mass transfer was presented by Bergles et al. [20,23]. Although there still exists disagreement among investigators regarding the method of correlating the heat transfer and pressure drop data of tape-generated swirl flow inside tubes with and without phase change, there are two main general trends agreed upon. These are:

1. Both the heat transfer and the pressure drop increase in swirl flow.
2. The enhancement in heat transfer and pressure drop increases with the decrease of pitch to diameter ratio of the twisted tape.

According to Bergles et al. [20,23], the information on the effect of twisted tape insertion on the heat transfer and pressure drop characteristics, during condensation of fluids inside tubes, is still lacking. The initiation of such a study is the main objective of the present investigation.

It was mentioned in the previous chapter that a new tape type insertion was developed lately by Kenics Corporation. This new tape, often called the static mixer, has no moving parts, is placed in-line, and results in the continuous mixing of the moving stream. It is constructed of a number of short elements of right and left-hand helices. When the stream is passed through mixers, the stream is divided and radially mixed simultaneously along the tube. Because of thorough mixing, the radial profiles of temperature, velocity, and composition are minimized. Research results on the static mixer's principles of operation, pressure drop, and heat transfer characteristics during single phase flow were presented by Kenics Corporation in a series of eight technical notes, and were summarized by Chen [22]. He observed that the static mixer distinguished itself among tape insertion by its low pressure drop, simplicity, low initial cost, low operation cost, no maintenance cost, no moving parts (no noise), predictability, and reproducibility of the data. A recent investigation was reported by Genetti and Priebe [10], exploring the effect of a static mixer

on the heat transfer characteristics of a double-pipe heat exchanger. Heat transfer coefficients were measured for heat transfer to a motor oil (10w) flowing through a 3/4 inch static mixer. The heat transfer with the static mixer was enhanced significantly over the case without the mixer for laminar flow.

Because of these unique heat transfer and pressure drop characteristics, the static mixers were selected for investigation in this study.

Condensation and Two-Phase Flow

The analysis of condensation heat transfer began with the pioneering work of Nusselt [24] in 1916. Nusselt based his analysis on the assumption that the condensate film flowed down a vertical or an inclined isothermal plate under the action of gravity. He assumed that the condensate film was laminar and that there was a linear temperature gradient from the isothermal plate to the liquid-vapor interface. Using the basic approach set forth by Nusselt, many investigators have refined the analysis. Chato [25] and Chaddock [26] modified the analysis to account for the stratified flow model of laminar condensation in horizontal and inclined tubes. Sparrow and Gregg [27] presented a solution to the boundary-layer type equations developed for the condensate layer. An integral treatment of the two-phase boundary layer in film condensation was presented by Koh [28]. He again used the boundary-layer type equations with a different set of boundary conditions to obtain a more complete solution than that given by the Nusselt analysis. The effect of tube inclination was examined by Hassan and Jakob [29], who found that the heat transfer rate varied as the



one-fourth power of the cosine of the inclination from the horizontal. Tepe and Mueller [30], by condensing benzene and methanol inside an inclined tube, measured coefficients one and half times higher than those predicted by Nusselt analysis.

The laminar character of the condensate film constitutes one of the basic assumptions of the Nusselt theory. Therefore, when the film becomes turbulent, Nusselt's analysis no longer applies. Numerous data and correlations are available in the literature for turbulent condensing flow inside tubes. These correlations use non-dimensional groups. However, most of these empirical correlations have not been tested with enough different types of fluids in order to claim any general applicability. Ackers et al. [31] conducted an analytical and experimental study of the effects of vapor velocity, temperature difference and fluid properties upon the heat transfer coefficient of a condensing vapor inside a horizontal tube. They recommended a correlation for Freon 12 and propane. Their equation is recommended by the ASHRAE HANDBOOK OF FUNDAMENTAL [32] for condenser design calculation. Carpenter and Colburn [33] analyzed the problem of condensation inside tubes by assuming that the major force acting on the condensate film was the vapor shear rather than gravity. They also assumed that the major resistance to heat transfer was offered by the laminar sublayer. The effect of interfacial shear on the rate of condensation was investigated by Rohsenow et al. [34]. Boyko et al. [35] developed a simple heat transfer correlation based on the analogy between heat transfer and hydraulic resistance, for condensing vapors inside horizontal tubes. Soliman et al. [36] reported a study of the interaction between friction, momentum, and gravity as they affect the heat transfer

process during annular condensation flow inside tubes. They formulated a general heat transfer correlation which would predict local heat transfer coefficients and also formulated a general pressure drop equation. The heat transfer coefficient correlation was based on a modification of the shear force in the Carpenter and Colburn [33] correlation.

Azer et al. [37] reported the experimental measurements of the heat transfer coefficients during forced convection condensation of Freon 12 inside horizontal tubes. Azer et al. [38] also investigated analytically the problem of annular flow condensation inside horizontal tubes. The analysis included a modification of Lockhart and Martinelli [39] parameter for predicting two-phase pressure gradient. The modified parameter was based on an experimentally measured pressure drop during annular condensation of Freon 12 inside a horizontal tube. Azer et al. [38] compared the prediction of their analysis with the existing data and found good agreement. Goodykoontz et al. [40,41] measured the local heat transfer coefficients and pressure drop during downward flow condensation of steam and refrigerant - 113 inside a vertical tube. They correlated their experimental results by the use of the Carpenter-Colburn method [33].

Bae et al. [42] reported the results of an analytical study of the problem of annular flow condensation inside tubes using a simple flat plate analysis. No suitable design correlations resulted from their analysis. Recently Vrabie et al. [43] investigated the condensation of Freon 12 vapor in both internally finned and unfinned horizontal tubes. They reported the experimental heat transfer and pressure drop data of both smooth and finned tubes. They also developed an analytical model and compared its prediction with their measurements. The agreement appeared to be reasonable.

Vrable et al. [43] also compared the predicted variation of the local pressure gradients versus quality in figures that included their experimental measurements for three different flow models: annular, dispersed, and slug. The data points fell predominantly near the annular flow model.

The starting point in the condenser design is the identification of the flow pattern during condensation process and the selection of the proper correlation for the prediction of the heat transfer coefficient. A comprehensive analytical and experimental study of flow patterns during condensation inside horizontal tubes was made by Soliman [44,45] over a wide range of flow conditions. Soliman [44,45] identified six major and three transition flow patterns. He also concluded that Baker's map [46] is not adequate for predicting the flow patterns during condensation. Based on his experimental data, a new flow pattern map was developed for condensation using Baker's coordinates [46]. Recently a mathematical model was developed by Soliman and Azer [47] for the annular flow during condensation inside horizontal tubes. The model was based on the idea that a small circumferential motion existed inside the liquid film, in addition to its axial motion. In their analysis a criterion was developed for the prediction of the annular to wavy flow transition. The result showed that the condenser's tube diameter, the condensing fluid flow rate, and the condensation temperature are the important parameters that affect the point of transition from annular to wavy flow.

CHAPTER III

EXPERIMENTAL INVESTIGATION

The objective of this investigation are:

1. Obtain a set of average heat transfer coefficients, and pressure drop data under forced flow conditions during condensation of refrigerant-113 inside horizontal tubes with and without static mixers.
2. Examine effects of the number and distribution of the static mixers inside the condenser tube on heat transfer and pressure drop.

Experimental Test Facility

Figure 1 shows a skematic view of the test facility that was used for this investigation. The same test facility was used by Soliman [45] in visualizing the refrigerant 113 flow patterns inside three horizontal condensers of inside diameters $3/16$, $1/2$, and $5/8$ ins, except for the addition of one test condenser with static mixers and a differential pressure cell for refrigerant pressure drop measurement. Plate I shows a photographic view of the entire experimental test facility. The main components of the system were:

1. Vapor generator.
2. Superheater.
3. Two horizontally mounted in parallel test condensers.
4. Vapor-liquid separator.
5. After condenser.

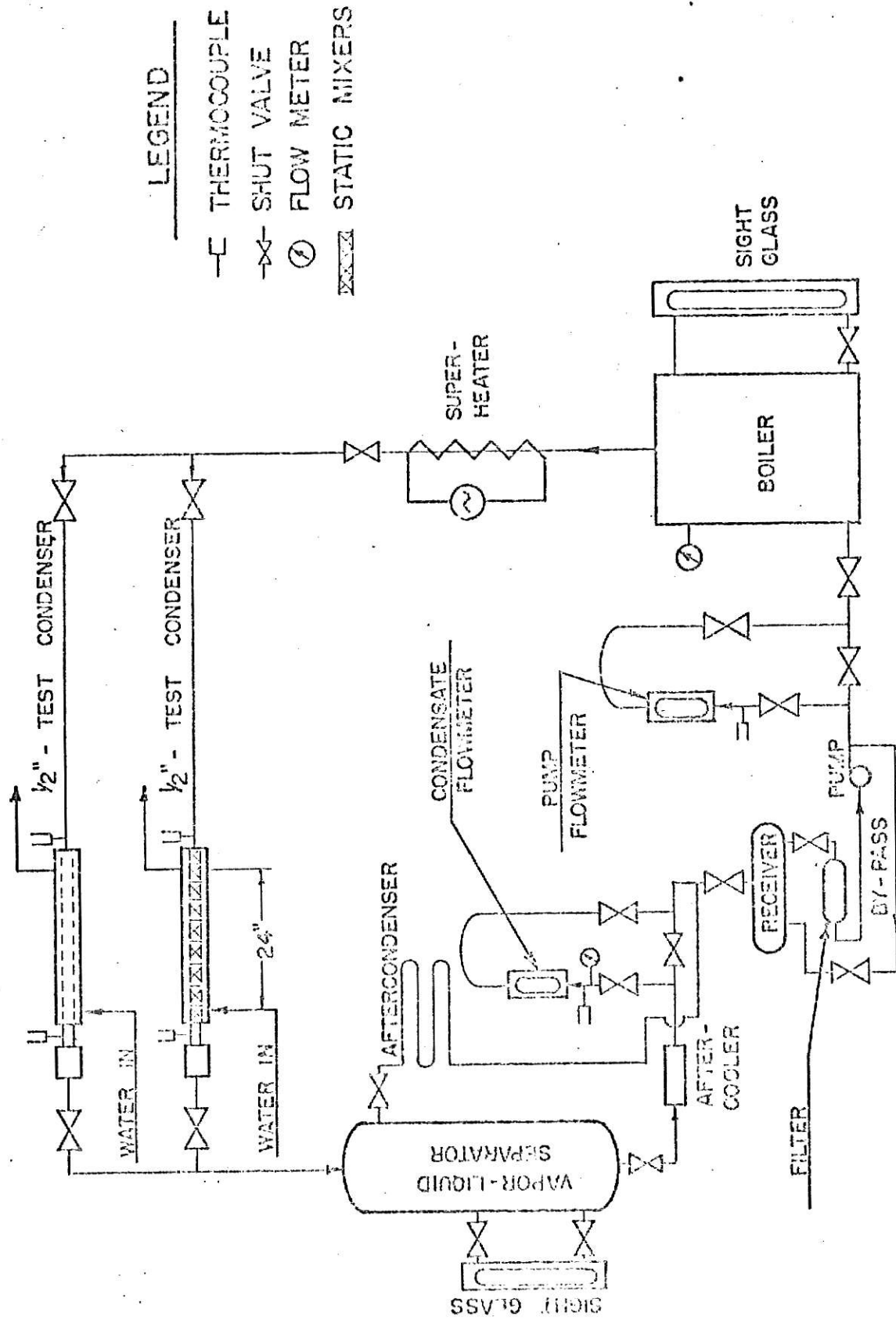


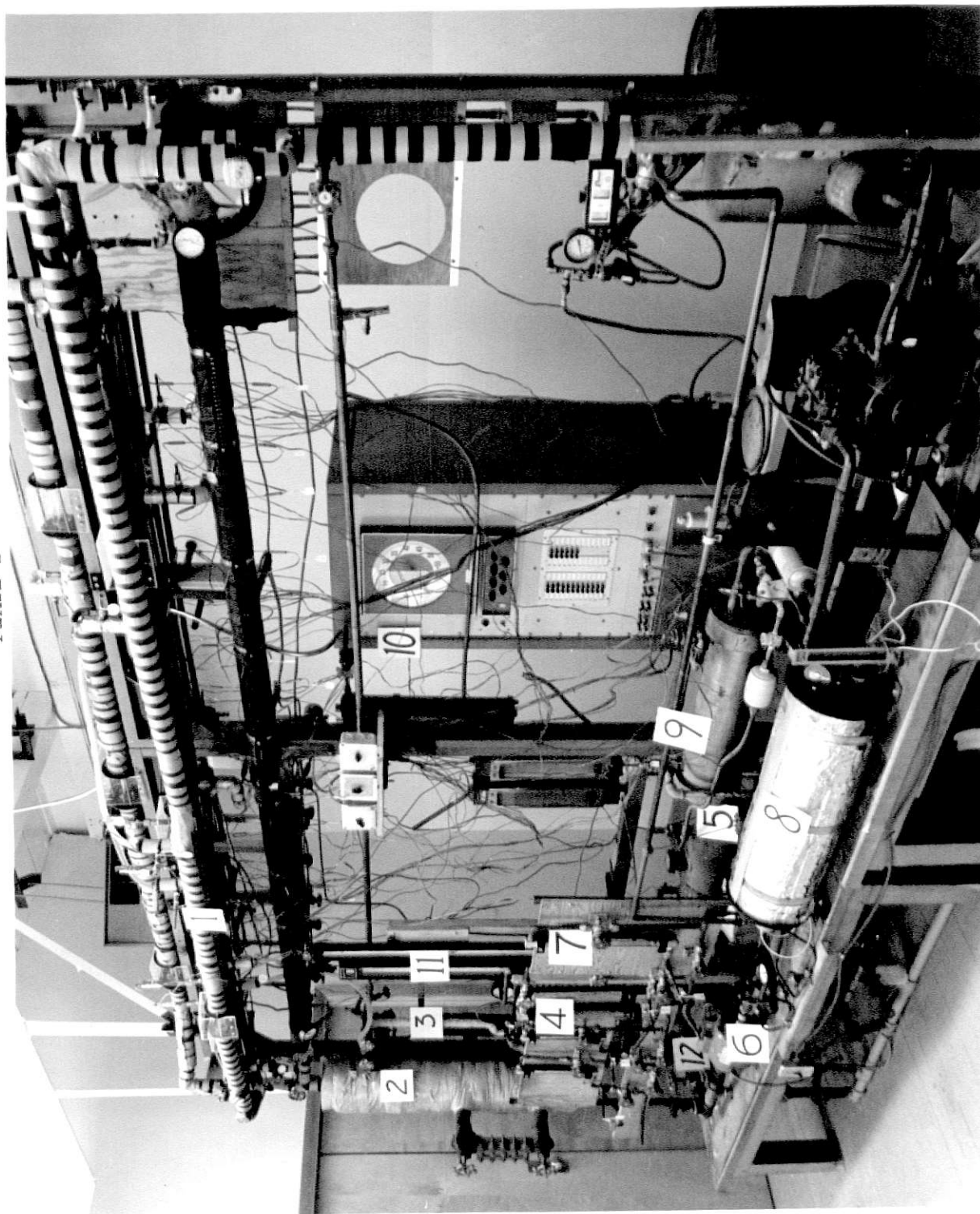
Fig. 1. A Schematic Diagram for the Refrigerant - 113 Flow Circuit.

EXPLANATION OF PLATE I

Photographic View of the Entire Test Facility

- | | |
|---------------------------|---------------------------------|
| 1. Test condenser | 7. Total circulation flow meter |
| 2. Vapor-liquid separator | 8. Vapor generator |
| 3. After condenser | 9. Superheater |
| 4. Condensate flow meter | 10. Multi-channel potentiometer |
| 5. Liquid receiver | 11. Mercury manometer |
| 6. Circulating pump | 12. Cooling water flow meter |

PLATE I



6. Liquid receiver.
7. Liquid circulating gear pump.

The auxillary components were:

1. Refrigerant drier-filter.
2. Pump by-pass valve.
3. Boiler sight glass.
4. Vapor-liquid separator sight glass.

The instrumentation of the test facility included:

1. Refrigerant flow meters of the variable area type.
2. Differential pressure cell for refrigerant pressure drop measurement.
3. Mercury in reservoir type manometer to indicate differential pressure cell output.
4. Multi-channel indicating potentiometer to measure thermocouple e.m.f. output.

Cooling Water Circuit

Figure 2 shows the flow diagram of the cooling water circuit. The cooling water circuit was designed to permit the control of the water flow rate and the inlet temperature of the cooling water to the test condenser. The control of the water flow rate was achieved by adjusting the openings of the valves in the circuit. The control of the cooling water temperature to the test condenser was achieved by mixing the return water from the after condenser, while the test condenser was in operation, with the city water in a mixing tank before the water was recirculated by the pump. The flow rate and the inlet temperature of the cooling water to the test

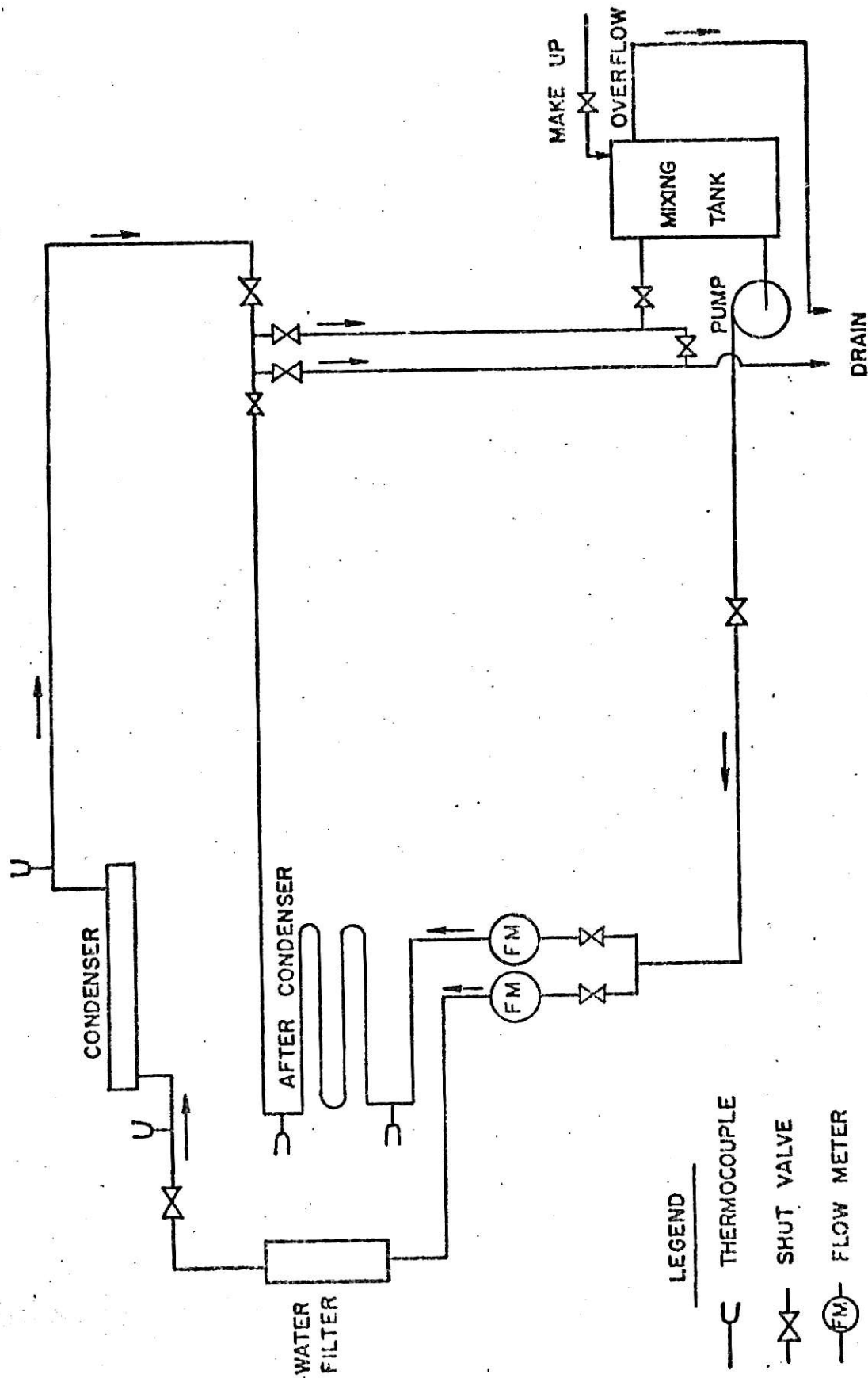


Fig. 2. A Schematic Diagram for the Cooling Water Circuit.

condensers controlled the condensing pressure and the cooling rate of the refrigerant. Water flow rate to the test condensers and after condenser were measured by the integrating type flow meters.

Test Condensers and Instrumentation

Two similar test condensers, one with static mixers and other without, with the same inside diameter of 1/2 in. were designed and constructed for this investigation. The two condensers were mounted horizontally in the test facility as shown in Fig. 1. While operating the facility, only one condenser was used at a time. ✓

Figure 3 shows a schematic diagram of one test condenser. Each test condenser consisted of one double pipe counter flow condenser and a transparent section. Figure 4 shows the construction details of one section of the test condenser. Each transparent section was made of standard clear, high pressure glass tubes of overall length of 8.5 in., and was fitted to the exit of the test condenser by two stuffing boxes. Three 3/8 in. diameter tie rods were used to secure together the two stuffing boxes at the ends of each transparent section to prevent leakage. Plate II shows a photographic view of the test condenser.

Each transparent section assembly in the two test condensers was leak-proof tested individually, for 24 hours, under pressure as high as 150 psi. Such a pressure was higher than any pressure encountered during the experiments. They were also leak-proof tested under vacuum as low as 1000 microns of mercury. The two test condensers and the two transparent sections were mounted individually on supports and checked individually by a liquid level to make sure that they were all mounted horizontally

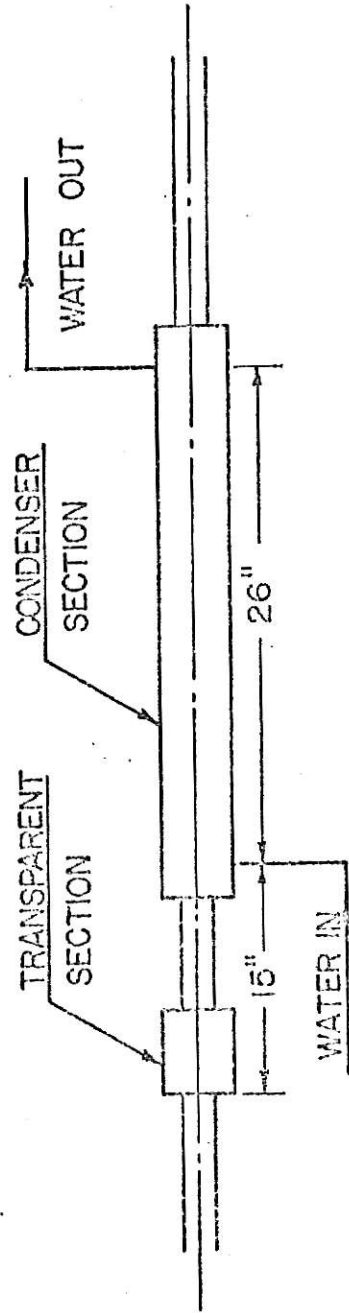


Fig. 3. A Schematic Diagram for one Test Condenser.

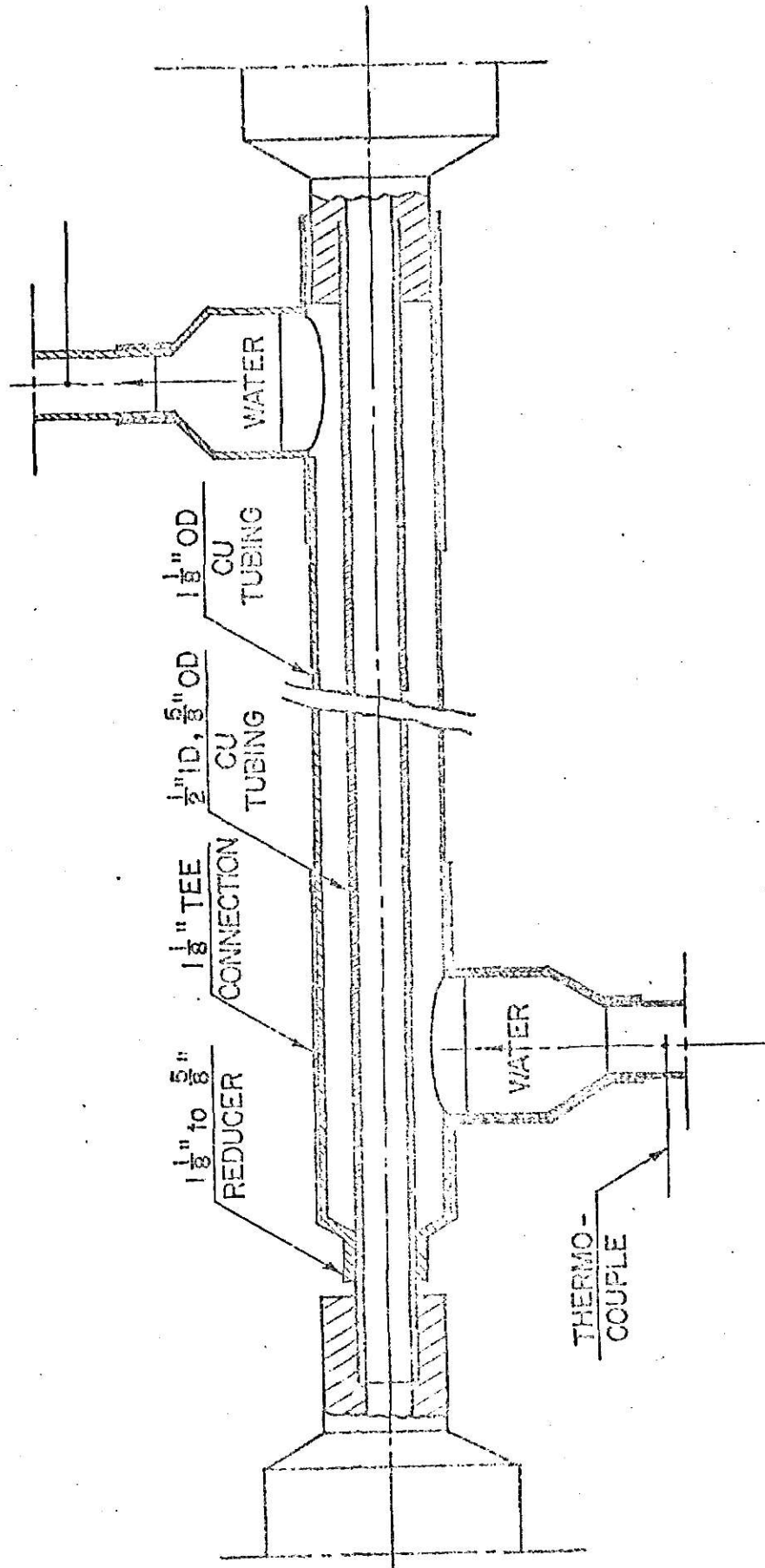


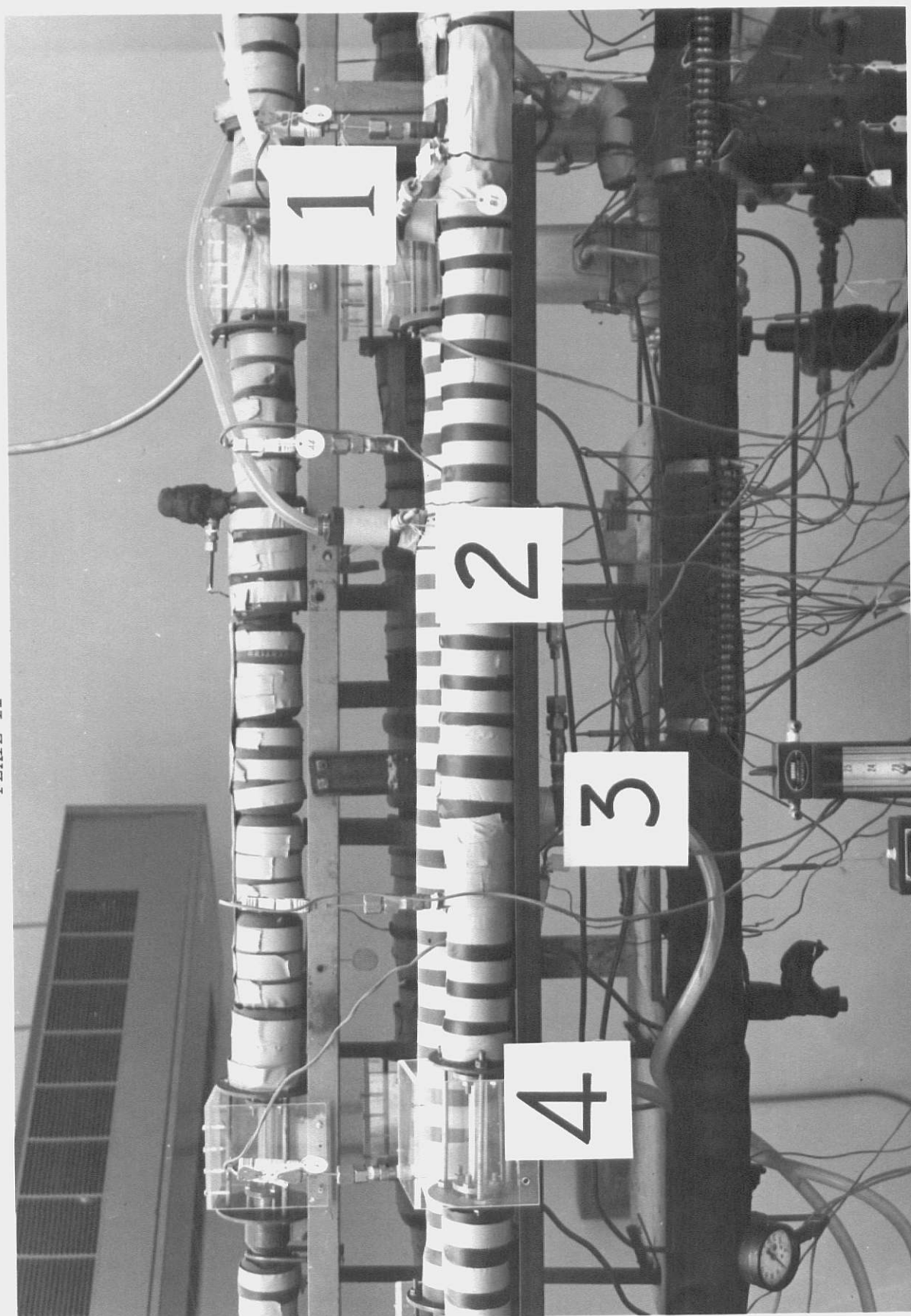
Fig. 4. Construction Details of a Condenser Section.

EXPLANATION OF PLATE II

Photographic View of the Test Condenser

1. Cooling water outlet
2. Condenser section
3. Cooling water inlet
4. Visual section

PLATE II



and coaxially aligned before welding them together. An expansion joint was added at the exit of each condenser to accommodate any expansion, due to heating during operation. With the two condensers assembled, one with static mixers and the other without, they were checked again for leaks under pressurized and evacuated conditions. The entire system was then evacuated down to 1000 microns and charged with refrigerant from a refrigerant-113 tank.

Each of the test condensers was instrumented to measure the inlet and outlet temperatures of the cooling water. The inlet and outlet temperatures of the refrigerant to the entire test condensers were also measured. All temperatures were measured by copper-constantan thermocouples of 24 B&S gage wire. Inlet and outlet pressures of the refrigerants were also measured. Figure 5 illustrates the placement of the static pressure taps for pressure drop measurements. Copper tubing, 1/4 in. O.D., silver brazed into the pressure tap, was used to connect the pressure tap and a differential pressure transducer, as shown in Fig. 6. The pressure difference between the two pressure taps of the test condenser could be measured by this arrangement. A mercury in glass manometer measured the differential pressure output of the transducer. The differential pressure cell was calibrated such that the reading of the attached mercury in the glass manometer could be converted into a pressure difference between the two taps in inches of water. The differential pressure cell, together with the appropriate valves, were placed on the same horizontal plane as the pressure taps to minimize the effect of hydrostatic head.

All thermocouples were calibrated against the temperatures of boiling water and melting ice. Their e.m.f. output was measured by a Honeywell

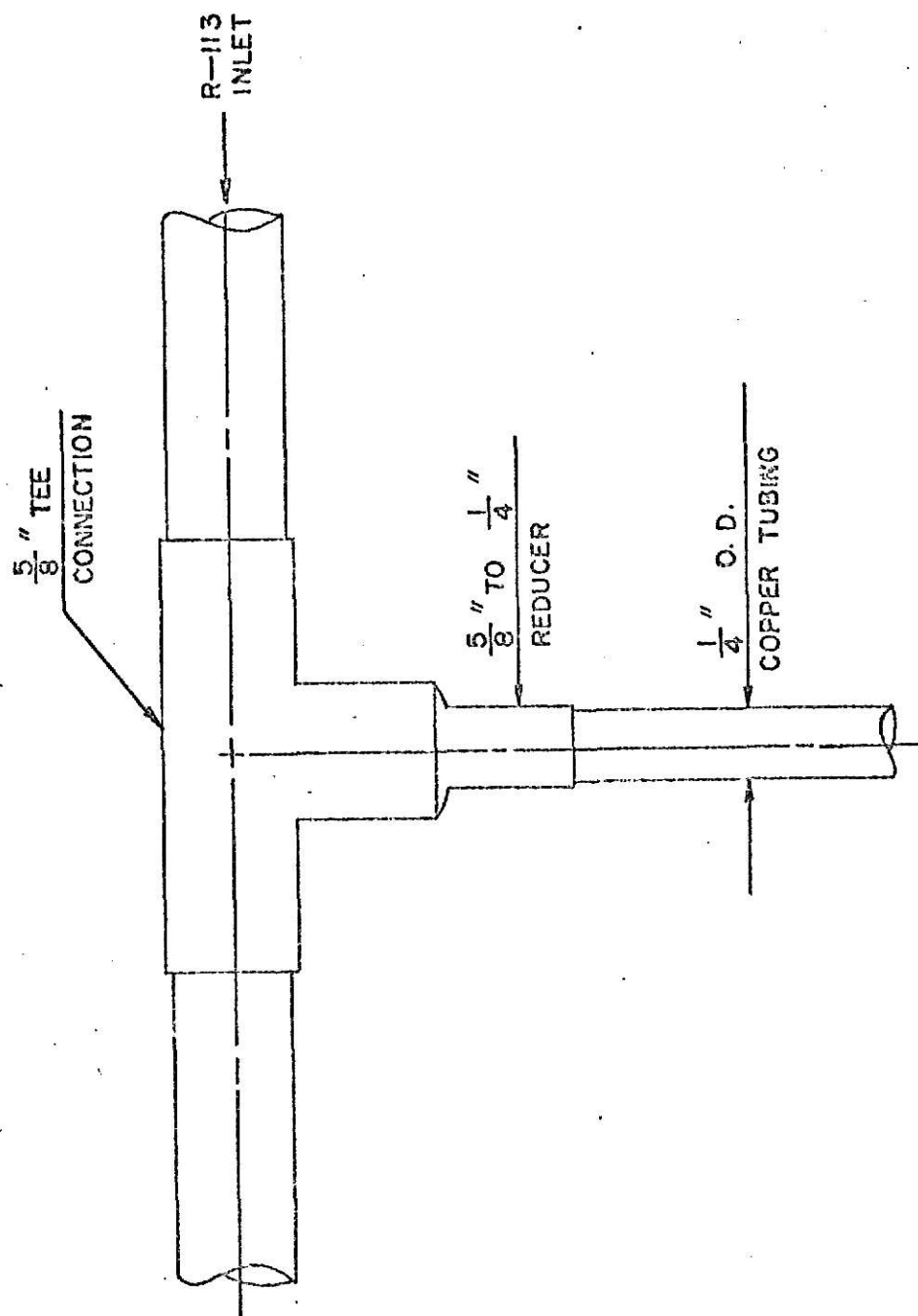


Fig. 5. A Schematic Diagram of Pressure Tap.

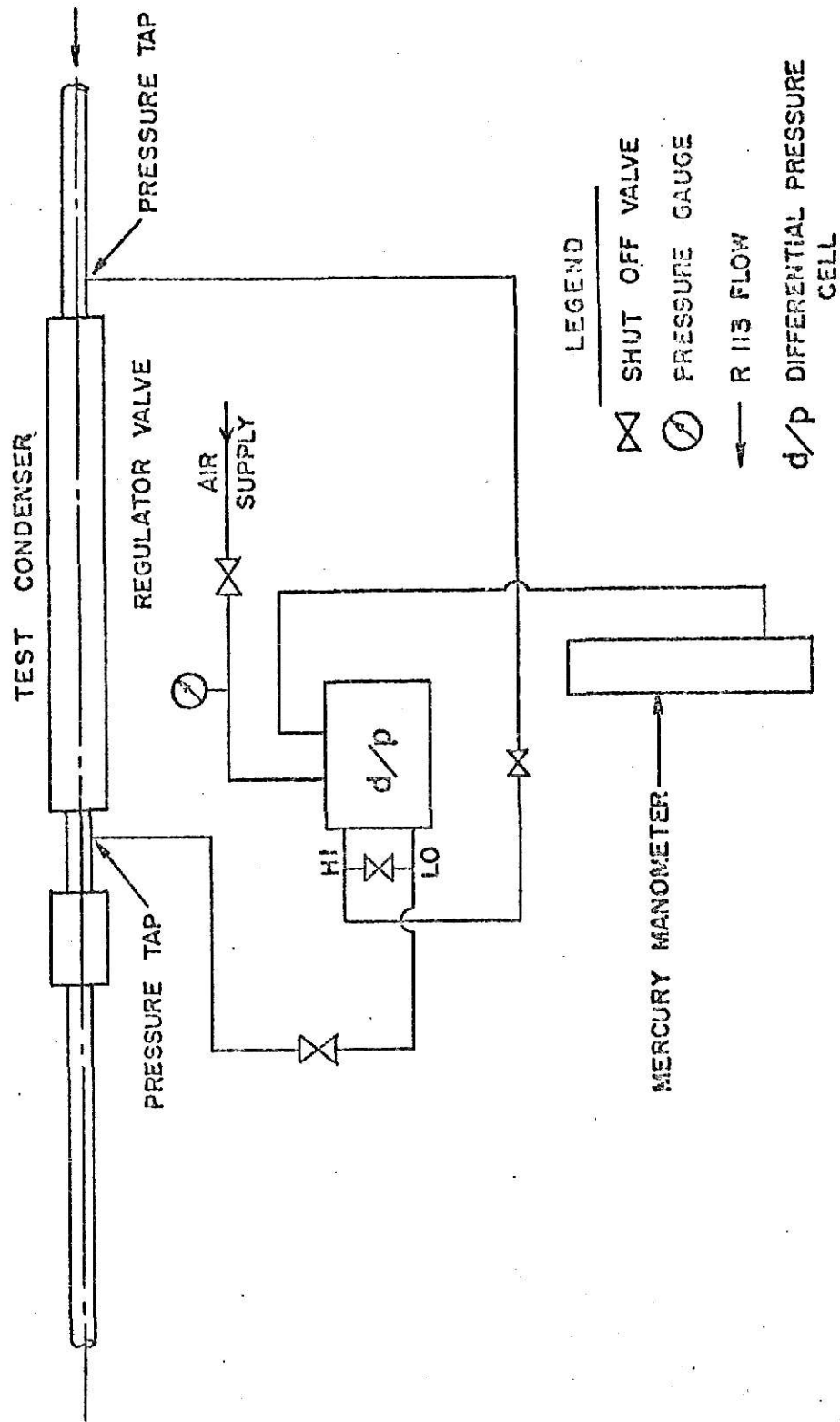


Fig. 6. Schematic Diagram for Pressure Drop Measurement.

Electronic - 18, multi-channel potentiometer. The input power to the vapor generator was controlled by a manual A.C. power variac with outputs ranging anywhere between 0 and 5.75 K.W. The circulation rate of refrigerant-113 was controlled by the pump by-pass valve and the input power to the vapor generator. An electrically heated superheater was installed between the vapor generator and the test condenser. The input power to the superheater was controlled by a manual A.C. power variac with outputs ranging anywhere between 0 and 1 K.W. Two different Fisher-Porter variable area type flow meters were used in the test facility. One flow meter was used to measure the flow rate into the vapor generator, which represented the total flow rate of the refrigerant through the test condenser. The second flow meter was used to measure the flow rate from the liquid-vapor separator which represented the amount of refrigerant condensed in the test condenser. The cooling water rate was measured by an integrating type flow meter mounted on the cooling water circuit. The two test condensers were insulated with 2.5 ins. thick rubber insulation. The vapor generator and the vapor-liquid separator were also insulated with 3/4 in. thick fiber glass insulation.

Differential Pressure Cell Calibration

The d/p cell used was Foxboro Type 13A. It was calibrated according to the manufacturer's recommended calibration procedure before it was connected to the test condenser. A linear least squares regression correlation was used to obtain the calibration curve given by Eqn. (1). Figure 7 shows a plot of the calibration curves.

$$Y = 2.192X - 13.437$$

(1)

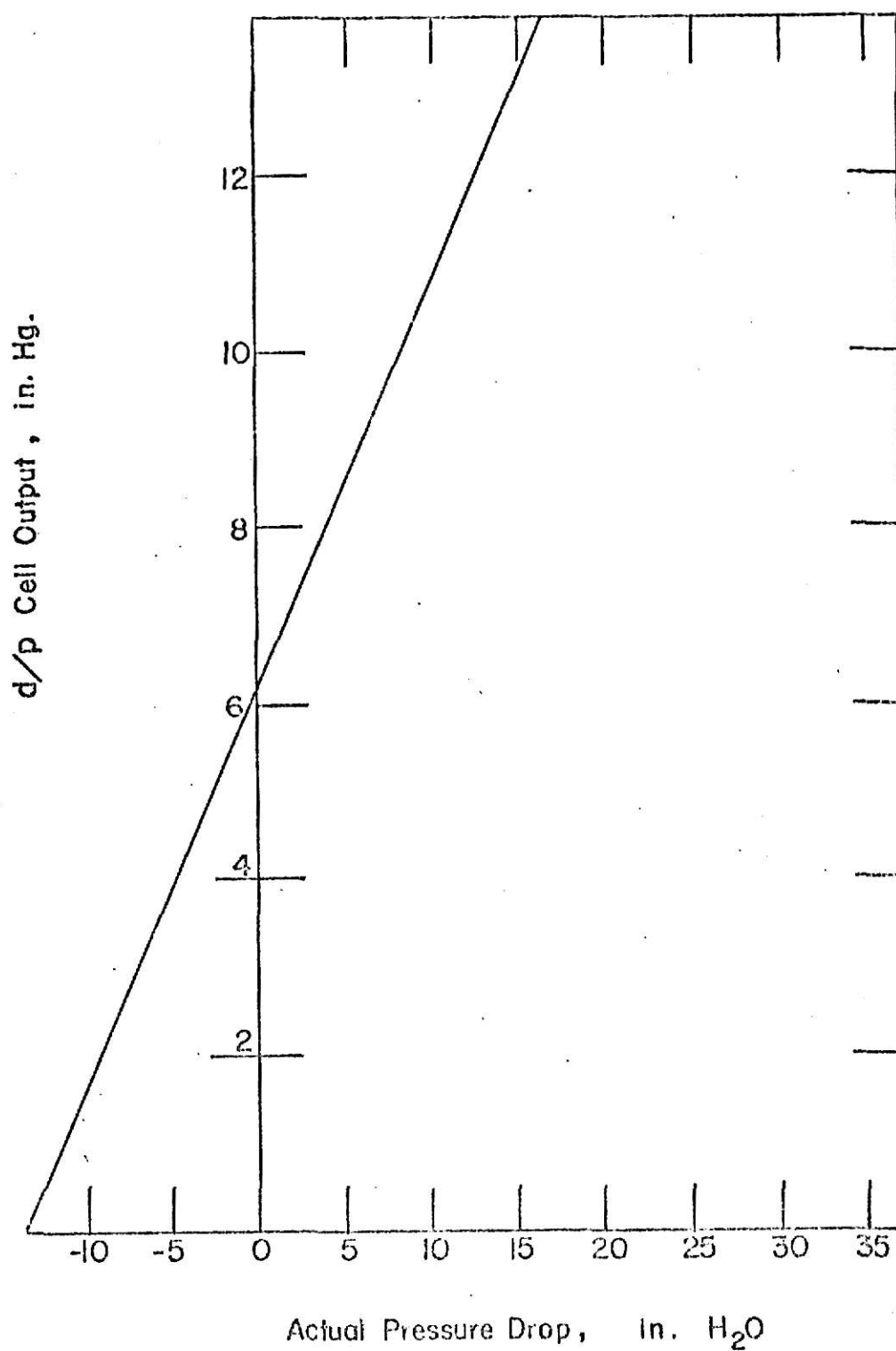


Fig. 7. d/p Cell Calibration Curve.

where

Y = pressure drop in inches of water

X = d/p cell output in inches of mercury

A total of 110 points was used to calibrate the pressure cell.

The Static Mixer Unit

The static mixer unit which has been developed recently has no moving parts, is placed in-line, and results in the continuous mixing of the moving stream. The unit is constructed of a number of short elements of right and left-hand helices. These elements are alternated and oriented so that each leading edge is at 90° to the trailing edge of the one ahead. The element assembly is then inserted inside the tube. Figure 8 shows a photographic view of the right and left-hand elements and the static mixer unit inside the tube. The essential features whereby particularly efficient mixing of two streams in tube is achieved may be seen by reference to Fig. 9.

The material used to fabricate the static mixer unit in this study is stainless steel. Each static mixer element is $3/4$ in. in length and $7/16$ in. in diameter. The static mixers used in this investigation were supplied by the Kenics Corporation.

Principles of Static Mixers Operation

When a fluid stream is passed through the mixer, two unique mixing actions operate simultaneously in the unit, namely, flow division and radial mixing, resulting in nearly plug flow characteristics.

(a) Flow Division:

At the leading edge of each element the flow divides and follows the semi-circular channel created by the element shape. At each succeeding

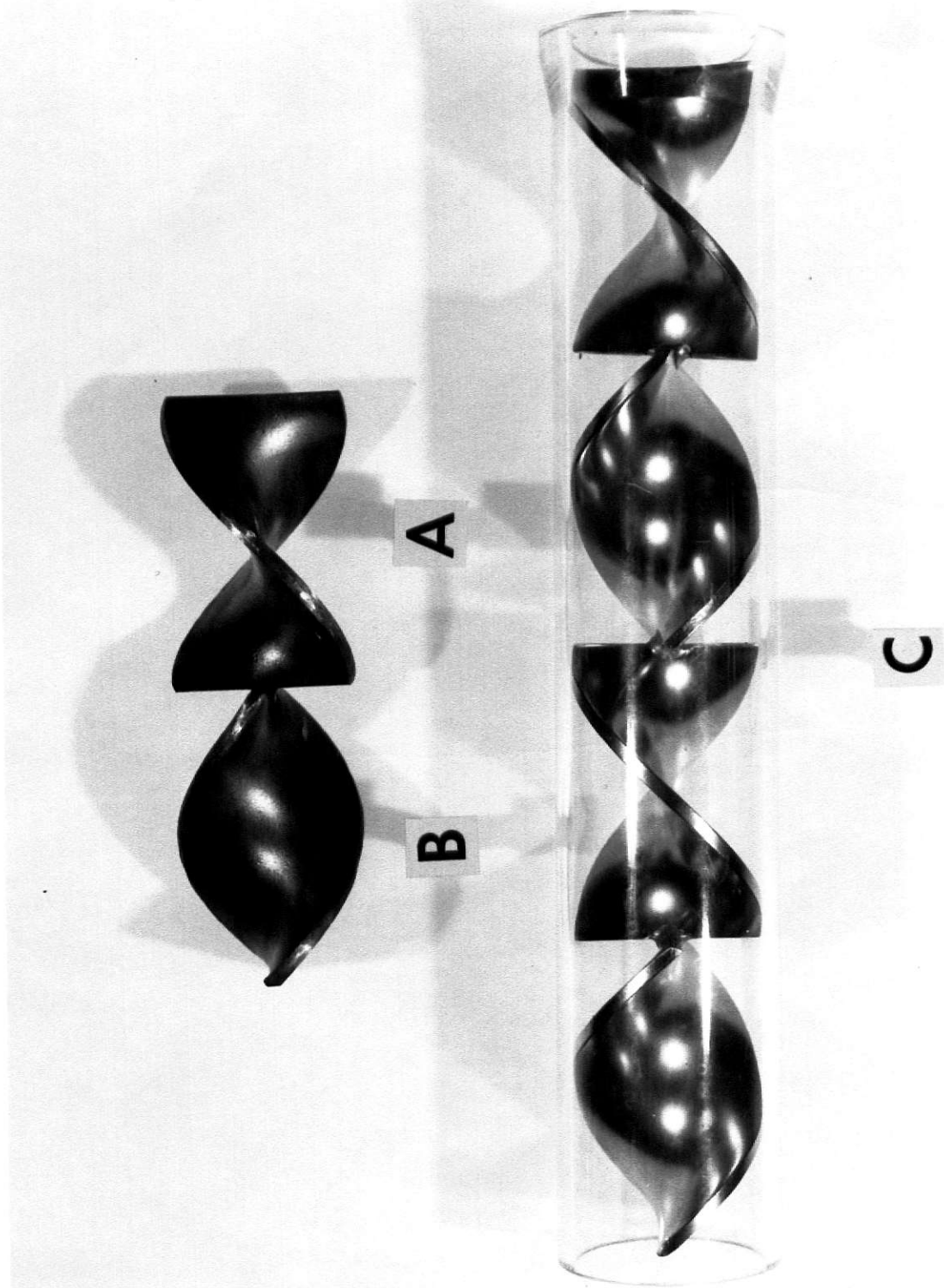


Fig. 8. Photographic Views of (A) Left-hand Element (B) Right-hand Element (C) the Static Mixer Unit Inside the Tube.

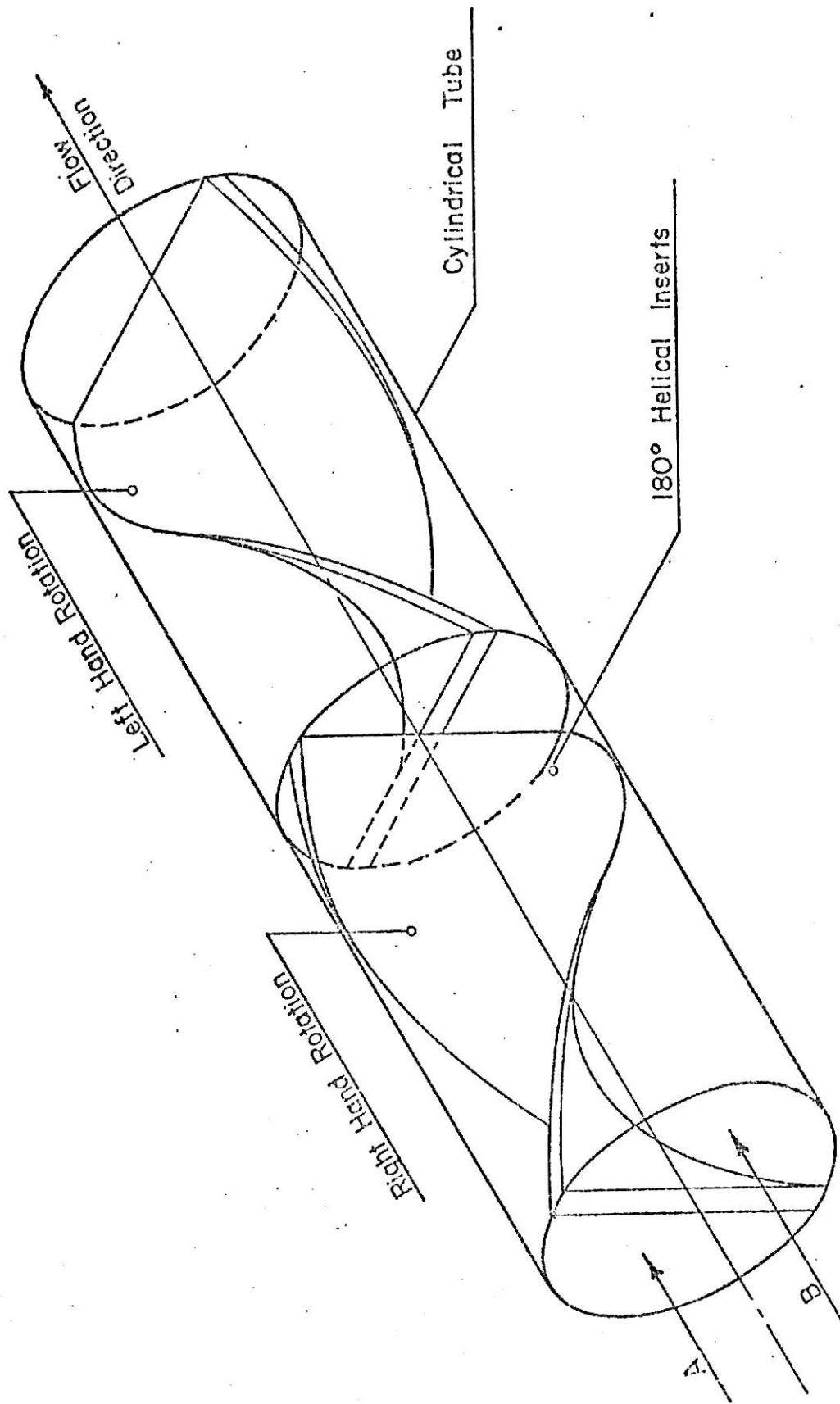


Fig. 9. Essential Features of the Kenics Static Mixing System.

element the two flows are further divided, resulting in an exponential progression of flow division. The progression is described by the formula:

$$S = 2^n$$

where S is the number of striations produced and n is the number of elements in the unit. After the flow has passed through 20 elements, over a million striations (2^{20}) exist. Figure 10 shows division of flow in the mixer.

The thickness of striations, d , is defined as:

$$d = \frac{D}{2^n}$$

where D is the inside diameter of the unit. Figure 11 shows the thickness of striations vs. the number of mixer elements for various static mixer unit diameter [22].

(b) Radial Mixing:

In either laminar or turbulent flow, if the static mixer is used as a device for the enhancement of heat transfer, rotational circulation of the flowing stream around the hydraulic center of each semi-circular channel causes radial mixing of the stream, thus forcing the stream from the center to the outer wall of the channel. At the same time, the flow reverses its rotation at each element junction due to the alternate right and left-hand alignment of the elements. Figure 12 shows the rotation of flow in each element and at the interface of the two elements. The overall effect of radial mixing is to cause the stream to be continuously and completely inverted radially so that particles entering at the center of the stream are forced to the outer wall and back again on a continuous basis. Because

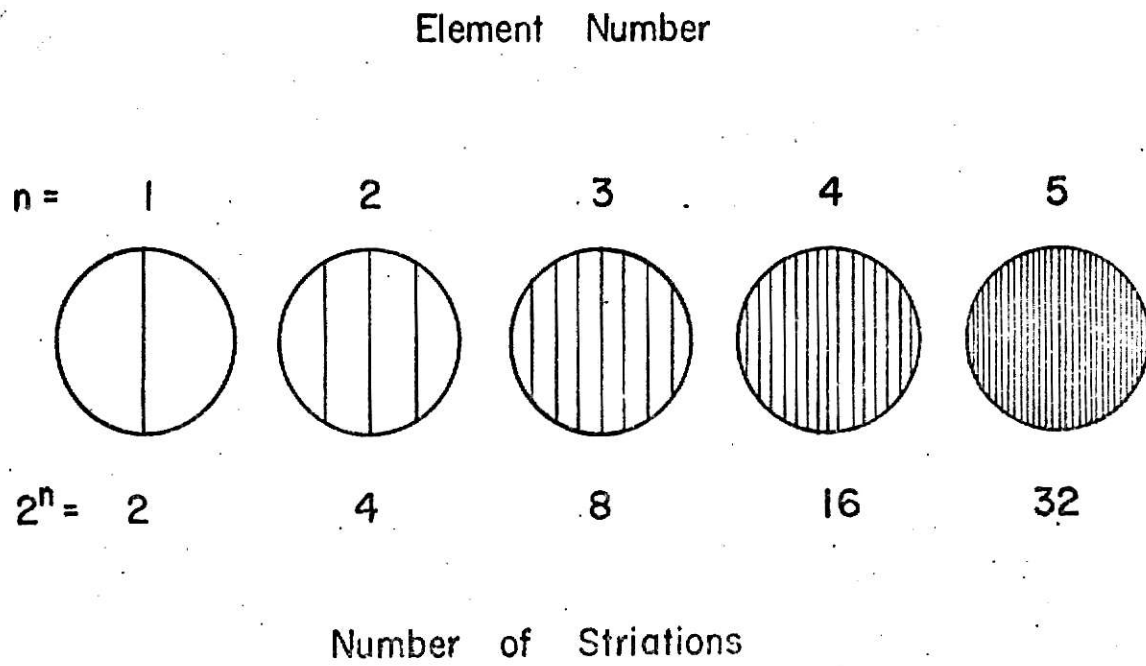


Fig. 10. Flow Division in the Static Mixer Unit.

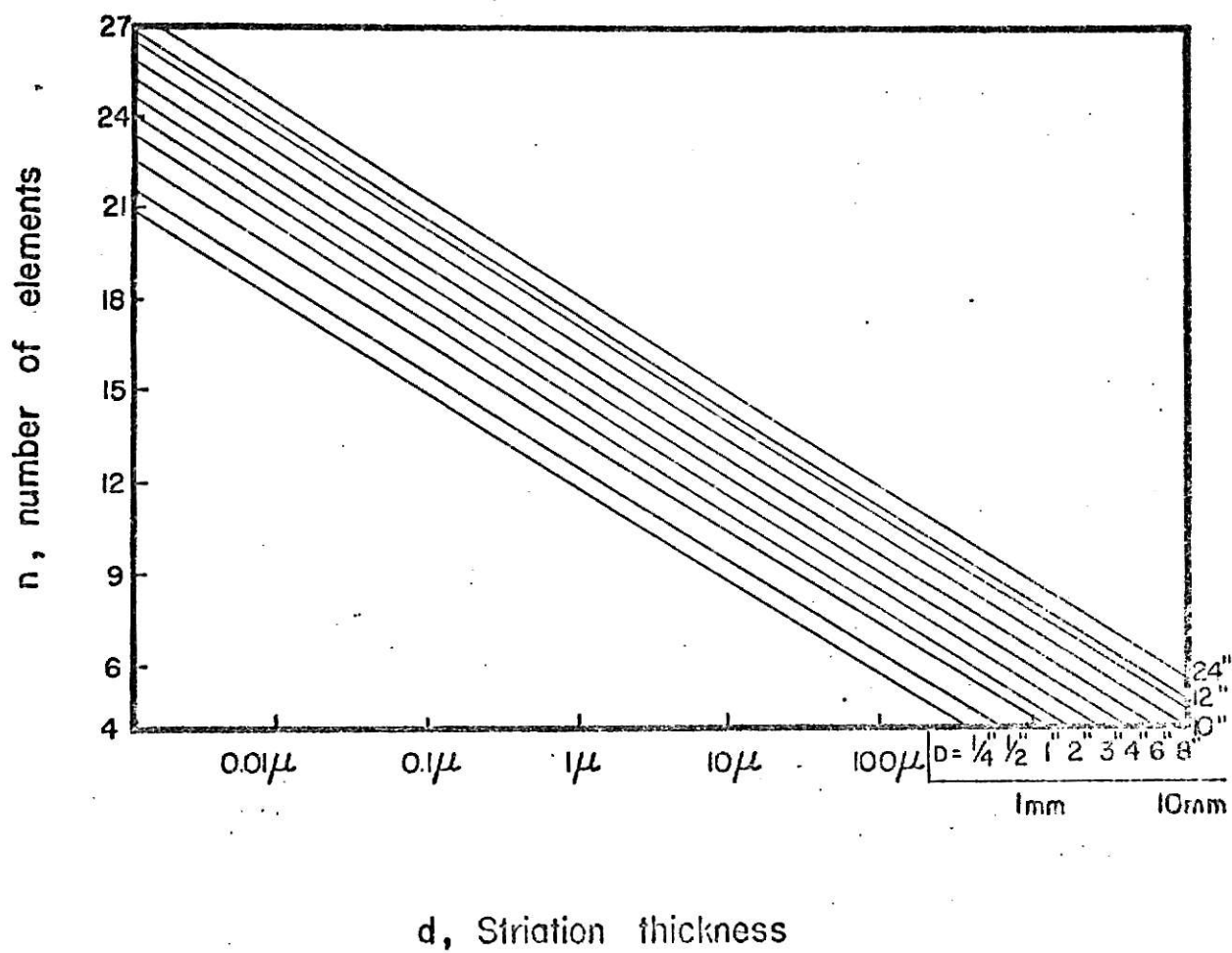
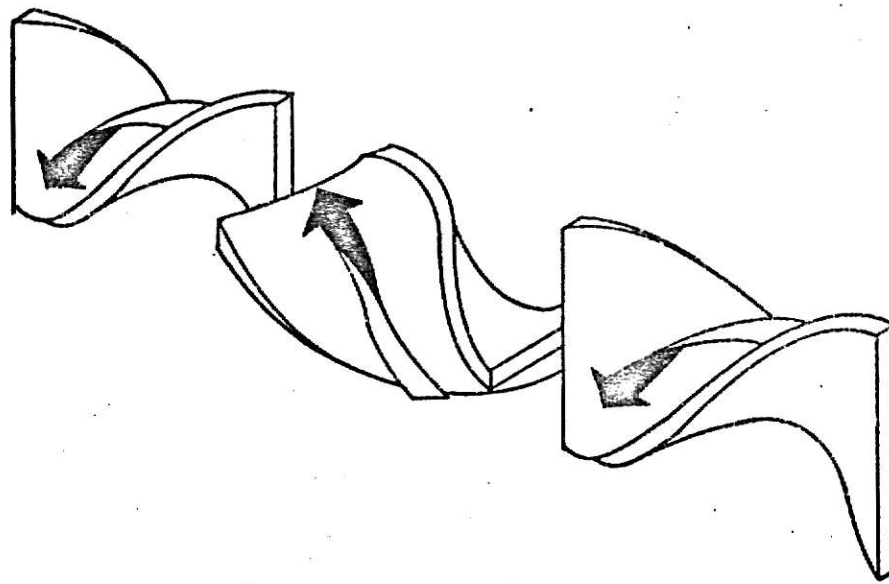


Fig. II. Striation Thickness vs. No. of Static Mixer Elements.
[22]



Material Rotation

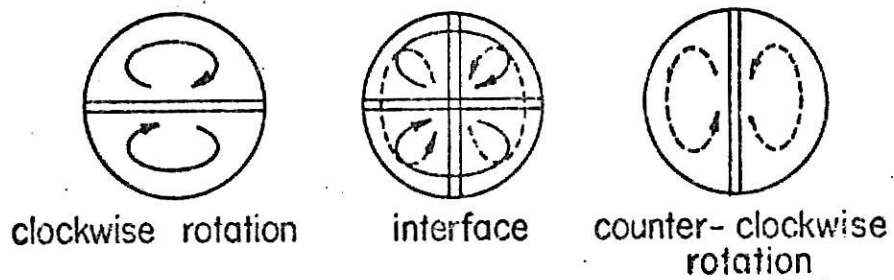


Fig. 12. Radial Mixing in the Static Mixer Unit.

of thorough radial mixing, the radial gradients in temperatures, velocity and composition are virtually minimized. Thus this mixing should give rise to enhancement of the heat transfer rate from the interior of the mixer to the wall or vice versa.

In order to test the effect of number and arrangement of the static mixer elements on the heat transfer and pressure drop characteristics of condensers, six different arrangements of the mixer elements were tested. These were:

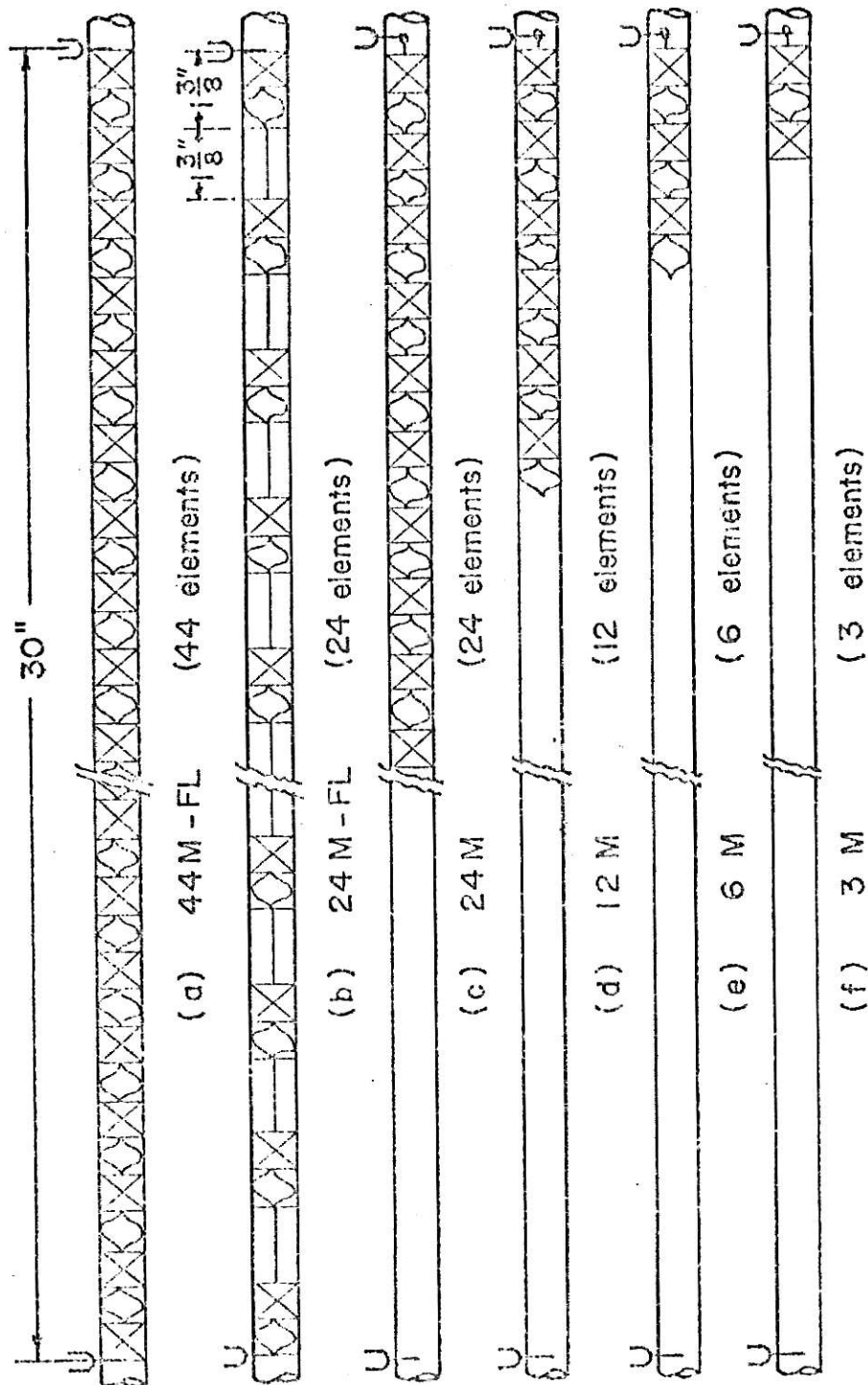
1. 44M-FL mixers (44 elements) to cover the entire length of the condenser.
2. 24M-FL mixers (24 elements) where the elements were spaced to cover the entire length of the condenser.
3. 24M mixers (24 elements) where the elements were arranged to occupy the upstream half of the condenser.
4. 12M mixers (12 elements).
5. 6M mixers (6 elements).
6. 3M mixers (3 elements).

In arrangements 4 through 6, the mixer elements were placed near the condenser inlet. The 24M-FL and 24M mixers have the same number of mixer elements. In the first, the mixer elements were separated at every two elements with a spacer, which was a stainless steel wire with 0.031 in. in diameter, having the same length as two elements, while in the second they were placed in a continuous manner, in which the elements were connected together directly. The ~~24M-FL~~ ^{44M-FL} and ~~24M~~ ^{24M-FL} mixers were held in place by the thermocouple sheath at each of the condensers. In the remaining four arrangements, the mixer's

assemblies were held in place by a stainless steel wire loop hooked around the upstream thermocouple in the condenser. A schematic diagram of the six different arrangements of mixers inside the condenser tube are shown in Fig. 13. (a) - (f).

Testing Procedures

The test started by filling the cooling water mixing tank to the overflow level, then the circulating water pump was put into operation. The liquid level inside the vapor-generator was checked through the sight glass attached to it, to ensure that the electric heaters were completely immersed in the liquid. Liquid had to be pumped from the liquid receiver to the vapor generator at the start of some tests to ensure that the desirable liquid level in the vapor generator was achieved. The desired amount of electric power was then applied to the heaters inside the vapor generator. This power was controlled by a manual power variac and its amount was proportional to the total refrigerant mass flow rate. When the liquid level inside the vapor generator started to decrease, the refrigerant circulating pump was put into operation, and the by-pass valve of the pump was adjusted to keep a steady liquid level inside the vapor generator. Then the electric power was applied to the superheater to insure superheated refrigerant at the condenser inlet. The adjustments of the electric power applied to the vapor generator, the by-pass valve opening of the refrigerant circulating pump, the electric power input to the superheater, the cooling water flow rate, and the cooling water inlet temperature to the test condenser were used to change the refrigerant mass flow rate, saturation pressure, inlet superheat, and cooling rate from one run to another. Steady



M stands for mixer, FL stands for full length

Fig. 13. A Schematic Diagram of 6 Different Arrangements of Mixers in Tubes.

state conditions were established when the readings of all pressure, temperature, and flow rate measuring instruments and the liquid levels in the vapor-liquid separator and the vapor generator remained steady for an extended period of time. Steady state conditions could be reached in any run within two hours; however, the system was allowed to run for a period of three hours before any readings and visual data were recorded.

During each run, only one test condenser with or without mixers was in operation. The following measurements were taken after reaching the steady state:

1. Water flow rate to the test condenser.
2. Refrigerant's inlet gage pressure.
3. Refrigerant's inlet and exit temperatures in millivolts.
4. Inlet and exit coolant temperatures of the test condenser section.
5. Flow rates of the two refrigerant flow meters in gallons/minutes.
6. Pressure difference of the d/p cell output in inches of mercury.

In addition to the above readings, the ambient temperature and the barometric pressure were recorded during each run. Also, for each run the flow pattern was observed, sketched, and a written description was recorded.

After recording all the necessary data for runs with one arrangement of static mixers in the test condenser, the condenser section was opened and the static mixers were rearranged. After each rearrangement, the same procedures, described earlier, of assembly, testing against leaks, evacuating and recharging were followed. Measurements of heat transfer

and pressure drop for the condenser without mixers were made on the separate condenser which was mounted in parallel to the condenser with static mixers.

CHAPTER IV

EXPERIMENTAL RESULTS AND DISCUSSION

Experimental Results

Seven sets of average heat transfer coefficient and pressure drop data were taken during the present investigation. Six sets were for the different mixers arrangements, and one for the condenser without the mixers. Table I summarizes the range of operating conditions and the number of experimental runs for each data set.

Fifteen measurements were recorded for each individual run. These were: ambient temperature, barometric pressure, input power to boiler and superheater, water inlet and outlet temperatures, water flow rate, refrigerant inlet and outlet temperatures, condensate flow meter temperature, circulating pump temperature, condensate flow rate, circulating pump flow rate, refrigerant inlet pressure, and the d/p cell output. The raw experimental data of all the runs, such as temperatures in millivolts and flow rates in gallons/minute, were fed into a digital computer program. The computer program reduced the data into useful form, and computed operating conditions, heat balance errors, thermophysical properties of refrigerant and the average heat transfer coefficients. Refrigerant thermophysical properties were computed in the computer program from state equations developed by Downing et al. [48]. Tables B-1 and B-2 of Appendix B include a summary of the operating conditions and heat balance errors, respectively, of all the runs performed in this investigation. The values of saturation temperature

Table I. Ranges of Operating Conditions

No. of Mixers and Arrangement	No. of Runs	Condensing Temperature °F	Condensing Pressure psi	Gx10 ⁻⁴ lbm/hr ft ²	Inlet Super- heat °F	Water Inlet Temperature °F	Water Flow Rate (lbm/hr)x10 ⁻²	Overall Average Heat Flux (B/hr ft ²)x10 ⁻³
44M-FL	32	90.3-205	17-38	2.4-7.9	2-52.6	62-64.5	4.8-9.4	2.5-6.8
24M-FL	23	97-186.2	18-32	2.4-7.5	13.2-47.8	63.5-65.6	9.3	2.3-6.6
24M	23	98-189.1	21-33	2.4-7.6	17-50.1	62.2-66.1	9.3	2.5-6.8
12M	5	104.3-189.8	22.7-34.2	3.7-7.1	17.3-42	63.5-64.9	9.3	3.7-5.9
6M	5	141.1-185.2	23.8-35.2	3.7-7.1	14.6-38	63.7-64.6	9.3	3.6-6.0
3M	5	143-184.9	24.3-34.2	3.7-7.1	16.9-35.6	64.6-65.1	9.3	3.5-5.6
OM	27	99-185.2	20.4-38	2.4-7.8	5-44.3	62.2-65	9.3-9.4	2.5-5.8

listed correspond to the average pressure between the inlet and the outlet pressures of the test condenser. The values of the saturated liquid thermal conductivity of the refrigerant used in calculating the Nusselt number were evaluated at this saturation temperatures. The inlet superheat was calculated from the difference between inlet temperature and saturation temperature. The Reynolds number calculated was based on the total mass velocity, pipe diameter, and the saturated liquid viscosity. The total mass velocity was calculated from the total mass flow rate and the total cross-sectional area of the condenser's inner tube. The heat rate gained by the cooling water was calculated from the water flow rate and the overall temperature difference between inlet and outlet of the condenser. The rate of heat loss from the condensing fluid was estimated from the total mass flow rate of the refrigerant, and its overall enthalpy change between inlet and outlet. The heat balance error was calculated from the ratio of the difference between the heat gain rate of the coolant and heat loss rate of the refrigerant, to the heat gain rate of the coolant. Only heat transfer data with heat balance error within $\pm 12\%$ are reported in the present investigation. The static pressures at the inlet and outlet of the test condenser were obtained from the static pressure at the inlet of the test condenser, which was measured by a Heise pressure gage, and the pressure difference between the inlet and outlet pressure taps, which was measured by the d/p cell.

Table C of Appendix C lists the heat transfer and pressure drop parameters that were used in constructing the different heat transfer

and pressure drop correlation diagrams that will be discussed later in this chapter. The heat transfer coefficient was obtained from the logarithmic mean temperature difference. The tube diameter rather than a hydraulic diameter was used as the significant length dimension in the Nusselt and Reynolds numbers, to facilitate direct comparison of the heat transfer coefficients of the inner tubes with and without mixers. The dryness fraction at the exit of the condenser was computed from the ratio of the difference between the readings of the pump and the condensate flow meters, to the readings of the pump flow meter. The last column of Table C list the flow patterns observed during all the runs at the transparent sections. Five flow patterns were identified in this study at the exit of the condensers. The descriptions and characteristics of the flow patterns which were visually identified were (see Fig. 14):

1. Wavy Flow: The two phases were separated with the liquid flowing at the bottom of the tube. The difference in velocity between the vapor and the liquid caused the liquid surface to be wavy.
2. Slug Flow: The liquid flowed at the bottom of the tube, and the vapor flowed at the top with the surface of the liquid being wavy. Frequently the wave grew in size to wet the upper tube wall.
3. Semi-annular Flow: The flow had an appearance similar to the annular flow except the film thickness did not cover the entire wall periphery. A small portion of the upper half of the tube appeared dry while the liquid film thickness increased in the downward circumferential direction with the maximum film thickness being at the bottom.

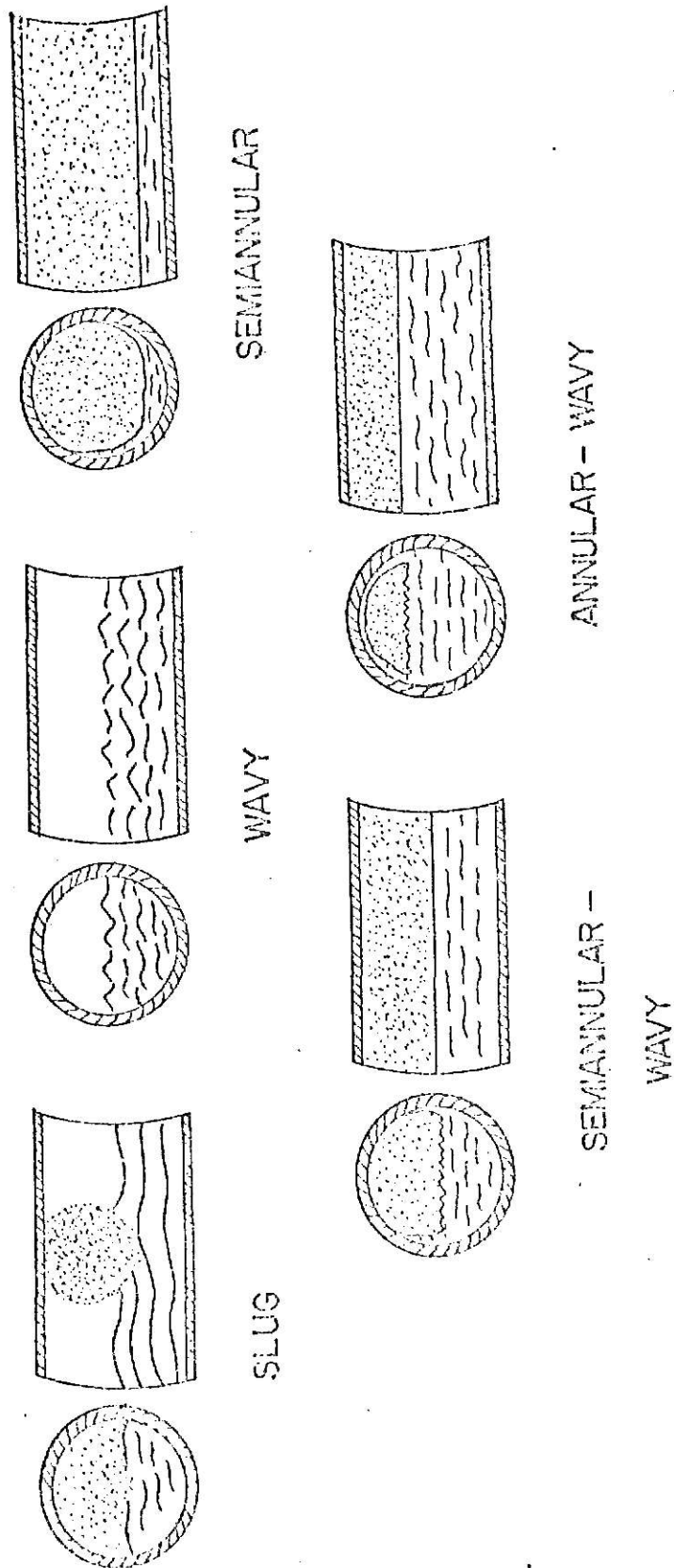


Fig. 14. Schematic Views of the Observed Flow Patterns.

4. Annular-Wavy Flow: The flow had an annular appearance, with the upper tube wall becoming occasionally dry and the thickness of the liquid film appearing larger all the time at the lower part of the tube.
5. Semiannular-Wavy Flow: The flow had the semiannular appearance except the liquid film thickness was bigger at the lower tube wall. Most of the upper tube wall appeared dry all the time.

Smooth Pipe Results

The overall heat transfer rate equation for the experimental test condenser is given by

$$q = U_i A_i \Delta T = \frac{A_i \Delta T}{\frac{1}{h_i} + \frac{A_i \ln(r_o/r_i)}{2\pi KL} + \frac{A_i}{A_o} \frac{1}{h_o}} \quad (2)^*$$

The outside film coefficient h_o was calculated from the following equation which was recommended by McAdams [49].

$$\frac{h_o}{c_{pb} G_w} \left(\frac{c_p \mu}{k} \right)_b^{2/3} \left(\frac{\mu_a}{\mu_b} \right)^{0.14} = \frac{0.023}{(D_e G_w / \mu_b)^{0.2}} \quad (3)$$

The mean temperature difference ΔT in Eqn. (2) is the logarithmic-mean temperature difference which can be evaluated by:

* All symbols used are defined in the nomenclature, Appendix A.

$$\Delta T = \frac{(T_s - T_{wi}) - (T_s - T_{wo})}{\ln[(T_s - T_{wi}) / (T_s - T_{wo})]} \quad (4)$$

The sensible heat part of the superheated vapor represented a small percentage of the total energy transfer from the condenser's tube in all the runs of this study. Also, there was a 2-in. cool-down section at the beginning of each test condenser to cool the superheated vapor to its saturation temperature. Therefore, the saturation temperature T_s , rather than the inlet superheat, was used in evaluating ΔT in Eqn. (4). A sample of the calculations is included in Appendix D.

Twenty seven smooth tube test runs were conducted. In ASHRAE Handbook of Fundamentals [32], the following equation was recommended for the correlation of heat transfer coefficients during condensation inside horizontal tubes.

$$(h_i D / K_l) / (Pr_l)^{1/3} \left(\frac{h_{fg}}{c_p \Delta T_i} \right)^{1/6} \left(\frac{\rho_l}{\rho_v} \right)^{1/3} = 0.1 \left(\frac{DG_v}{\mu_l} \right)^{2/3} \quad (5)$$

Equation (5) applies when

$$20,000 < \frac{DG_v}{\mu_l} \left(\frac{\rho_l}{\rho_v} \right)^{1/2} < 100,000$$

Seventy five percent of the experimental runs of the smooth pipe results fell into that range. Figure 15 shows a plot of the smooth tube results. In the figure, the ordinate is the left-hand-side of Eqn. (5), and the abscissa is Reynolds number. It is obvious that the results of the

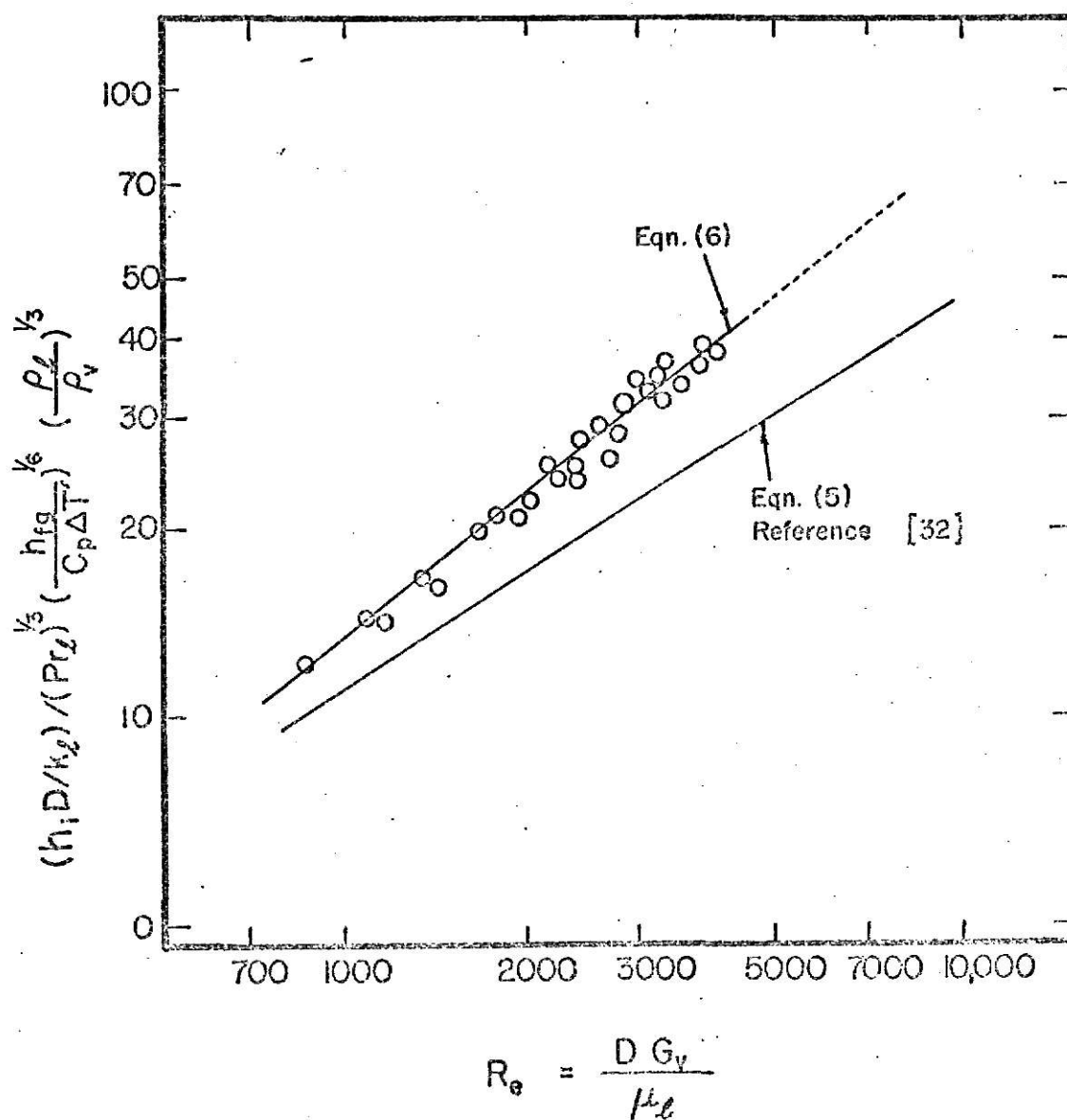


Fig. 15. Correlation of Smooth Tube Results with Eqn. of Reference [32].

present investigation gave higher values for the Nusselt number. On the same figure the least square linear regression correlation of the data is shown. The correlation equation is given by

$$(h_i D / K_\ell) / (Pr_\ell)^{1/3} \left(\frac{h_{fg}}{c_p \Delta T'} \right)^{1/6} \left(\frac{\rho_\ell}{\rho_v} \right)^{1/3} = 0.056 \left(\frac{DG_v}{\mu_\ell} \right)^{0.8} \quad (6)$$

Table II lists the values of the outside and inside film coefficients h_o and h_i of the smooth pipe data. In all runs the outside film coefficient h_o was higher than the condensation heat transfer coefficient h_i . On the basis of the results in Table II, the question arises concerning the validity of using the static mixers, inside the condenser's tube, to increase the heat transfer rate, and consequently reduce the size of the condenser's heat transfer area. Undoubtedly there are sometimes many factors that affect the decision of using any augmentative techniques such as the mixers. Some of these factors are: heat transfer rate increase to reduce the surface area, initial cost, pumping power, operating and maintainance costs, safety, and reliability. The increase in heat transfer rate and the possible reduction in surface area were the main concern of the present investigation. In the following section the attempt was made to relate the percentage increase in condensation heat transfer coefficient to the area reduction.

In the heat transfer rate expression given by Eqn. (2), the condenser's tube wall resistance is negligible compared to the inside and outside convection resistances. Also for thin wall condenser tubes, A_i/A_o is close to unity. In the present investigation this ratio was 0.8. By

Table II. The h_o and h_i Values of Smooth Pipe Tests

Run No.	h_o BTU/hr ft ² °F	h_i BTU/hr ft ² °F	Run No.	h_o BTU/hr ft ² °F	h_i BTU/hr ft ² °F
79	368.3	169.4	93	370.7	305.0
80	370.4	192.4	94	372.5	282.2
81	370.4	192.7	95	368.4	280.2
82	370.4	197.1	96	371.2	273.8
83	370.3	215.4	97	370.5	319.6
84	368.5	241.0	98	370.3	317.1
85	372.5	249.7	99	370.3	338.5
86	372.8	223.6	100	370.3	335.4
87	370.3	234.4	101	370.4	327.2
88	370.7	248.5	102	367.5	306.8
89	370.9	263.5	103	372.4	334.8
90	370.3	273.8	104	370.3	327.4
91	368.6	280.5	105	370.3	342.0
92	370.3	287.8			

neglecting the pipe wall resistance, Eqn. (2) can be rewritten as follows:

$$U_i A_i = \frac{A_i}{\frac{1}{h_i} + \frac{0.8}{h_o}}$$

Several cases are now considered.

Case I If $h_i \gg h_o$,

$$U_i A_i = \frac{A_i h_o}{0.8}$$

In such a case the outside resistance controls the heat transfer rate.

An increase of $x\%$ in h_o results in a certain area reduction as follows:

$$U_i A_{ix} = \frac{A_{ix}}{0.8} \left[1 + \frac{x}{100}\right] h_o = \frac{A_i}{0.8} h_o$$

where A_{ix} is the new inside surface area which transfers the same amount of heat rate. Therefore

$$\frac{A_{ix}}{A_i} = \frac{100}{100 + x} \quad (7)$$

Case II If $h_i \ll h_o$,

$$U_i A_i = A_i h_i$$

An increase of $x\%$ in h_i results in a certain area reduction as follows:

$$U_i A_{ix} = A_{ix} \left[1 + \frac{x}{100}\right] h_o = A_i h_i$$

Therefore,

$$\frac{A_{ix}}{A_i} = \frac{100}{100 + x} \quad (8)$$

Equations (7) and (8) are the same. From the above two cases, it is clear that the heat transfer promoter will be most beneficial when placed on the side of the heat transfer area with the lower film coefficient.

Case III If $h_i \approx h_o$,

In such a case it can be shown that the area ratio is given by

$$\frac{A_{ix}}{A_i} = \frac{180 + 0.8x}{180 + 1.8x} \quad (9)$$

Case IV If $h_i \approx \frac{1}{2} h_o$

In such a case it can be shown that

$$\frac{A_{ix}}{A_i} = \frac{140 + 0.4x}{140 + 1.4x} \quad (10)$$

Actual numerical values of the area ratio $\frac{A_{ix}}{A_i}$ are plotted in Fig. 16 for the above four cases as a function of the percentage increase in inside or outside film coefficients which could be caused by an augmentative technique. The smooth pipe results in Table II show that h_i was always less than h_o and the ratio h_i/h_o was between 0.5 and 1.0. According to the results of cases III and IV in Fig. 16, the use of the static mixer to improve h_i may be justifiable.

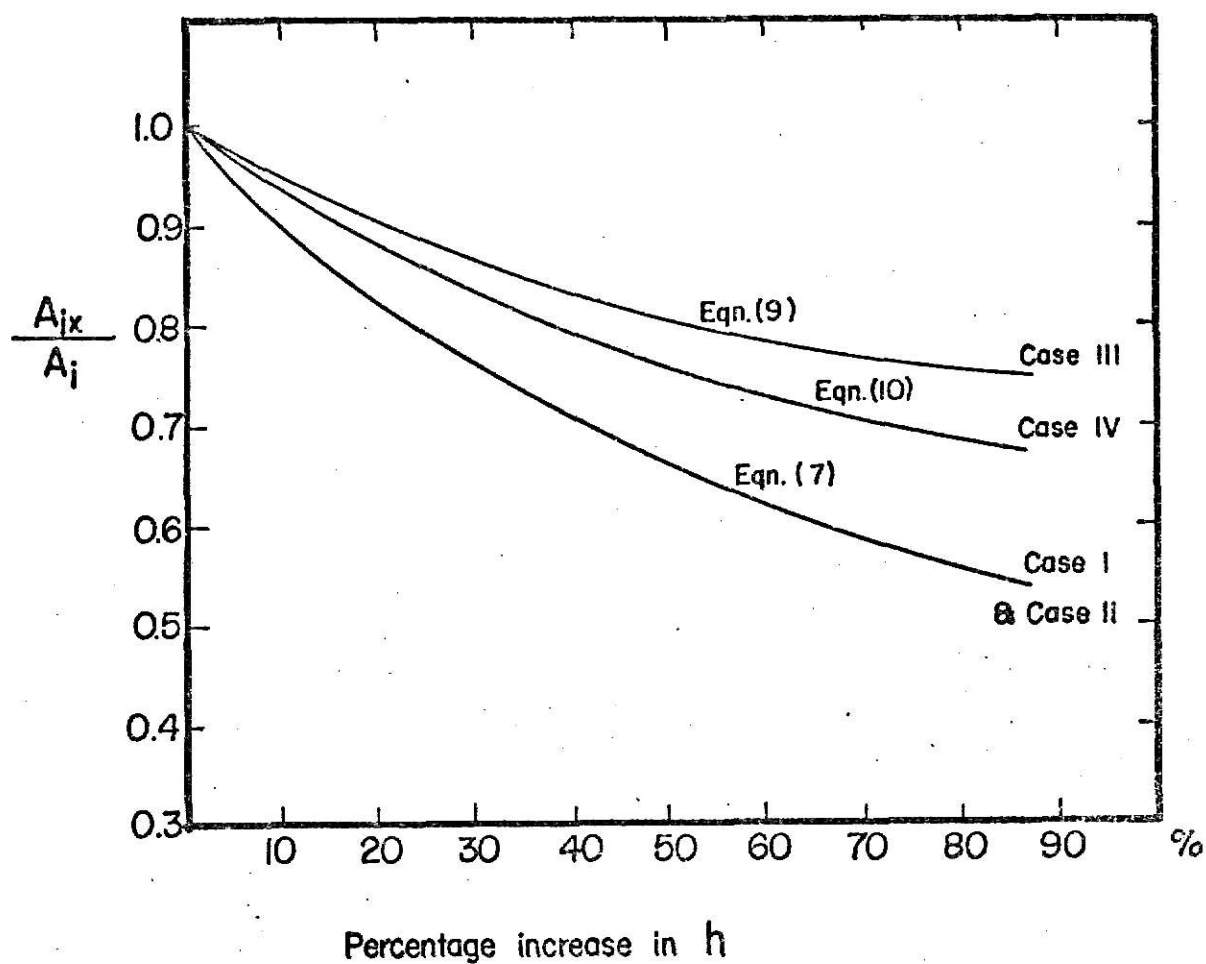


Fig. 16. Reduction in Exchanger's Size with the Percentage Increase in h_i or h_o .

Results of Condensation with Static Mixers

At the present time there are no generally acceptable criteria for the evaluation of the heat transfer results for enhanced heat transfer surfaces during two-phase flow. In general any criterion should be directed toward a certain objective, and subject to certain constraints. In keeping with this principle, Bergles et al. [50] developed eight performance evaluation criteria which define the merits of surfaces with promoters to augment heat transfer in single-phase flow. Four of these criteria are directed towards the use of promoters for improvement of existing exchangers. Therefore, the basic geometry is fixed. The remaining four criteria are directed towards evaluating the advantages of using promoters in the design of a new heat exchanger. Therefore, the length and number of tubes are unrestricted and the objective is to reduce the exchanger's size. A summary of these criteria is shown in Table III. Interested reader is referred to reference [50] for additional details.

The attempt was made in the present investigation to adapt three of Bergles et al. [50] criteria to evaluate the results of condensation heat transfer with the static mixers. In the first performance evaluation criterion of Bergles et al. [50], the ratio of the heat transfer coefficient with promoter, h_m , to its value without promoter, h_s , was compared at the same Reynolds number under these constraints: fixed geometry (diameter D , length L , and number of tubes N), fixed mass flow rate of fluid W , fixed fluid inlet temperature, and fixed ΔT . Such a criterion can be expressed as follows:

Table III. - Summary of Criteria Evaluation

Criterion Number	Fixed					Objective		
	Basic Geometry	Flow Rate	Pressure Drop	Pumping Power	Heat Duty	Increase Heat Transfer	Reduce Pumping Power	Reduce Exchanger Size
1	x	x				x		
2	x		x			x		
3	x			x		x		
4	x				x		x	
5				x	x			x
6			x		x			x
7		x			x			x
8		x	x		x			x

$$q_m/q_s = (h_m/h_s)_{W,\Delta T,L,D,N,T_{in}} \quad (11)$$

where

q_m = heat transfer rate with promoter

q_s = heat transfer rate without promoter

In Eqn. (11), values of W and T_{in} were their corresponding values on the heat transfer surface with the promoter. In the present investigation the ratio (h_{im}/h_{is}) was compared at constant condensing fluid Reynolds numbers under these constraints: fixed geometry, fixed mass flow rate of coolant M_w , and fixed coolant inlet temperature T_{wi} . In such a case let

$$R_1 = (h_{im}/h_{is})_{M_w,T_{wi},D,L,N,Re} \quad (12)$$

It should be pointed out that every effort was made in the present investigation to maintain the coolant flow rate and its inlet temperature constant in all test runs with and without mixers.

Before plotting the ratio R_1 versus Reynolds number for the different mixers arrangements, it is significant to examine the effect of using the different static mixers arrangements on the condensation heat transfer coefficient. Figure 17 shows a plot of Nusselt number versus Reynolds number for all experimental data points with and without static mixers. Unless otherwise stated the following symbolic notations were used for the different mixer arrangements.

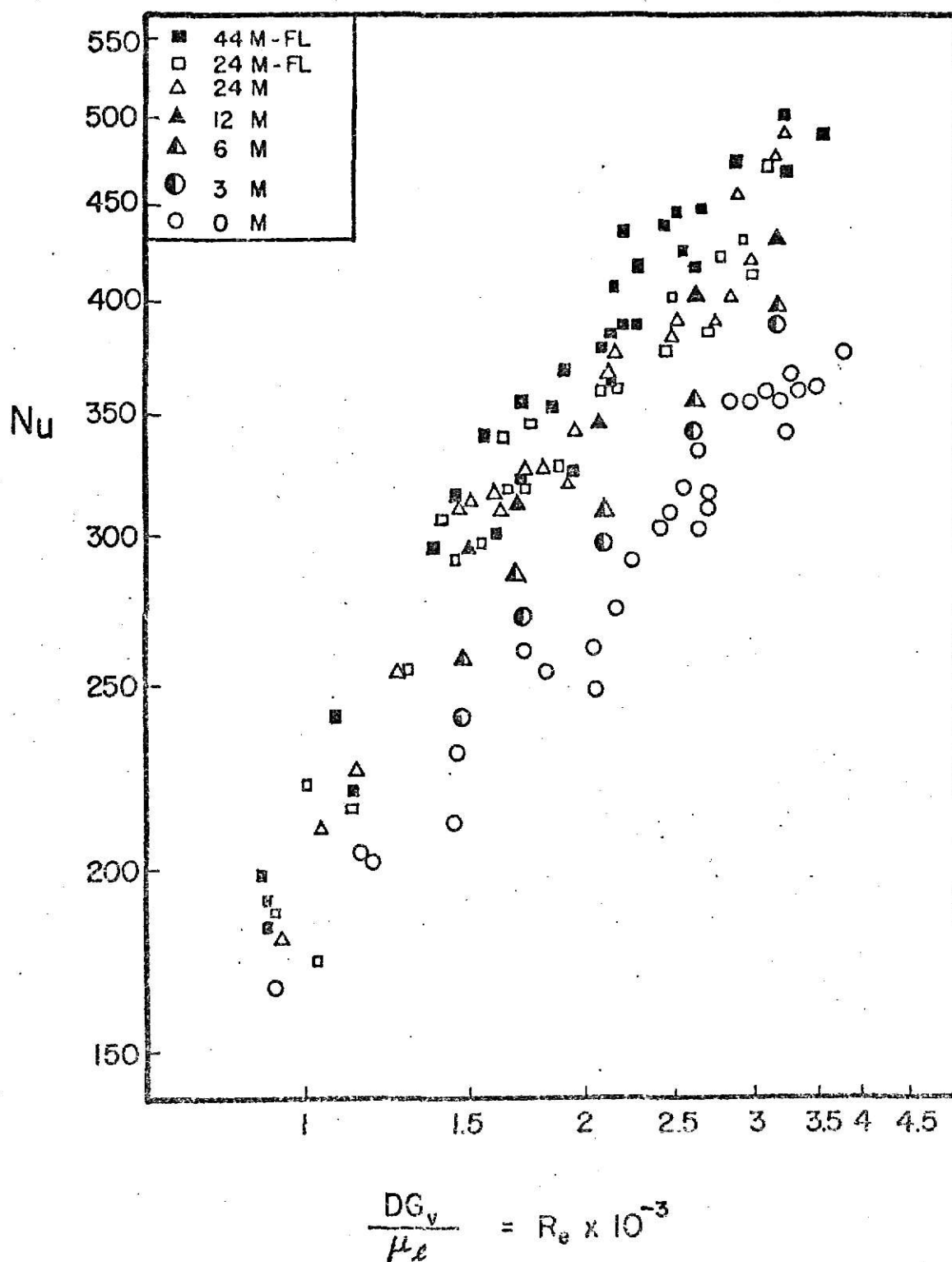


Fig. 17. Nusselt Number vs. Reynolds Number.

- 44M-FL (44 elements, see Fig. 13)
- 24M-FL (24 elements, see Fig. 13)
- △ 24M (24 elements, see Fig. 13)
- ▲ 12M (12 elements, see Fig. 13)
- ▲ 6M (6 elements, see Fig. 13)
- ⊙ 3M (3 elements, see Fig. 13)
- 0M (smooth tube)

where M stands for mixers and FL stands for full length.

In general, in all the different mixer arrangements as well as the smooth pipe results, the Nusselt number increase with the increase of the Reynolds number. Also, at the same Reynolds number, the Nusselt number increased as the number of mixer elements was increased. The lowest Nusselt number occurred in smooth tube tests.

Figure 18 shows a plot of R_1 defined in Eqn. (12) versus Reynolds number. This figure was reproduced from the results of Fig. 17. The following observations can be made from the figure.

1. The ratio (h_{im}/h_{is}) increases with Re . At a Re value in the neighborhood of 2000, the ratio starts decreasing. This was the case for almost all mixers arrangements. The reason for such a behavior could be attributed to the fact that for $Re > 2000$, the Nusselt number for the smooth tube increases at a faster rate, with increase of Re , than its rate of increase in the cases with static mixers.

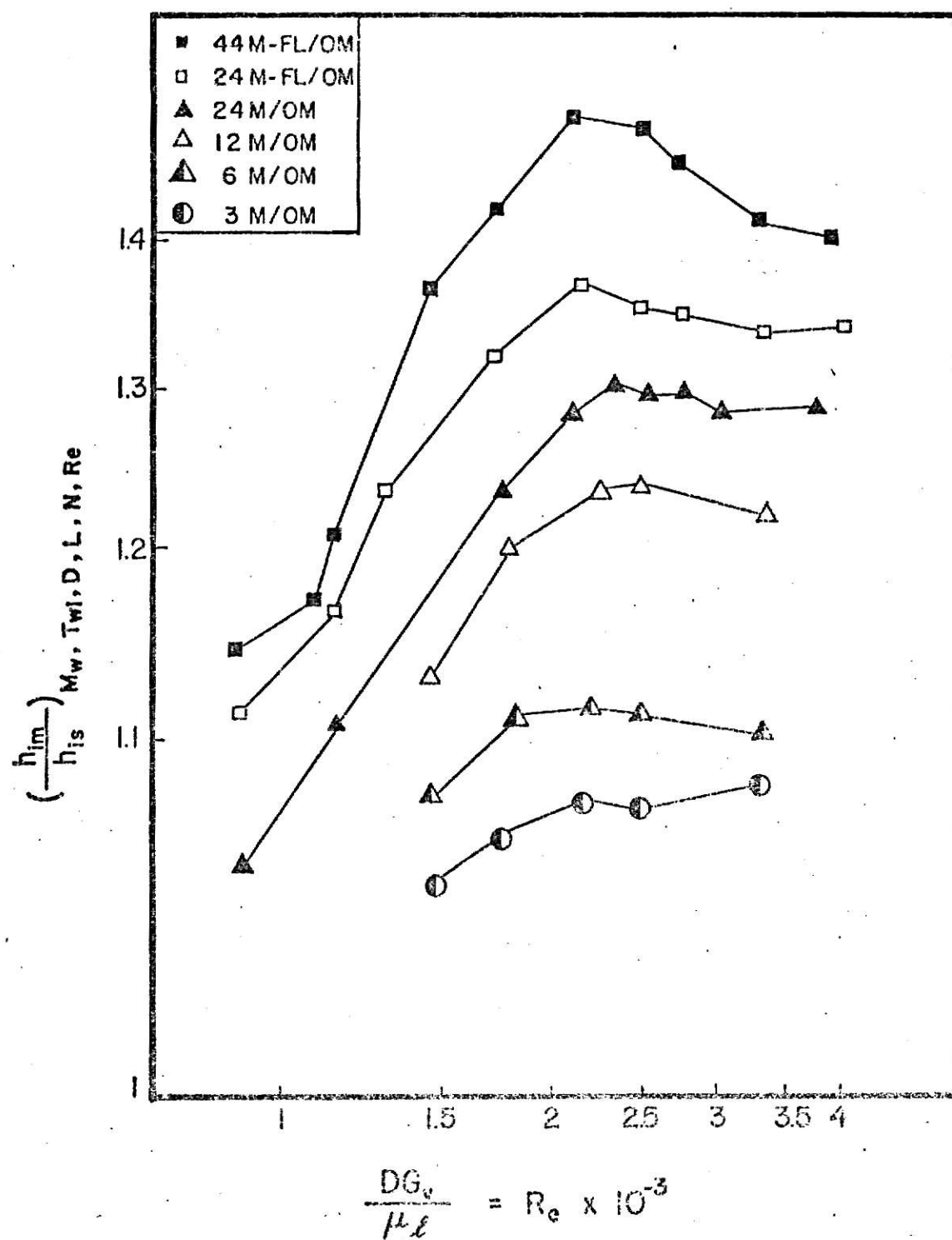


Fig. 18. The Change in Heat Transfer Coefficient Ratio with Reynolds Number.

2. The maximum improvement in heat transfer occurred by the use of largest number of mixers that occupied the entire length of the condenser tube.
3. In experiments where 24 mixer elements were used, the better improvement in heat transfer was achieved when the mixers were spaced to cover the entire length.

It was pointed out earlier that for the same heat duty, a certain percentage increase in the condensation coefficient results in a specified decrease in heat transfer area of the heat exchanger, as shown in Fig. 15. According to the results in Fig. 17, it is clear that the greatest reduction in surface area can be achieved by the use of 44M-FL mixers. However, other factors must be considered before such a choice is made. Among such factors are the pressure drop and the pumping power requirements.

In the second performance criterion of Bergles et al. [50], the ratio of heat transfer coefficient with promoter, h_m , to its value without promoter, h_s , was compared, at different Reynolds numbers under the following constraints: fixed geometry, fixed pressure drop ΔP , fixed ΔT , and fixed inlet temperature T_{in} . Such a criterion can be expressed as follows:

$$q_m/q_s = (h_m/h_s)_{\Delta P, \Delta T, L, D, N, T_{in}} \quad (13)$$

In Eqn. (13), $\Delta P, \Delta T, T_{in}$ and refer to their corresponding values on the heat transfer surface with promoter.

Before adapting the evaluation criterion of Eqn. (13) for use in the present investigation, it is essential to examine the pressure drop results. Figure 19 shows a plot of the pressure drop ΔP versus Reynolds number of all experimental runs of the test condensers with and without static mixers. For condenser runs without mixers, there was a slight increase in pressure between the inlet and outlet over the entire range of Reynolds number covered. This can be explained in the following:

Inside a horizontal condenser, the static pressure gradient can be written as

$$\left(\frac{dP}{dZ}\right) = \left(\frac{dP}{dZ}\right)_f + \left(\frac{dP}{dZ}\right)_{mo} \quad (14)$$

where

$$\left(\frac{dP}{dZ}\right)_f = \text{two-phase friction pressure gradient}$$

$$\left(\frac{dP}{dZ}\right)_{mo} = \text{pressure gradient due to momentum changes}$$

$\left(\frac{dP}{dZ}\right)_f$ is always negative, while $\left(\frac{dP}{dZ}\right)_{mo}$ could be positive or negative, and the final pressure change ΔP , between the inlet and outlet, depends on the relative contribution of the two pressure gradients. Soliman [45] developed two expressions for calculating $\left(\frac{dP}{dZ}\right)_f$ and $\left(\frac{dP}{dZ}\right)_{mo}$ during condensation. Interested readers are referred to reference [45] for additional details. Therefore, the increase in pressure the inlet and outlet could be attributed to the fact that the pressure change due to momentum change was positive compared to the pressure change due to

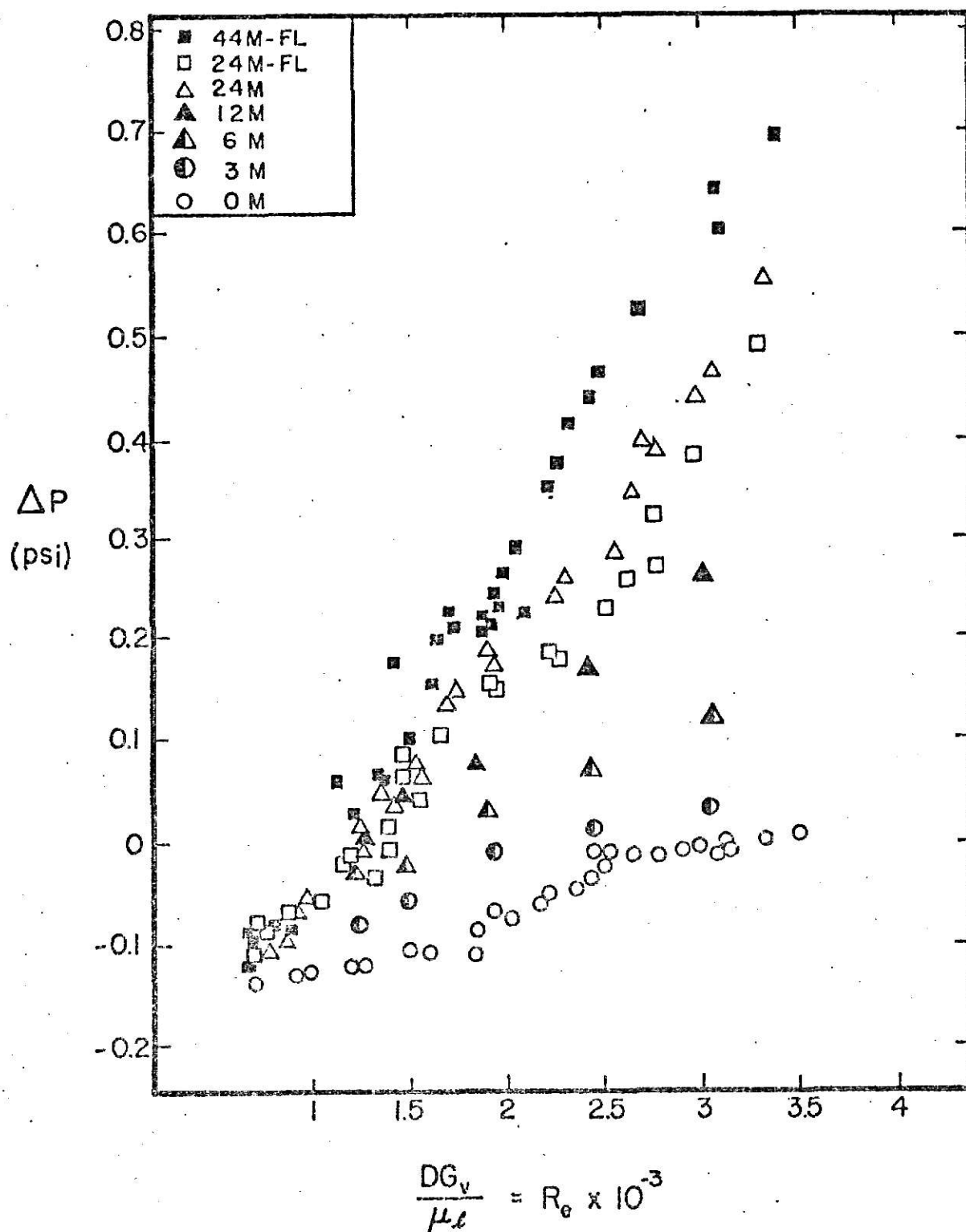


Fig.19. Pressure Drop vs. Reynolds Number.

two-phase friction which was negative. The same reasoning applies to condensation inside horizontal tubes with static mixers. Figure 19 shows that in a few cases with static mixers pressure rise occurred during condensation. In general, as the number of static mixers inside the condenser was increased, the Reynolds number range, over which there was a pressure increase, decreased. Also, as Reynolds number increased the pressure drop increased for all the mixers arrangements. At the same Reynolds number, the pressure drop increased, with the increase of the number of mixers. It is also important to note that at the same Reynolds number, among the condenser tests, which included 24 mixers, the pressure drop was higher, in the case where the mixer elements were arranged to occupy the upstream half of the condenser (24M), than in the case where the mixer elements were arranged to cover the entire length of the condenser (24M-FL).

Figure 20 shows a plot of the average condensation film coefficient h_i versus the pressure drop for all experimental runs. As one would expect, h_i increased with the increase in ΔP . An increase in ΔP signifies an increase in the mass flow rate, and consequently an increase in Re . The direct use of the criterion of Eqn. (13) to evaluate the results of the present investigation was not possible for two reasons. First, ΔT and T_{in} were not the same in experimental runs with and without mixers. Second, the ratio $(\frac{h_m}{h_s})$ at constant ΔP could not be evaluated because ΔP was negative for all smooth tube runs and positive for tubes with mixers. As a result, similar criterion was adopted in which

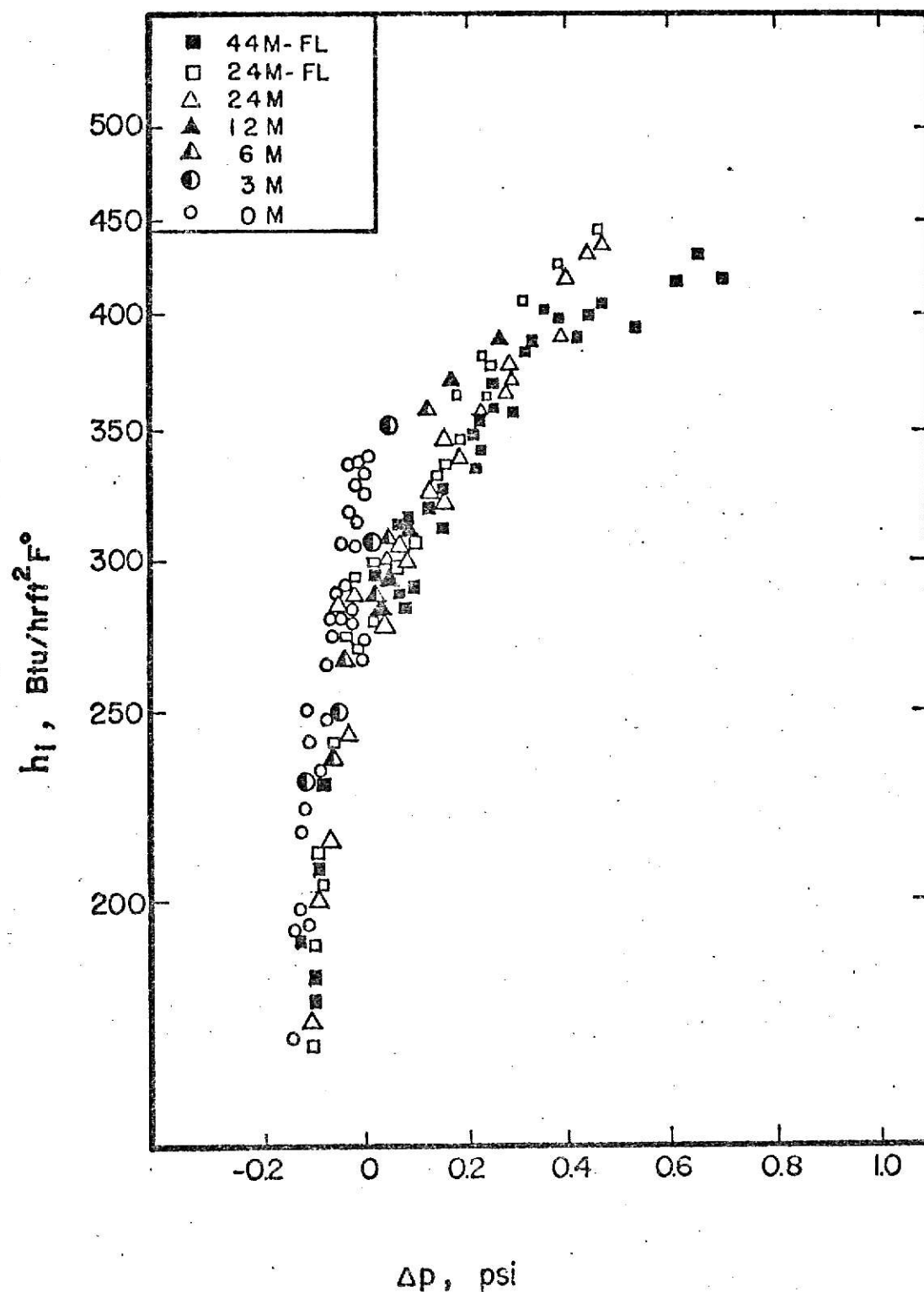


Fig. 20. Condensation Heat Transfer Coefficient vs. Pressure drop.

$\left(\frac{h_{im}}{h_{i44}}\right)$ was evaluated under the constraints: fixed geometry, ΔP , T_{wi} and coolant flow rate M_w . Let

$$R_2 = \left(\frac{h_{im}}{h_{i44}}\right)_{\Delta P, L, D, N, T_{wi}, M_w} \quad (15)$$

Figure 21 shows a plot of the ratio R_2 versus ΔP . Only data points with positive ΔP were considered in the plot. It was developed from the results in Fig. 20. Had the smooth pipe results showed positive ΔP , the denominator in the ratio (h_{im}/h_{i44}) could have been replaced by h_s for smooth pipes. The results in Fig. 21 show that for a pressure drop $\Delta P < 0.1$, depending on the value of ΔP , the ratio (h_{im}/h_{i44}) is less than or greater than 1.0 depending on the mixer arrangements. A ratio of $R_2 < 1.0$ indicates that for the same pressure drop the best improvement in heat transfer was obtained with 44 mixers. A ratio of $R_2 > 1.0$ indicates that the improvement in heat transfer, for the same pressure drop, can be obtained with a lower number of mixers. At pressure drop $\Delta P > 0.1$, the heat transfer ratio $R_2(h_{im}/h_{i44})$ increased with the reduction of the number of mixer elements, for the same pressure drop. It is also significant to point out that for the same pressure drop, a better heat transfer was achieved by the use of 24 mixers spaced to cover the entire length of the condenser than the use of 24 mixers in the upstream half of the condenser.

When the cost of fluid pumping is a major consideration, Bergles et al. [50] suggested the third criterion for the evaluation of performance of enhanced heat transfer surfaces for single phase flow. In this

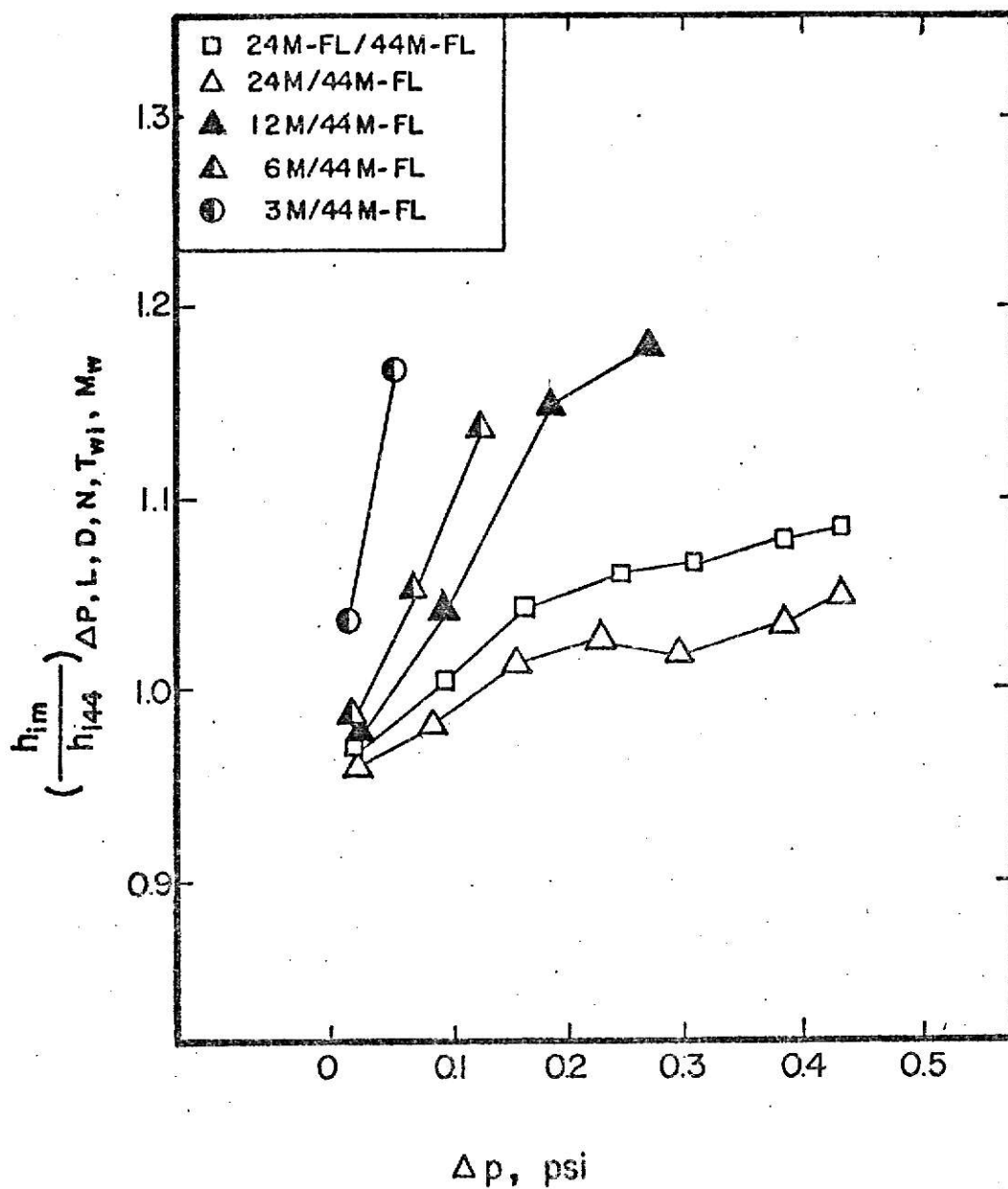


Fig. 21. The Change in Heat Transfer Coefficient Ratio with Pressure Drop.

criterion (h_m/h_s) was compared at different Reynolds numbers under the following constraints: fixed geometry, fixed pumping power, fixed ΔT , and fixed T_{in} . Such a criterion can be expressed as follows:

$$(q_m/q_s) = (h_m/h_s) P, \Delta T, L, N, D, T_{in} \quad (16)$$

The pumping power P can be evaluated from the product of the pump volume flow rate by the pressure drop ΔP . Due to the fact that the increase in pressure (negative ΔP) was encountered in experiments of smooth condensers as well as few other cases in condensers with static mixers, the above criterion was modified to render it applicable to the present results. First, the denominator in the ratio (h_m/h_s) was selected to be its corresponding value for the case of 44 mixers. Second, the evaluation criterion was applied to experimental runs with positive ΔP . Let,

$$R_3 = (h_{im}/h_{i44}) P, L, N, D, T_{wi}, M_w \quad (17)$$

Figure 22 shows a plot of the inside film coefficient h_{im} for all experimental runs with positive ΔP , versus the pumping power P . For all mixers arrangements, as h_{im} increased also the pumping power increased. To see the results in Fig. 22 in a better perspective, Fig. 23 was generated. Figure 23 shows a plot of $R_3 (h_{im}/h_{i44})$ versus the pumping power. The results in the figure indicate that, for the same pumping power, the highest h_{im} occurred with the least number of mixers. On that

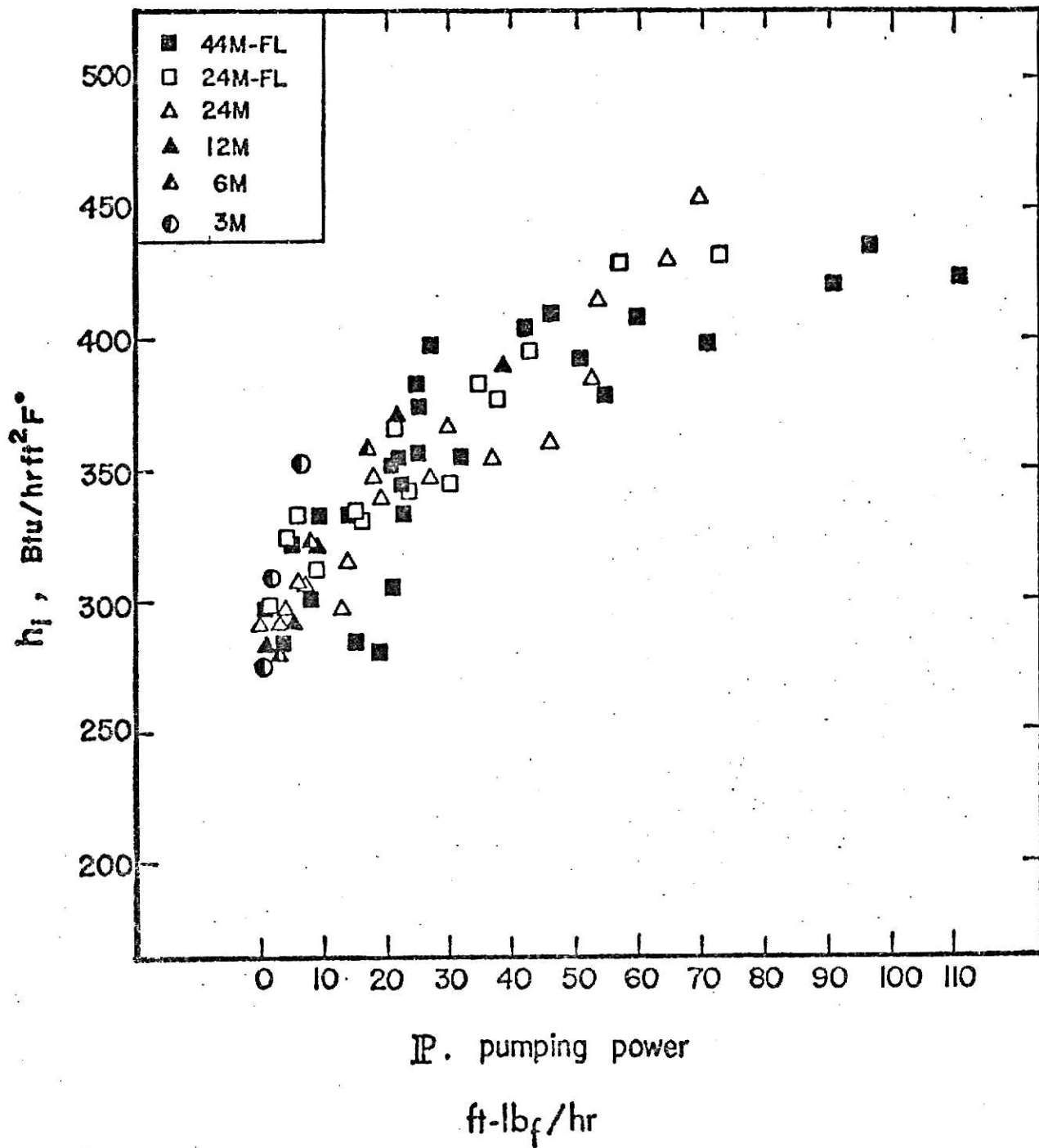


Fig. 22. Condensation Heat Transfer Coefficient vs. Pumping Power

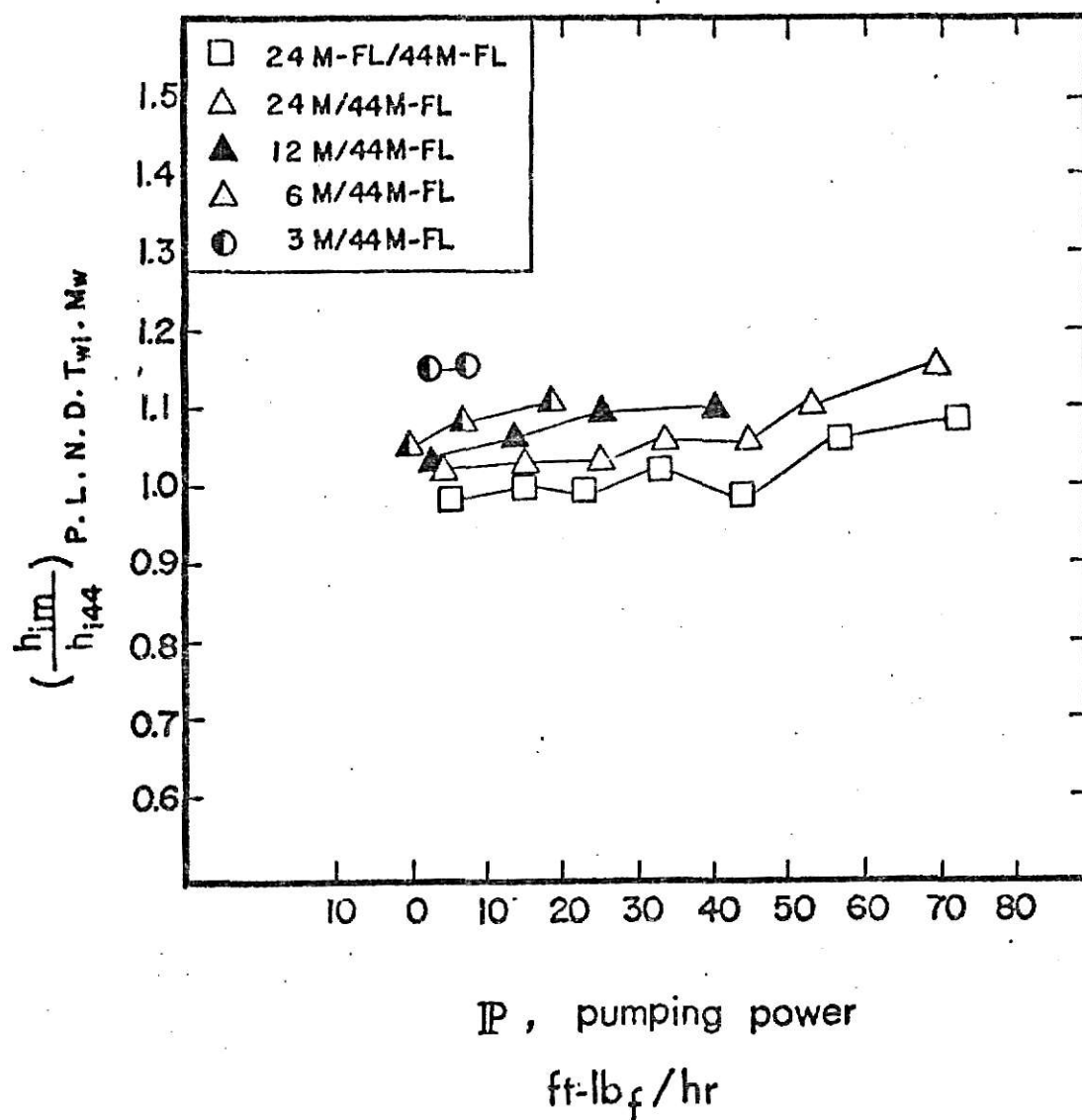


Fig. 23. The Change in Heat Transfer Coefficient Ratio with Pumping Power

basis one can extrapolate the results to indicate that if the pumping power is the major factor in one's consideration, the condenser tubes without mixers are preferable to tubes with mixers. Also, the results in Fig. 23 indicate that the use of 24 mixer elements placed to cover the upstream half of the condenser is preferable to the use of 24 mixer elements spaced to cover the entire length of the tube. In the latter arrangement the ratio (h_{im}/h_{i44}) is nearly unity upto pumping power of 45 ft-lb/hr, after which the ratio shows a trend for increase.

Comparison of Exit Flow Patterns in Tubes with and without Static Mixers

The attempt was made in this section to determine the effect of the use of static mixers on the flow pattern at the exit from the condenser. Among the different mixer arrangements, several runs, having nearly the same inlet conditions, were chosen and tabulated in Table IV. Three groups of data, each containing three runs, were examined. It can be seen from Table IV that the rate of heat gained by water increased from 4753 to 5074 BTU/hr for the first group, from 5439 to 5678 BTU/hr for the second group, and from 5770 to 6716 BTU/hr for the third group, with the addition of the mixers. The increase in heat transfer caused the flow pattern to change from wavy to slug for the first group, semi-annular to wavy for the second group, and from annular-wavy to semi-annular for the third group. The general trend is that the use of static mixers will move the flow patterns towards a single-phase liquid flow pattern, due to the increase in heat transfer rate.

Table IV. Comparison of Exit Flow Patterns with and without Mixers

Group	Run No	No. of Mixers	$G \times 10^{-4}$ lbm/hr ft ²	P psia	Inlet Superheat	Rate of Heat Gained by Water BTU/HR	Flow Pattern Observed	Nusselt No.	ΔP psi
(1)	90	0	5.11	32.53	15.39	4753.18	Wavy	303.1	-0.058
	49	24	5.25	29.1	16.1	4981.6	Slug	376.1	0.187
	23	44	5.14	28.09	24.91	5074.53	Slug	384.1	0.29
(2)	97	0	5.9	34.46	13.26	5439.9	Semi-Annular	356.1	-0.014
	73	24	5.9	33.3	16.7	5605.87	Wavy	400.6	0.345
	28	44	5.9	29.47	21.15	5678.42	Wavy	445.9	0.463
(3)	104	0	6.5	37.50	6.00	5770.36	Annular-Wavy	368.4	0.006
	76	24	6.6	32.9	17.57	6263.07	Semi-Annular	476.5	0.44
	32	44	6.7	37.95	4.46	6716.92	Semi-Annular	472.0	0.7

CHAPTER V

SUMMARY, CONCLUSIONS, AND RECOMMENDATIONS

Summary and Conclusions

Measurements of heat transfer and pressure drop were made during condensation inside a 0.5 in. I.D., 26 in. long horizontal tube. Seven sets of data including 120 points were taken. In six of these sets, data were taken while static mixers of different arrangements were placed inside the condenser tube. In the remaining data set, no mixers were used. The following mixers arrangements were tested:

1. Forty four mixer elements covering the entire length of the condenser.
2. Twenty four mixer elements, where the elements were spaced to cover the entire length of the condenser.
3. Twenty four mixer elements, where the elements were arranged to occupy the upstream half of the condenser tube.
4. Twelve mixer elements.
5. Six mixer elements.
6. Three mixer elements.

As in the third arrangement, the mixers in arrangements 4, 5, and 6 were placed to occupy the upstream section from the inlet of the condenser.

The range of flow conditions covered in this investigation were:

1. Saturation temperature, 130 - 174°F
2. Condensation pressure, 17 - 38 psia
3. Inlet superheat, 2 - 52.6°F
4. Coolant flow rate, 483.5 - 942.8 lbm/hr
5. Refrigerant mass velocity, 24000 - 79100 lbm/hrft²

6. Average rate of heat removal, 2510-6881 BTU/hr.

The results can be summarized in the following:

1. The average condensation heat transfer coefficient increased with the increase of Reynolds number in all mixers arrangements as well as in all smooth tube results. The Reynolds number was based on the I.D. of the tube and the mass velocity of the vapor, and it was in range $890 < Re < 3740$.
2. The ratio of condensation heat transfer coefficient with mixers to its value for the smooth tube (h_{im}/h_{is}) was compared for the different mixers arrangements, at the same Reynolds number. The values of the ratio, in all mixers arrangements, were greater than unity, and they increased to a maximum value at $Re \approx 2000$ and then decreased, over the Reynolds number range tested. Beyond such a critical value of the Reynolds number it appears that the average heat transfer coefficients increased at a faster rate in the smooth tube than (those) in the tubes with static mixers.
3. At the same Reynolds number, the largest increase in heat transfer occurred in the condenser with the largest number of mixers.
4. In all pressure drop results of the smooth tube, there was a slight pressure increase between the inlet and outlet, over the entire Reynolds number range tested. At the same Reynolds number, the highest pressure drop occurred in the condenser with the largest number of mixers. Also at the same Reynolds number, the pressure drop was higher in the case of 24 mixers arranged to occupy the upstream half of the condenser tube (24M) than in the case of 24 mixers spaced to cover the entire length of the condenser (24M-FL).

5. For the same pressure drop, higher heat transfer coefficients were achieved in the condenser with 24 mixers spaced to cover the entire condenser tube than in the condenser with 24 mixers placed in the upstream section of the condenser.
6. In general, the average heat transfer coefficient increased with the increase in pumping power in all condensers with static mixers.
7. For the same pumping power, the condenser with 24 mixers placed to cover the upstream half of the condenser gave a higher heat transfer rate compared to the condenser with 24 mixers spaced to occupy the entire length.
8. The static mixers inside the condenser had an effect on the development of the flow patterns. This effect was demonstrated qualitatively.

Recommendation for Further Studies

The attempt was made in this investigation to determine the effect of static mixers on the heat transfer and pressure drop characteristics during condensation of refrigerant-113 inside a 0.5 in. I.D., 26 in. long horizontal tube. Data were taken with different numbers and arrangements of static mixers over the Reynolds number range between 890 and 3740. The following recommendations are made for further studies:

1. Additional similar studies using different fluids, tube lengths, tube diameters, and types and arrangements of static mixers are needed. The objective of these studies is the development of design correlations for predicting the heat transfer and pressure drop for condensation inside horizontal tubes with static mixers.

2. Boiling and condensation are of the same importance from an engineering point of view. It is recommended that the static mixers be used in the boiling process, to investigate their effect on the heat transfer and pressure drop characteristics.

3. It's also recommended that effects of simultaneous applications of different augmentation techniques, such as surface vibration, fluid vibration, surface roughness, spiral fins, coiled tubes, twisted tapes, and static mixers, on the heat transfer and pressure drop characteristics of the condensation and boiling processes be investigated.

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APPENDICES

APPENDIX A

NOMENCLATURE

Symbol	Definition
A	tube cross-sectional flow area, ft^2
A_e	equivalent cross-sectional area for cooling water, ft^2
A_i	inside surface area of the condenser inner tube, ft^2
A_o	outside surface area of the condenser inner tube, ft^2
C_p	specific heat of cooling water, $\text{Btu/lbm } ^\circ\text{F}$
C_{pb}	specific heat of cooling water evaluated at bulk conditions, $\text{Btu/lbm } ^\circ\text{F}$
d	thickness of striations, mm or 10^{-6}m
D	inside diameter of condenser's inner tube, ft
D_1	outside diameter of smaller concentric tube (inside of annulus), ft
D_2	inside diameter of larger concentric tube (outside of annulus), ft
D_e	equivalent diameter of heat exchanger, ft
G_v	total mass velocity of refrigerant-113, based on the condenser tube inside area, lbm/hr ft^2
G_w	cooling water mass velocity, based on the equivalent cross-sectional area, lbm/hr.ft^2
h_{fg}	latent heat of vaporization, Btu/hr
h_i	inside heat transfer coefficient of condenser, $\text{Btu/hr.ft}^2 ^\circ\text{F}$
h_o	water-side heat transfer coefficient of condenser, $\text{Btu/hr.ft}^2 ^\circ\text{F}$
K	thermal conductivity of condenser tube, $\text{Btu/hr.ft } ^\circ\text{F}$
K_f	thermal conductivity of refrigerant-113, $\text{Btu/hr.ft } ^\circ\text{F}$
k	thermal conductivity of cooling water, $\text{Btu/hr.ft } ^\circ\text{F}$
L	length of condenser section, ft

Symbol	Definition
M_f	total mass flow rate of refrigerant-113, lbm/hr
M_l	liquid mass flow rate, lbm/hr
M_v	vapor mass flow rate, lbm/hr
M_w	cooling water mass flow rate, lbm/hr
Nu	Nusselt number = $\frac{h_i D}{k_f}$, dimensionless
P	static pressure, lbf/ft ²
P_{av}	average static pressure, lbf/ft ²
Pr	Prandtl number = $\frac{C_p \mu_l}{K_l}$
ΔP	pressure drop, $P_i - P_o$, psi
IP	pumping power, ft-lbf/hr
q	heat flux, Btu/hr
Re	Reynolds number = $\frac{G D}{\mu_l}$, dimensionless
T	temperature, °F
T_s	refrigerant-113 saturation temperature base on average pressure, °F
ΔT	logarithmic mean temperature difference, °F
$\Delta T'$	temperature difference between saturated liquid and inside surface, °F
U_j	overall heat transfer coefficient based on inside surface area of the condenser inner tube, Btu/hr.ft ² °F
X	dryness fraction
ρ_f	refrigerant density in pump, lbm/ft ³
ρ_l	saturated liquid density, lbm/ft ³
ρ_v	vapor density, lbm/ft ³

Symbol	Definition
μ	cooling water viscosity, c.p. or lbm/ft-hr
μ_a	cooling water viscosity evaluated at wall conditions, c.p. or lbm/ft.hr
μ_b	cooling water viscosity evaluated at bulk conditions, c.p. or lbm/ft.hr
μ_ℓ	refrigerant viscosity based on condensate condition centi-poise or lbm/ft.hr

Subscripts	Definition
ℓ	liquid
f	two-phase friction
m	tube with mixers
mo	momentum
s	smooth tube
wi	water at inlet condition
wo	water at outlet condition
Δ	difference

APPENDIX B

OPERATING CONDITIONS AND HEAT BALANCE

FOR ALL THE EXPERIMENTAL RUNS

Explanation of Terminology Used in Computer Output

Term	Meaning
PAVE	average pressure, psia
TSAT	saturation temperature, °F
DEGSUP	degree superheat, °F
LOGDT	logarithmic-mean temperature difference, °F
TWIN	inlet water temperature, °F
TWOUT	outlet water temperature, °F
TOTMW	total water mass flow rate, lbm/hr
GV	total refrigerant mass velocity, lbm/hr ft ²
X	dryness fraction
Q BY WATER	heat rate gained by water, BTU/hr
Q BY FRON	heat rate lost by refrigerant, BTU/hr
ERROR	heat balance error, %
44M-FL	44 mixer elements
24M-FL	24 mixer elements spaced to cover the entire length of the condenser
24M	24 mixer elements placed in the upstream section of the condenser
12M	12 mixer elements
6M	6 mixer elements
3M	3 mixer elements
OM	smooth tube

TABLE B-1: OPERATING CONDITIONS

RUN NO.	NO. OF MIXERS	PAVE PSIA	TSAT DEG F	DEGSUP DEG F	LOGDT DEG F	TWIN DEG F	TWOUT DEG F
1	44M-FL	18.50	130.12	51.19	66.10	62.66	65.36
2	44M-FL	19.79	134.00	44.97	69.71	62.93	65.63
3	44M-FL	19.94	134.45	45.29	69.76	63.29	66.07
4	44M-FL	20.18	135.13	44.40	69.66	63.83	67.08
5	44M-FL	20.77	136.80	43.20	72.65	62.57	65.72
6	44M-FL	20.16	135.07	25.88	68.78	64.46	68.09
7	44M-FL	21.59	139.01	42.08	74.18	62.79	64.82
8	44M-FL	22.17	140.55	39.78	75.46	62.92	67.24
9	44M-FL	24.46	146.46	33.40	81.10	63.11	67.57
10	44M-FL	22.10	140.33	20.02	73.91	64.45	68.36
11	44M-FL	23.19	143.25	37.86	76.85	64.15	68.63
12	44M-FL	24.65	146.93	33.15	82.40	62.12	66.91
13	44M-FL	24.89	147.50	31.79	82.86	62.13	67.09
14	44M-FL	28.08	154.92	44.13	90.03	62.13	67.61
15	44M-FL	26.49	151.29	28.38	86.07	62.61	67.78
16	44M-FL	25.59	149.18	14.41	82.26	64.01	68.58
17	44M-FL	28.09	154.93	49.14	89.90	62.12	67.88
18	44M-FL	24.90	147.53	2.04	81.37	64.01	68.27
19	44M-FL	27.69	154.05	26.33	88.91	62.52	67.71
20	44M-FL	28.39	155.60	23.72	89.61	62.75	69.16
21	44M-FL	29.88	158.78	21.50	92.14	62.75	70.43
22	44M-FL	31.46	162.06	16.92	94.77	62.75	71.70
23	44M-FL	28.09	154.93	24.91	89.76	62.43	67.95
24	44M-FL	28.05	154.86	24.94	89.88	62.03	67.87
25	44M-FL	28.04	154.83	24.63	89.54	62.34	68.18
26	44M-FL	28.99	156.91	22.37	91.57	62.39	68.22
27	44M-FL	30.88	160.86	20.00	94.32	63.56	69.47
28	44M-FL	29.47	157.93	21.15	92.83	62.03	68.09
29	44M-FL	32.24	163.65	15.39	98.43	62.03	68.33
30	44M-FL	35.30	169.52	12.85	102.71	63.38	70.18

TABLE B-1: CONTINUED

PUN NO.	AC. OF MIXERS	PAVE PSIA	TSAT DEG F	DEGSUP DEG F	LOGCT DEG F	TWIN DEG F	TWOUT DEG F
31	44M-FL	35.28	169.49	12.46	102.63	63.38	70.26
32	44M-FL	37.95	174.37	4.46	108.22	62.52	69.69
33	24M-FL	19.24	132.37	40.58	67.31	63.69	66.42
34	24M-FL	21.74	139.42	34.69	73.58	64.46	67.19
35	24M-FL	19.22	132.33	44.19	66.45	64.41	67.33
36	24M-FL	20.74	136.71	35.89	70.38	64.82	67.82
37	24M-FL	21.02	137.47	40.85	70.55	65.00	68.80
38	24M-FL	22.33	140.99	41.08	75.08	64.10	67.69
39	24M-FL	22.82	142.30	38.30	76.70	63.56	67.60
40	24M-FL	21.71	139.34	27.07	72.80	64.64	68.41
41	24M-FL	22.61	141.73	34.40	75.28	64.28	68.58
42	24M-FL	23.68	144.52	32.28	78.01	64.37	68.63
43	24M-FL	23.68	144.51	33.74	78.76	63.47	68.00
44	24M-FL	24.17	145.73	22.23	78.66	64.91	69.21
45	24M-FL	25.65	149.31	23.89	82.54	64.46	69.04
46	24M-FL	27.13	152.76	21.29	86.13	64.10	69.12
47	24M-FL	29.61	158.23	27.32	91.23	64.14	69.79
48	24M-FL	26.61	151.57	12.60	84.62	64.46	69.39
49	24M-FL	29.10	157.13	16.10	89.45	65.00	70.31
50	24M-FL	32.08	163.34	12.73	96.59	63.83	69.62
51	24M-FL	32.26	163.69	16.24	94.99	65.67	71.66
52	24M-FL	34.06	167.22	13.89	100.22	63.83	70.10
53	24M-FL	33.04	165.25	7.24	98.39	63.69	69.97
54	24M-FL	32.01	163.19	15.91	94.80	65.13	71.53
55	24M-FL	33.97	167.04	0.93	99.43	64.23	70.92
56	24M	21.25	138.11	47.14	71.30	65.45	68.17
57	24M	21.24	138.09	50.64	71.14	65.45	68.44
58	24M	21.24	138.07	44.69	70.96	65.54	68.66
59	24M	21.22	138.02	47.72	71.79	64.49	67.94
60	24M	22.61	141.73	43.98	73.58	66.12	70.14

TABLE B-1: CCNTINUED

RUN NO.	NC. OF MIXERS	PAVE PSIA	TSAT DEG F	DEGSUP DEG F	LOGDT DEG F	TWIN DEG F	TWOUT DEG F
61	24M	22.20	140.64	41.31	74.34	64.23	68.31
62	24M	24.68	147.00	36.20	79.10	65.76	70.01
63	24M	23.67	144.50	40.06	77.79	64.57	68.82
64	24M	24.66	146.95	35.07	80.56	64.15	68.59
65	24M	25.77	149.58	34.36	81.36	65.94	70.46
66	24M	27.13	152.77	26.99	85.98	64.41	69.13
67	24M	26.63	151.60	29.78	83.15	66.06	70.78
68	24M	27.41	153.40	30.66	86.13	64.71	69.79
69	24M	27.71	154.09	30.67	85.58	65.94	71.04
70	24M	30.08	159.20	22.93	91.65	64.77	70.28
71	24M	30.07	159.19	22.62	92.40	63.96	69.56
72	24M	31.56	162.27	20.95	96.62	62.70	68.54
73	24M	33.03	165.22	16.71	97.78	64.41	70.40
74	24M	33.01	165.18	16.59	99.75	62.25	68.54
75	24M	32.00	163.18	20.06	94.71	65.31	71.57
76	24M	32.98	165.13	17.57	99.49	62.25	68.94
77	24M	32.97	165.10	16.12	97.86	63.83	70.59
78	24M	32.92	165.01	18.08	98.00	63.29	70.63
79	OM	20.47	135.94	44.05	71.66	62.93	65.63
80	OM	21.06	137.60	44.18	71.13	65.00	67.92
81	OM	21.07	137.61	44.10	71.14	65.00	67.92
82	OM	24.76	147.19	34.44	81.74	63.74	67.15
83	OM	23.40	143.80	36.78	78.18	63.37	67.34
84	OM	26.25	150.72	30.15	85.10	63.56	67.65
85	OM	27.25	153.05	29.65	87.48	63.42	67.69
86	OM	22.75	164.67	20.31	99.03	63.38	67.87
87	OM	21.23	161.60	18.09	95.51	63.83	68.31
88	OM	21.23	161.60	21.52	95.29	63.96	68.61
89	OM	31.04	161.19	23.38	94.11	64.68	69.45
90	OM	32.53	164.23	15.39	97.77	63.91	68.98

TABLE B-1: CONTINUED

RUN NO.	NO. OF MIXERS	PAVE PSIA	TSAT DEG F	DEGSUP DEG F	LOGDT DEG F	TWIN DEG F	TWCUT DEG F
91	OM	31.62	162.41	18.77	97.59	62.21	67.37
92	OM	32.52	164.22	15.39	97.68	63.51	69.14
93	OM	32.22	163.61	19.47	96.93	63.96	69.34
94	OM	34.81	168.63	13.89	102.13	63.78	69.16
95	OM	34.65	168.32	11.18	101.69	63.92	69.30
96	OM	33.20	165.55	6.63	99.31	63.65	68.77
97	OM	34.46	167.96	13.26	101.84	63.20	69.00
98	OM	35.01	168.99	10.59	103.04	63.00	68.85
99	OM	35.00	168.98	10.59	102.92	63.00	69.07
100	OM	35.00	168.98	10.59	102.93	63.00	69.04
101	OM	36.21	171.20	9.08	104.68	63.47	69.52
102	OM	36.19	171.18	0.84	104.33	63.92	69.74
103	OM	35.56	170.00	11.22	103.74	63.20	69.25
104	OM	37.50	173.56	6.00	106.53	63.92	70.08
105	OM	38.00	174.45	5.11	107.32	63.92	70.26
106	12M	22.69	141.96	41.77	76.50	63.42	67.46
107	12M	24.18	145.76	39.06	78.66	64.95	69.21
108	12M	27.16	152.84	31.33	86.80	63.56	68.49
109	12M	30.11	159.28	29.82	91.64	64.77	70.46
110	12M	34.07	167.23	16.74	99.69	64.32	70.68
111	6M	23.83	144.90	37.76	79.22	63.78	67.55
112	6M	24.81	147.32	33.73	81.48	63.74	67.91
113	6M	29.19	157.32	26.98	90.22	64.68	69.47
114	6M	32.17	163.51	21.15	96.11	64.64	70.10
115	6M	35.14	169.23	13.82	101.48	64.64	70.81
116	3M	24.34	146.17	35.07	79.19	65.13	68.80
117	3M	25.72	149.48	32.68	82.79	64.64	68.72
118	3M	28.71	156.29	27.00	89.48	64.46	69.12
119	3M	32.19	163.56	19.71	95.74	65.13	70.46
120	3M	34.18	167.44	16.97	99.50	64.91	70.90

TABLE B-2: HEAT BALANCE

RUN NO.	NC. OF MIXERS	TCTMW LBM/HR	GV LBM/HR.SQFT	X	Q BY WATER BTU/HR	Q BY FRON BTU/HR	EPGR %
1	44M-FL	936.93	0.243E 05	0.0116	2527.08	2591.01	-2.530
2	44M-FL	936.93	0.240E 05	0.0130	2526.24	2554.12	-1.104
3	44M-FL	931.11	0.240E 05	0.0130	2592.89	2560.68	1.243
4	44M-FL	931.11	0.285E 05	0.0093	3024.90	2917.78	3.541
5	44M-FL	936.93	0.294E 05	0.0294	2947.75	2956.49	-0.297
6	44M-FL	936.93	0.358E 05	0.0156	3401.02	3197.85	5.974
7	44M-FL	936.93	0.374E 05	0.0149	3776.20	3439.27	8.922
8	44M-FL	936.93	0.402E 05	0.0139	4046.29	3710.63	8.296
9	44M-FL	936.93	0.425E 05	0.0235	4176.16	3804.90	8.890
10	44M-FL	936.93	0.419E 05	0.0237	3657.44	3613.67	1.197
11	44M-FL	942.83	0.430E 05	0.0234	4222.28	3883.76	8.017
12	44M-FL	931.11	0.453E 05	0.0227	4457.29	4019.50	9.922
13	44M-FL	925.36	0.470E 05	0.0118	4595.59	4223.46	8.098
14	44M-FL	936.93	0.517E 05	0.0265	5132.85	4712.79	8.184
15	44M-FL	936.93	0.509E 05	0.0426	4836.34	4381.50	9.405
16	44M-FL	936.93	0.478E 05	0.0381	4283.69	3974.56	7.216
17	44M-FL	925.36	0.542E 05	0.0864	5332.84	4706.84	11.739
18	44M-FL	936.93	0.459E 05	0.0241	3988.00	3782.22	5.160
19	44M-FL	936.93	0.509E 05	0.0329	4862.89	4433.80	8.824
20	44M-FL	713.85	0.498E 05	0.0326	4574.76	4277.45	6.499
21	44M-FL	576.57	0.498E 05	0.0326	4429.63	4199.80	5.189
22	44M-FL	483.58	0.498E 05	0.0756	4330.12	3949.21	8.797
23	44M-FL	936.93	0.531E 05	0.0312	5074.53	4552.27	10.292
24	44M-FL	936.93	0.575E 05	0.0297	5468.88	4933.49	9.698
25	44M-FL	942.83	0.586E 05	0.0386	5501.11	4974.02	9.582
26	44M-FL	936.93	0.592E 05	0.0476	5466.43	4942.14	9.591
27	44M-FL	936.93	0.608E 05	0.0569	5542.07	4972.48	10.278
28	44M-FL	936.93	0.625E 05	0.0559	5678.42	5162.19	9.091
29	44M-FL	936.93	0.652E 05	0.0627	5908.00	5255.98	11.036
30	44M-FL	936.93	0.717E 05	0.0626	6367.12	5639.28	11.431

TABLE B-2: CONTINUED

RUN NO.	NO. OF MIXERS	TCTMW LBM/HR	GV LBM/HR.SQFT	X	Q BY WATER RTU/HR	Q BY FRON BTU/HR	ERROR %
31	44M-FL	936.93	0.717E 05	0.0648	6449.89	5710.46	11.464
32	44M-FL	936.93	0.763E 05	0.0922	6716.92	5951.93	11.389
33	24M-FL	936.93	0.240E 05	0.0130	2558.31	2476.46	3.200
34	24M-FL	936.93	0.263E 05	0.0110	2563.33	2688.28	-4.874
35	24M-FL	936.93	0.268E 05	0.0105	2731.28	2742.18	-0.399
36	24M-FL	936.93	0.291E 05	0.0089	2813.74	2914.57	-3.583
37	24M-FL	936.93	0.363E 05	0.0308	3567.05	3286.28	7.871
38	24M-FL	936.93	0.324E 05	0.0345	3361.49	3091.39	8.035
39	24M-FL	936.93	0.386E 05	0.0725	3783.14	3369.90	10.923
40	24M-FL	936.93	0.369E 05	0.0303	3533.12	3220.76	8.841
41	24M-FL	936.93	0.414E 05	0.0373	4031.71	3650.35	9.459
42	24M-FL	936.93	0.414E 05	0.0270	3938.71	3689.61	7.499
43	24M-FL	936.93	0.439E 05	0.0483	4244.69	3819.88	10.008
44	24M-FL	936.93	0.430E 05	0.0491	4027.88	3634.55	9.765
45	24M-FL	936.93	0.458E 05	0.0449	4291.45	3872.02	9.774
46	24M-FL	936.93	0.506E 05	0.0593	4701.83	4143.99	11.864
47	24M-FL	936.93	0.575E 05	0.0683	5285.65	4746.85	10.194
48	24M-FL	936.93	0.517E 05	0.0694	4616.07	4154.17	10.006
49	24M-FL	936.93	0.564E 05	0.0693	4981.60	4524.84	9.169
50	24M-FL	936.93	0.618E 05	0.0545	5423.71	4938.96	8.938
51	24M-FL	936.93	0.640E 05	0.0615	5611.09	5126.62	8.634
52	24M-FL	936.93	0.673E 05	0.0613	5874.68	5350.89	8.916
53	24M-FL	936.93	0.664E 05	0.0702	5875.66	5299.50	9.806
54	24M-FL	936.93	0.719E 05	0.0966	5948.24	5620.26	5.514
55	24M-FL	936.93	0.763E 05	0.0880	6264.05	5733.95	8.463
56	24M	936.93	0.240E 05	0.0130	2552.97	2533.85	0.749
57	24M	936.93	0.268E 05	0.0105	2801.81	2808.81	-0.250
58	24M	936.93	0.291E 05	0.0089	2926.87	2952.61	-0.880
59	24M	936.93	0.328E 05	0.0172	3229.58	3259.03	-0.912
60	24M	936.93	0.369E 05	0.0152	3771.06	3401.63	9.796

TABLE B-2: CONTINUED

RUN NO.	NO. OF MIXERS	TOTMW LBM/HR	GV LBM/HR.SQFT	X	Q BY WATER BTU/HR	Q BY FRON BTU/HR	ERROR %
61	24M	936.93	0.378E 05	0.0400	3821.85	3417.12	10.590
62	24M	936.93	0.408E 05	0.0376	3981.78	3578.17	10.136
63	24M	936.93	0.402E 05	0.0278	3987.73	3624.83	9.100
64	24M	936.93	0.425E 05	0.0366	4157.32	3728.38	10.318
65	24M	936.93	0.436E 05	0.0359	4231.60	3821.66	9.687
66	24M	936.93	0.458E 05	0.0347	4415.47	3954.65	10.437
67	24M	936.93	0.469E 05	0.0459	4418.89	3998.43	9.515
68	24M	936.93	0.503E 05	0.0433	4761.01	4282.42	10.052
69	24M	936.93	0.514E 05	0.0530	4774.52	4332.84	9.251
70	24M	936.93	0.570E 05	0.0491	5157.70	4651.68	9.811
71	24M	936.93	0.575E 05	0.0487	5246.55	4742.01	9.617
72	24M	936.93	0.630E 05	0.0814	5464.27	4986.55	8.743
73	24M	936.93	0.641E 05	0.0803	5605.87	4983.00	11.111
74	24M	936.93	0.669E 05	0.0617	5186.24	5312.50	9.747
75	24M	936.93	0.664E 05	0.0702	5863.81	5260.75	10.284
76	24M	936.93	0.713E 05	0.0630	6263.07	5629.43	10.117
77	24M	936.93	0.730E 05	0.0732	6334.30	5696.98	10.061
78	24M	936.93	0.791E 05	0.0760	6881.77	6133.41	10.874
79	0M	931.11	0.240E 05	0.0130	2510.54	2498.75	0.469
80	0M	936.93	0.302E 05	0.0185	2737.67	2861.49	-4.523
81	0M	936.93	0.303E 05	0.0203	2737.67	2882.04	-5.273
82	0M	936.93	0.354E 05	0.0156	3195.20	3310.39	-3.605
83	0M	936.93	0.368E 05	0.0288	3249.19	3288.80	-1.219
84	0M	931.11	0.419E 05	0.0472	3806.47	3556.50	6.567
85	0M	942.83	0.431E 05	0.0487	4018.56	3654.34	9.064
86	0M	942.83	0.462E 05	0.0403	4229.71	3852.81	8.911
87	0M	936.93	0.469E 05	0.0560	4200.89	3824.08	8.970
88	0M	936.93	0.492E 05	0.0333	4356.82	4106.50	5.746
89	0M	936.93	0.513E 05	0.0405	4468.62	4324.73	3.220
90	0M	936.93	0.539E 05	0.0512	4753.18	4310.98	9.303

TABLE B-2: CONTINUED

RUN NO.	NO. OF MIXERS	TOTMW LBM/HR	GV LBM/HR.SQFT	X	Q BY WATER BTU/HR	Q BY FRON BTU/HR	ERROR %
91	OM	931.11	0.553E 05	0.0702	4808.74	4345.84	9.626
92	OM	936.93	0.578E 05	0.0915	4897.59	4348.12	11.219
93	OM	936.93	0.596E 05	0.0834	5037.42	4614.28	8.400
94	OM	942.83	0.596E 05	0.0843	5070.24	4557.94	10.104
95	OM	931.11	0.596E 05	0.0933	5006.40	4483.86	10.437
96	OM	942.83	0.596E 05	0.0924	4832.50	4428.74	8.355
97	OM	936.93	0.635E 05	0.0704	5439.99	4847.55	10.890
98	OM	936.93	0.661E 05	0.0957	5477.32	4939.86	9.812
99	OM	936.93	0.685E 05	0.1032	5683.29	5075.40	10.696
100	OM	936.93	0.702E 05	0.1215	5654.43	5061.43	10.487
101	OM	936.93	0.711E 05	0.1344	5668.71	5034.77	11.183
102	OM	931.11	0.711E 05	0.1419	5422.06	4942.26	8.849
103	OM	942.83	0.719E 05	0.1213	5705.80	5192.85	8.990
104	OM	936.93	0.755E 05	0.1350	5770.36	5384.94	6.679
105	OM	936.93	0.790E 05	0.1400	5961.62	5579.39	6.412
106	12M	936.93	0.380E 05	0.0396	3783.77	3428.37	9.393
107	12M	936.93	0.419E 05	0.0369	3985.80	3701.40	7.135
108	12M	936.93	0.498E 05	0.0436	4621.26	4234.36	8.372
109	12M	936.93	0.608E 05	0.0671	5324.82	4999.71	6.106
110	12M	936.93	0.711E 05	0.0892	5954.58	5486.41	7.862
111	6M	936.93	0.369E 05	0.0405	3530.54	3279.54	7.110
112	6M	936.93	0.419E 05	0.0501	3908.05	3608.31	7.670
113	6M	936.93	0.498E 05	0.0546	4489.29	4125.26	8.109
114	6M	936.93	0.597E 05	0.0756	5116.75	4713.71	7.877
115	6M	936.93	0.713E 05	0.1070	5785.20	5311.69	8.185
116	3M	936.93	0.369E 05	0.0405	3440.87	3246.05	5.662
117	3M	936.93	0.419E 05	0.0501	3819.92	3595.23	5.882
118	3M	936.93	0.498E 05	0.0656	4365.01	4088.05	6.345
119	3M	936.93	0.597E 05	0.1028	4988.29	4583.50	8.115
120	3M	936.93	0.713E 05	0.1199	5616.28	5310.22	5.449

APPENDIX C

HEAT TRANSFER, PRESSURE DROP, AND FLOW PATTERN PARAMETERS

Explanation of Terminology Used in Computer Output

Term	Meaning
UI	overall heat transfer coefficient, $B/hr.ft^2. ^\circ F$
HO	water side heat transfer coefficient, $B/hr.ft^2. ^\circ F$
HI	inside heat transfer coefficient, $B/hr.ft^2. ^\circ F$
RE	Reynolds number
NU	Nusselt number
DELP	pressure drop, psia

Flow Patterns

S	slug
W	wavy
A-W	annular-wavy
S-A	semi-annular
SA-W	semiannular--wavy

TABLE C: HEAT TRANSFER, PRESSURE DROP, AND
FLOW PATTERN PARAMETERS

RUN NO.	NO. OF MIXERS	UI G/HR.FT2.DF	HO G/HR.FT2.DF	HI G/HR.FT2.DF	RE	NU	DELP PSIA	FLOW PATTERN
1	44M-FL	134.8	370.2	190.9	0.890E 03	199.3	-0.121	S
2	44M-FL	127.8	370.0	177.2	0.902E 03	186.2	-0.097	S
3	44M-FL	131.1	368.2	183.9	0.905E 03	193.4	-0.089	S
4	44M-FL	153.1	368.2	230.5	0.108E 04	242.7	-0.081	S
5	44M-FL	143.1	370.1	208.0	0.112E 04	219.6	-0.089	S
6	44M-FL	174.3	367.9	282.5	0.135E 04	297.3	0.061	S
7	44M-FL	179.5	370.4	294.9	0.145E 04	312.5	0.029	S
8	44M-FL	189.1	370.3	321.8	0.157E 04	341.9	0.065	S
9	44M-FL	181.6	370.2	300.7	0.172E 04	322.6	0.084	S
10	44M-FL	174.5	367.9	282.8	0.163E 04	300.3	0.179	S
11	44M-FL	193.7	372.3	334.2	0.171E 04	356.7	0.104	S
12	44M-FL	190.7	368.5	327.8	0.184E 04	352.1	0.155	S
13	44M-FL	195.5	368.5	343.6	0.191E 04	369.4	0.226	S
14	44M-FL	201.0	372.7	356.3	0.220E 04	387.9	0.234	S
15	44M-FL	198.1	370.3	348.9	0.212E 04	377.6	0.211	S
16	44M-FL	182.3	368.3	303.7	0.196E 04	327.4	0.215	W
17	44M-FL	205.2	369.7	385.2	0.230E 04	419.4	0.226	A-W
18	44M-FL	172.3	366.7	279.0	0.187E 04	299.9	0.203	S
19	44M-FL	192.8	370.3	332.9	0.215E 04	361.9	0.211	W
20	44M-FL	180.0	297.8	351.1	0.212E 04	382.7	0.220	W
21	44M-FL	169.5	251.2	271.3	0.216E 04	407.0	0.246	W
22	44M-FL	161.1	218.1	397.1	0.220E 04	437.9	0.266	S-A
23	44M-FL	199.3	370.3	352.7	0.225E 04	384.1	0.290	S
24	44M-FL	214.5	370.3	403.3	0.244E 04	439.1	0.353	S
25	44M-FL	216.6	372.1	408.9	0.249E 04	445.2	0.376	S
26	44M-FL	210.5	370.3	389.2	0.254E 04	425.2	0.416	S
27	44M-FL	207.2	370.5	377.9	0.267E 04	415.8	0.440	W
28	44M-FL	215.7	370.2	407.4	0.270E 04	445.9	0.463	W
29	44M-FL	211.6	370.3	393.1	0.291E 04	434.7	0.526	W
30	44M-FL	219.6	370.7	417.4	0.330E 04	466.3	0.605	S-A

TABLE C: CONTINUED

RUN NO.	NC. OF MIXERS	UI B/HR.FT2.DF	HC B/HR.FT2.DF	HI B/HR.FT2.DF	RE	NU	DELP PSIA	FLOW PATTERN
31	44M-FL	221.6	370.6	428.6	0.330E 04	478.8	0.645	S-A
32	44M-FL	218.8	370.2	418.8	0.361E 04	472.0	0.700	S-A
33	24M-FL	134.0	369.2	189.6	0.893E 03	198.7	-0.101	S
34	24M-FL	122.8	369.4	167.9	0.102E 04	178.0	-0.097	S
35	24M-FL	144.9	369.7	212.1	0.997E 03	222.2	-0.089	S
36	24M-FL	141.0	369.2	203.8	0.111E 04	215.1	-0.081	S
37	24M-FL	178.3	370.0	291.8	0.139E 04	308.4	-0.034	S
38	24M-FL	157.9	370.4	240.7	0.127E 04	255.9	-0.058	S
39	24M-FL	173.9	370.3	280.2	0.152E 04	298.6	-0.042	S
40	24M-FL	171.1	368.6	273.8	0.143E 04	290.2	-0.018	S
41	24M-FL	188.8	369.8	321.5	0.163E 04	342.2	-0.010	S
42	24M-FL	180.3	369.9	297.4	0.165E 04	318.1	0.013	S
43	24M-FL	190.0	370.0	324.7	0.175E C4	347.3	0.041	S
44	24M-FL	180.6	368.8	298.7	0.173E 04	320.1	0.069	S
45	24M-FL	183.3	369.5	305.9	0.188E 04	329.9	0.100	S
46	24M-FL	192.5	369.6	332.3	0.212E 04	360.4	0.148	S
47	24M-FL	204.3	371.0	367.9	0.249E 04	402.9	0.179	S
48	24M-FL	192.3	368.4	332.6	0.215E 04	360.0	0.155	S
49	24M-FL	196.4	369.5	344.1	0.243E 04	376.1	0.187	S
50	24M-FL	198.0	369.8	348.9	0.275E 04	385.5	0.234	W
51	24M-FL	208.3	370.3	381.7	0.286E 04	422.0	0.266	W
52	24M-FL	206.7	370.5	376.2	0.306E 04	418.6	0.274	W
53	24M-FL	210.6	369.4	390.3	0.299E 04	432.7	0.313	W
54	24M-FL	221.2	370.2	427.7	0.320E 04	472.4	0.384	S-A
55	24M-FL	222.1	368.9	432.5	0.347E 04	481.1	0.463	S-A
56	24M	126.3	370.8	174.2	0.925E 03	194.3	-0.105	S
57	24M	138.9	371.2	199.0	0.103E 04	210.5	-0.089	S
58	24M	145.4	370.5	213.0	0.112E 04	225.3	-0.073	S
59	24M	158.6	370.9	242.3	0.126E 04	256.3	-0.034	S
60	24M	180.7	371.0	298.0	0.145E 04	317.2	-0.010	S

TABLE C: CCNTINUED

RUN NO.	NO. OF MIXERS	UI B/HR.FT2.DF	HO B/HR.FT2.DF	HI B/HR.FT2.DF	RE	NU	DELP PSIA	FLOW PATTERN
61	24M	181.3	370.5	299.7	0.148E 04	318.4	0.006	S
62	24M	177.5	370.7	289.5	0.165E 04	310.9	0.037	S
63	24M	180.8	370.8	298.2	0.161E 04	318.9	0.053	S
64	24M	181.9	370.5	301.6	0.172E 04	323.9	0.077	S
65	24M	183.4	370.8	305.4	0.180E 04	329.5	0.069	S
66	24M	181.1	370.3	299.3	0.192E 04	324.7	0.140	S
67	24M	187.4	370.5	316.8	0.196E 04	343.0	0.148	S
68	24M	194.9	370.8	338.7	0.212E 04	367.8	0.187	S
69	24M	196.7	370.9	344.1	0.217E 04	374.2	0.175	S
70	24M	198.4	370.6	349.6	0.248E 04	383.5	0.242	S
71	24M	200.2	370.6	355.3	0.250E 04	389.7	0.258	W
72	24M	199.4	370.8	352.6	0.279E 04	388.9	0.286	W
73	24M	202.1	370.6	361.3	0.289E 04	400.6	0.345	W
74	24M	208.1	370.6	380.7	0.301E 04	422.1	0.384	W
75	24M	218.3	370.8	416.3	0.295E 04	459.9	0.392	S-A
76	24M	222.0	370.7	429.8	0.321E 04	476.5	0.440	S-A
77	24M	228.2	370.6	454.3	0.328E 04	503.5	0.463	S-A
78	24M	247.6	370.8	537.5	0.355E 04	595.7	0.558	S-A
79	OM	123.5	368.3	169.4	0.913E 03	178.6	-0.137	S
80	OM	135.7	370.4	192.7	0.116E 04	203.7	-0.129	S
81	OM	135.7	370.4	192.7	0.116E 04	203.7	-0.133	S
82	OM	137.8	370.4	197.1	0.144E 04	211.7	-0.121	S
83	OM	146.5	370.3	215.4	0.147E 04	230.1	-0.121	S
84	OM	157.7	368.5	241.0	0.174E 04	260.5	-0.105	S
85	OM	162.0	372.5	249.7	0.181E 04	271.0	-0.109	S
86	OM	150.6	372.8	223.6	0.207E 04	247.6	-0.093	W
87	OM	155.1	370.3	234.4	0.207E 04	258.2	-0.085	W
88	OM	161.2	370.7	248.5	0.217E 04	273.8	-0.065	W
89	OM	167.4	370.9	263.5	0.226E 04	290.1	-0.073	W
90	OM	171.4	370.3	273.8	0.241E 04	303.1	-0.058	W

4/10/2002

TABLE C: CONTINUED

RUN NO.	NO. OF MIXERS	UI B/HR.FT2.DF	HC B/HR.FT2.DF	HI B/HR.FT2.DF	RE	NU	DELP PSIA	FLOW PATTERN
91	OM	173.7	368.6	280.5	0.245E 04	309.5	-0.050	W
92	OM	176.8	370.3	287.8	0.259E 04	318.5	-0.046	S-A
93	OM	183.2	370.7	305.0	0.266E 04	337.2	-0.034	S-A
94	OM	175.0	372.5	282.2	0.273E 04	314.7	-0.022	S-A
95	OM	173.6	368.4	280.2	0.273E 04	312.4	-0.018	S-A
96	OM	171.6	371.2	273.8	0.269E 04	303.7	-0.010	S-A
97	OM	188.3	370.5	319.6	0.290E 04	356.1	-0.014	S-A
98	OM	187.4	370.3	317.1	0.304E 04	353.9	-0.010	S-A
99	OM	194.7	370.3	338.5	0.315E 04	377.8	-0.006	SA-W
100	OM	193.7	370.3	335.4	0.323E 04	374.4	-0.002	SA-W
101	OM	190.9	370.4	327.2	0.331E 04	366.6	-0.010	SA-W
102	OM	183.2	367.5	306.8	0.331E 04	343.8	-0.009	SA-W
103	OM	193.9	372.4	334.8	0.332E 04	374.3	-0.010	A-W
104	OM	191.0	370.3	327.4	0.355E 04	368.4	0.006	A-W
105	OM	195.9	370.3	342.0	0.374E 04	385.5	0.009	A-W
106	12M	174.4	370.7	281.3	0.150E 04	299.5	0.013	S
107	12M	178.7	370.9	292.5	0.169E 04	313.4	0.045	S
108	12M	187.7	370.8	317.6	0.209E 04	344.6	0.077	S
109	12M	204.9	371.5	369.6	0.265E 04	405.5	0.171	W
110	12M	210.6	370.8	389.1	0.324E 04	432.9	0.266	S-A
111	6M	157.1	370.6	239.0	0.148E 04	255.8	-0.065	S
112	6M	169.1	370.4	267.9	0.170E 04	287.9	-0.026	S
113	6M	175.4	370.8	284.0	0.214E 04	310.5	0.029	S
114	6M	187.7	370.9	317.5	0.266E 04	351.0	0.069	W
115	6M	201.0	370.7	357.7	0.328E 04	399.4	0.124	S-A
116	3M	153.2	370.4	230.1	0.149E 04	246.8	-0.081	S
117	3M	162.7	370.5	252.1	0.172E 04	271.9	-0.046	S
118	3M	172.0	370.7	275.1	0.213E 04	300.3	-0.010	K
119	3M	183.7	370.7	306.3	0.266E 04	338.6	0.017	S-A
120	3M	199.0	370.9	351.3	0.325E 04	391.0	0.045	SA-W

APPENDIX D

CALCULATIONS SAMPLE FOR RUN NUMBER 12

Recorded Data

The following readings were recorded for run 12:

Reference (ambient) temperature	= 83 °F
Atmosphere pressure	= 14.23 psi
Condenser inlet pressure	= 10.5 psig
Refrigerant-113 inlet temperature	= 2.35 m.v.
Refrigerant-113 exit temperature	= 1.48 m.v.
Condensate flow rate at exit	= 0.06 gpm
Liquid flowmeter temperature	= 0.69 m.v.
Pump flow rate	= 0.081 gpm
Water inlet temperature	= -0.47 m.v.
Water exit temperature	= -0.3635 m.v.
Water flow rate through the condenser	= 1.863 gpm
d/p cell output	= 9.1 in. Hg
Pump temperature	= 0.47 m.v.

Step 1: Conversion of all Measurements into Useful Data

Reference (ambient) temperature	= 83 °F
Refrigerant-113 inlet temperature	= 180.72° °F
Refrigerant-113 exit temperature	= 145.68 °F
Water inlet temperature	= 62.123 °F
Water exit temperature	= 66.910 °F

Liquid flowmeter temperature	= 112.78 °F
Pump temperature	= 103.4 °F
Condenser inlet pressure	= 24.73 psia
d/p cell output	= 0.155 psi

Step 2: Thermophysical Properties

The condenser exit pressure was the difference between the inlet pressure and the pressure drop from the d/p cell output, then

$$P_o = 24.73 - 0.155 = 24.575 \text{ psia}$$

$$P_{av} = \frac{24.73 + 24.575}{2} = 24.65 \text{ psia}$$

All properties were taken from reference [32] and [45], consequently

$$T_s = 146.93 \text{ °F}$$

$$K_l = 0.0388 \text{ B/hr ft °F}$$

$$\mu_l = 1.028 \text{ lbm/hr ft}$$

Step 3: Heat Balance Error

$$M_f = 0.081 \frac{\text{gallons}}{\text{minute}} \times 60 \frac{\text{minutes}}{\text{hour}} \times 0.13368 \frac{\text{ft}^3}{\text{gallon}}$$

$$\times 95.0 \frac{\text{lbm}}{\text{ft}^3} \quad \rightarrow 1.34$$

$$= 61.72 \text{ lbm/hr}$$

$$M_l = 0.08 \frac{\text{gallons}}{\text{minute}} \times 60 \frac{\text{minutes}}{\text{hour}} \times 0.13368 \frac{\text{ft}^3}{\text{gallon}}$$

$$\times 94.0 \frac{\text{lbm}}{\text{ft}^3}$$

$$= 60.316 \text{ lbm/hr}$$

$$M_v = 61.72 - 60.316 = 1.404 \text{ lbm/hr}$$

$$\text{Enthalpy of refrigerant at inlet} = 105.85 \text{ B/lbm}$$

$$\text{Enthalpy of condensate at outlet} = 39.338 \text{ B/lbm}$$

$$\text{Enthalpy of vapor at outlet} = 100.336 \text{ B/lbm}$$

$$\text{Enthalpy of refrigerant at outlet}$$

$$= \frac{(60.316 \times 39.338) + (1.404 \times 100.336)}{61.72}$$

$$= 40.725 \text{ B/lbm}$$

$$\text{Heat lost by refrigerant} = 61.72 (105.85 - 40.725)$$

$$= 4019.5 \text{ B/hr}$$

$$M_w = 1.86335 \frac{\text{gallons}}{\text{minute}} \times 60 \frac{\text{minutes}}{\text{hour}} \times 0.13368 \frac{\text{ft}^3}{\text{gallon}}$$

$$\times 62.3 \text{ lbm/ft}^3$$

$$= 931.1 \text{ lbm/hr}$$

$$\text{Heat gained by water} = 931.1 \times (66.91 - 62.123)$$

$$= 4457.17 \text{ B/hr}$$

$$\text{Percentage heat balance error} = \frac{4457.17 - 4019.5}{4457.17}$$

$$= 9.82\%$$

Step 4 Calculation of Inside Heat Transfer Coefficient, Nusselt Number, and Reynolds Number

$$q = U_i A_i \Delta T$$

$$q = 4457.17 \text{ B/hr}$$

$$A_i = \pi dL = \pi \left(\frac{1}{2 \times 12} \right) \left(\frac{26}{12} \right) = 0.283616 \text{ ft}^2$$

$$\begin{aligned} \Delta T &= \frac{(146.93 - 62.123) - (146.93 - 66.91)}{\ln[(146.93 - 62.123)/(146.93 - 66.91)]} \\ &= 82.39 \text{ } ^\circ\text{F} \end{aligned}$$

$$U_i = \frac{4457.17}{(0.283616)(82.39)} = 190.74 \text{ B/hr ft}^2 \text{ } ^\circ\text{F}$$

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{A_i \ln(\gamma_o/\gamma_i)}{2\pi KL} + \frac{A_i}{A_o} \frac{1}{h_o}}$$

$$\gamma_i = 2.08 \times 10^{-2} \text{ ft}$$

$$\gamma_o = 2.60 \times 10^{-2} \text{ ft}$$

$$A_o = \left(\frac{5}{8 \times 12} \right) \times \frac{26}{12} = 3.545 \times 10^{-1} \text{ ft}^2$$

$$K = 220 \text{ B/hr ft}^2 \text{ } ^\circ\text{F}$$

$$L = \frac{26}{12} = 2.1666 \text{ ft}$$

Calculation of h_o :

For water in annulus:

$$\frac{h_o}{C_{pb} G_w} \left(\frac{C_{\mu}}{k} \right)_b^{2/3} \left(\frac{\mu_a}{\mu_b} \right)^{0.14} = \frac{0.023}{(D_e G_w / \mu_b)^{0.2}}$$

$$\text{water bulk temperature} = 64.51^\circ \text{F}$$

$$\left(\frac{C_{\mu}}{k} \right)_b = 7.22$$

$$C_{pb} = 0.999 \text{ B/lbm } ^\circ \text{F}$$

$$G_w = \frac{M_w}{A_e}$$

$$M_w = 931.1 \text{ lbm/hr}$$

$$A_e = \pi \left[\left(\frac{1}{2 \times 12} \right)^2 - \left(\frac{5}{8 \times 12} \right)^2 \right] = 3.3236 \times 10^{-3} \text{ ft}^2$$

$$G_w = \frac{931.1}{3.3236 \times 10^{-3}} = 280148.0 \text{ lbm/hr ft}^2$$

$$\text{water viscosity at bulk temperature } \mu_b = 2.6 \text{ lbm/ft-hr}$$

$$\text{water viscosity at wall temperature } \mu_a = 1.4592 \text{ lbm/ft-hr}$$

$$D_e = D_2 - D_1 = \frac{1}{12} - \frac{5}{8 \times 12} = 3.125 \times 10^{-2} \text{ ft}$$

$$\frac{h_o}{(0.999)(280148.0)} (7.22)^{2/3} \left(\frac{1.4592}{2.6} \right)^{0.14} = \frac{0.023}{\left(\frac{3.125 \times 10^{-2} \times 280148.0}{2.6} \right)^{0.2}}$$

$$h_o = 368.2 \text{ Btu/hr ft}^2 ^\circ \text{F}$$

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{A_i \ln(\gamma_o/\gamma_i)}{2\pi KL} + \frac{A_i}{A_o} \frac{1}{h_o}}$$

$$190.74 = \frac{1}{\frac{1}{h_i} + \frac{(0.283616) \ln(2.60/2.08)}{2\pi(220)(2.1666)} + \frac{(0.283616)}{(0.3545)} \frac{1}{368.2}}$$

$$h_i = 327.8 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}$$

$$Nu = \frac{h_i D}{K_\ell}$$

$$K_\ell = 0.0388 \text{ B/hr ft } ^\circ\text{F}$$

$$Nu = \frac{327.8 \times (\frac{1}{2} \times \frac{1}{12})}{0.0388} = 352.0$$

$$Re = \frac{DG_v}{\mu_\ell}$$

$$G_v = \frac{M_f}{A} = \frac{61.72}{\pi(\frac{1}{2 \times 2 \times 12})^2} = 45264.5 \text{ lbm/hr ft}^2$$

$$D = \frac{1}{2 \times 12}$$

$$\mu_\ell = 1.028 \text{ lbm/hr ft}$$

$$Re = \frac{45264.5}{2 \times 12 \times 1.028} = 1834.6$$

APPENDIX E

PROPERTIES OF REFRIGERANT-113

Chemical Formula	$\text{C}_2\text{Cl}_2\text{F}_4$
Molecular Weight	187.39
Boiling Temperature at Atmospheric Pressure	117.6 °F
Freezing Temperature at Atmospheric Pressure	-31 °F
Critical Temperature	417.4 °F
Critical Pressure	498.9 psia
Critical Density	36.0 lbm/ft ³
Density of Liquid at 86 °F	96.96 lbm/ft ³
Specific Volume of Saturated Vapor at 5 °F	27.04 ft ³ /lbm
Specific Heat of Liquid at 86 °F	0.218 B/lbm °F
Thermal Conductivity	
Saturated Liquid at 5 °F	0.044 B/ft hr °F
Saturated Liquid at 86 °F	0.037 B/ft hr °F
Vapor at Saturation Pressure at 5 °F	0.0035 B/ft hr °F
Vapor at 1/2 Atmosphere Pressure at 86 °F	0.0045 B/ft hr °F
Viscosity	
Saturated Liquid at 5 °F	1.28 C.P.
Saturated Liquid at 86 °F	0.638 C.P.
Vapor at Saturation Pressure at 5 °F	0.0079 C.P.
Vapor at 1/2 Atmosphere Pressure at 86 °F	0.0096 C.P.
Color	clear and colorless
Flammability	Nonflammable

AUGMENTATION OF CONDENSATION HEAT TRANSFER
WITH IN-LINE STATIC MIXERS

by

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B.S., Taipei Institute of Technology, Taipei, R.O.C., 1970

AN ABSTRACT OF A MASTER'S THESIS

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ABSTRACT

Measurements of heat transfer and pressure drop were made during condensation inside a 0.5 in. I.D., 26 in. long horizontal tube. Seven sets of data points were taken. In six of these sets, data were taken while static mixers of different arrangements were placed inside the condenser's tube. In the remaining data set, no mixers were used. The six different arrangements of mixers were: forty four mixer elements covering the entire length of the condenser (44M-FL), twenty four mixer elements spaced to cover the entire length of the condenser (24M-FL), twenty four mixer elements arranged to occupy the upstream half of the condenser (24M), twelve mixer elements, six mixer elements, and three mixer elements. The last three mixer arrangements were also placed to occupy the upstream section from the inlet of the condenser. A wide range of condensing conditions was covered.

The average condensation heat transfer coefficient increased with the increase of Reynolds number in all mixers arrangements as well as the smooth tube results. Reynolds number was based on the inside diameter of the tube and the mass velocity of the condensing fluid, and it was in the range $890 < Re < 3740$. The ratio of condensation heat transfer coefficient with the mixers to its value for the smooth tube (h_{im}/h_{is}), in all mixers arrangements over the Reynolds number range tested, was greater than unity. Its value increased to a maximum value at $Re \approx 2000$, then it decreased. Beyond such critical value of Reynolds number, the average heat transfer coefficient increased at a faster rate in the smooth tube than its rate of increase in the tubes with static mixers. At the same

Reynolds number, the largest increase in both heat transfer and pressure drop occurred in the condenser with the largest number of mixers. There was a slight pressure increase between inlet and outlet in all pressure drop results of the smooth tube. Also at the same Reynolds number, the pressure drop was higher in the case of 24M mixers arranged to occupy the upstream half of the condenser tube than in the case of 24M-FL mixers spaced to cover the entire length of the condenser. But for the same pressure drop, higher heat transfer coefficients were achieved in the condenser with 24M-FL mixers as compared to the 24M mixers.

In general as the average heat transfer coefficient increased in all condensers with static mixers the pumping power increased. For the same pumping power the condenser with 24M mixers showed a higher heat transfer rate compared to the condenser with 24M-FL mixers. The effect of the use of static mixers inside the condenser on the development of the flow patterns was qualitatively examined.