BOILING FILM COEFFICIENTS FOR SULFUR DIOXIDE IN A VERTICAL EVAPORATOR

by

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I. INTRODUCTION

Within the past twenty years the demands for a commercial grade of sulfur dioxide have far exceeded the demands for the more pure refrigerant grade. In fact, the use of this chemical has increased to the point where it is now regarded as a basic raw material for many industries. Sulfur dioxide has been used as a reducing agent, a liquid extractant, a chemical manufacturing raw material, a neutralization and acidification agent, and as a preservative, to mention only a few of its applications.

Most of the heat transfer data available for boiling sulfur dioxide are connected with its use as a refrigerant. These data generally offer information at relatively low temperatures and low flows. Because of the many additional industrial uses of sulfur dioxide, much more information is needed to describe boiling heat transfer at higher temperatures and greater flows. One of the purposes of this investigation was to evaluate boiling film coefficients over a wider range of temperature differences and greater flows.

In order for these higher temperatures and greater flows to be obtained an industrial evaporator was modified

for laboratory use. The industrial function of such an evaporator is to vaporize liquid sulfur dioxide. Although it is possible to draw off only vapor from a supply drum, the flow is relatively low. An industrial user of SO_2 would need to connect a number of supply drums to realize any appreciable flow for his process. By employing an evaporator the user needs only one supply drum and can obtain approximately ten times the flow of SO_2 vapor than he could previously.

It is a purpose of this investigation to provide reliable information regarding film boiling coefficients for future designers of SO_2 evaporators. The information contained in this thesis is expected to help fill the gap in the partial nucleate boiling regime.

II. REVIEW OF LITERATURE

INTRODUCTION

This review of literature deals primarily with the areas of boiling heat transfer in which this experiment was conducted. Only the most pertinent conclusions drawn by many investigators are included as they concern this problem. All applicable investigations of SO_2 boiling heat transfer are reviewed in detail. The areas of emphasis will be stable nucleate, partial nucleate boiling, and stable film boiling.

GENERAL BOILING INVESTIGATIONS

Although there have been many investigations regarding the process of evaporation, there is still no general correlation nor adequate description of all these heat transfer data and processes. Rohsenow[1*] stated that the best approach has been to group the investigations "... according to the nature of fluid motion and according to the physical disposition of the heating surface." The different types of heating surfaces that have been investigated are:

1) Horizontally submerged flat plates
2) Horizontally submerged tubes

*Superscripts refer to the bibliography.

3) Vertically submerged tubes or plates

4) Inside surface of a tube with fluid flowing therein.

The various types of resulting fluid motion are:

1) Evaporation without boiling

2) Nucleate boiling*

3) Film boiling.

A complete boiling curve for sulfur dioxide is not yet available, that is, a logarithmic plot of temperature difference**versus heat flux. McAdams[2] states that such a curve is probably similar in shape to the one for water, shown as Figure 1. This boiling curve was found as a result of experiments by Farber and Schorah[3] of pool boiling with an electrically heated wire submerged horizontally in a tank of water at the saturation temperature. This characteristic curve has been copied and explained further by Rohsenow and Kreith.[4] In regimes I and II, according to Rohsenow,

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*Included as a form of nucleate boiling is surface, or local, boiling which occurs when a liquid that is subcooled comes into contact with a heater surface hot enough to cause boiling immediately. The vapor bubbles that are formed will condense in the cold liquid after they break away from the hot surface and no net generation of vapor is obtained with this degassed liquid.[2]

**Difference in temperature between the surface of the heater and the bulk temperature of the boiling liquid.

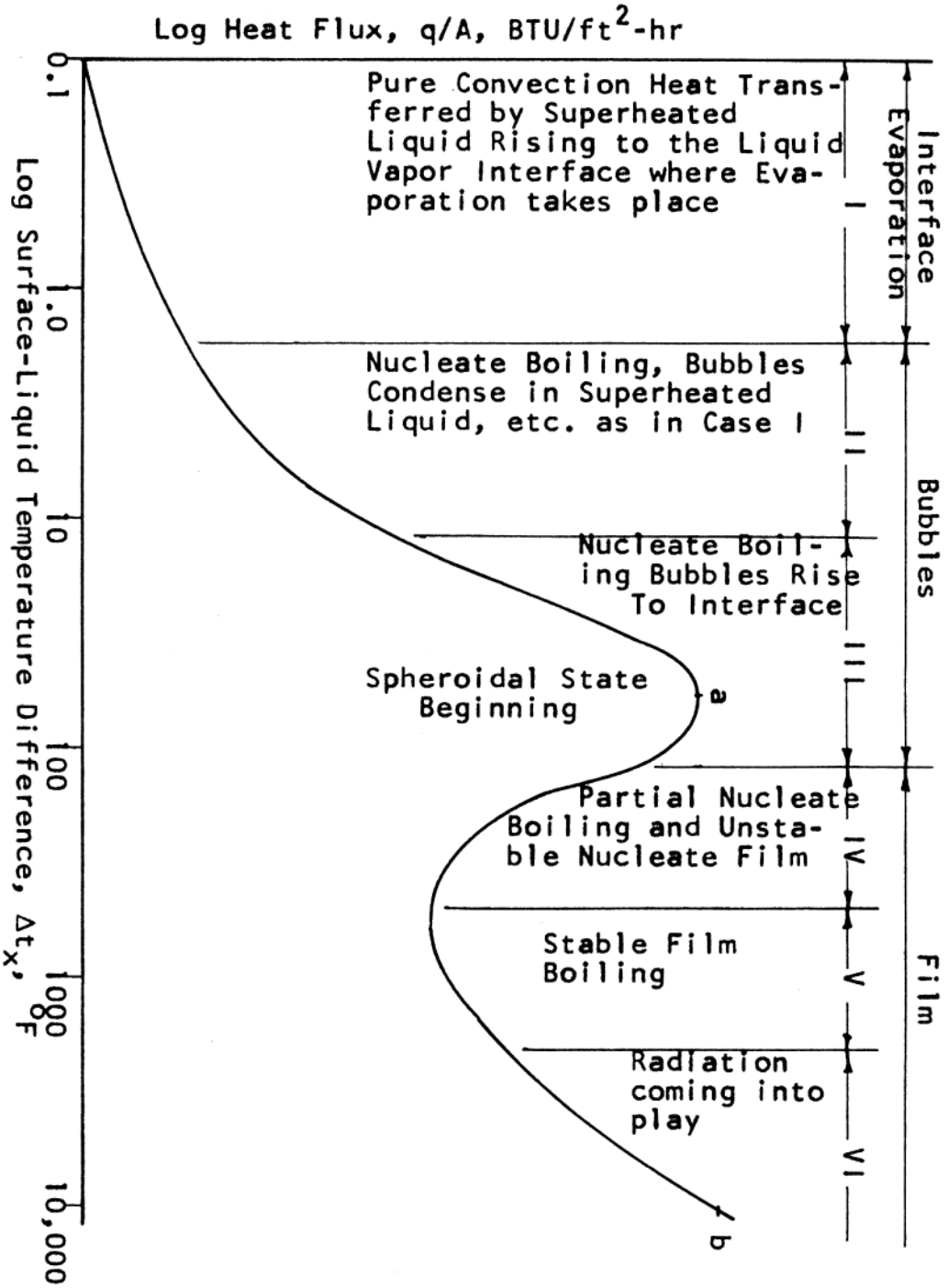

Figure 1. Boiling Curve for Water by Farber and Schorah[3]

superheated liquid is circulated by convection currents as the heater surface temperature rises above the saturation temperature and vapor is produced by evaporation at the free surface. Kreith explains, as the surface temperature is increased further, vapor bubbles are formed and rise at "favored spots" on the heater surface but condense before reaching the free surface. It is in this area that nucleate boiling begins, but complete and stable nucleate boiling does not start until regime III is attained. McAdams stated that the reason the term "favored spots" is used is because at a given temperature, "the vapor pressure over a very small concave liquid surface is less than the vapor pressure over a flat liquid surface. Hence, for a given pressure, a liquid must be hotter to evaporate into a small bubble of vapor than into the vapor space above the liquid." It is therefore difficult for vapor bubbles to form and they will originate "...preferentially at those active nuclei on the heating surface where the temperature and the nature of the surface are favorable." In regime III steam is transported to the vapor space by more and larger bubbles that have been formed by the increased temperature difference. In this range of nucleate boiling many observers have found heat flux to vary as the temperature difference raised to the nth power, where the value of n varies from 3 to 4 as presented by Giedt.[5] As the temperature difference

rises to about 100 F (for water) a maximum or peak heat flux is obtained, as well as, a critical temperature difference. This is represented in Figure 1 as point (a). A further increase of the temperature results in a decrease in the rate of heat flow. An explanation for this point was made by Kreith by examining the mechanism of heat transfer during boiling. As pointed out earlier, bubbles grow at certain favored spots on the heater surface until the "buoyant force or currents of the surrounding liquid can carry them away." Kreith also found that, "Although the product of the size of the departing bubbles times the frequency of their formation at any particular spot does not vary appreciably with heat flux, the number of spots at which bubbles form increased nearly in direct proportion with the excess temperature."* This increase in bubbles per unit area promotes convection heat transfer to the liquid. The part of the heater surface covered by vapor bubbles is effectively insulated because the thermal conductivity of the vapor is much smaller than that of the liquid. Therefore, Kreith says, "Increasing the number of

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*Excess temperature is defined as the temperature of the heater surface above the saturation temperature of the liquid.

bubbles promotes the flow of heat by virtue of the agitating motion of the bubbles, but at the same time the area available for heat transfer to the liquid diminishes. However, when the number of spots at which bubbles form becomes so large that an appreciable portion of the surface is covered by vapor, the insulating effects overshadow the beneficial effects of fluid agitation and the heat flux decreases with increasing excess temperature." The regimes IV, V, and VI are known as film boiling because all or an appreciable portion of the heater surface is blanketed by vapor. Kreith further pointed out that the maximum or peak heat flux occurs just before the transition from nucleate to film boiling takes place. In regime IV an unstable film forms and large bubbles originate at the outer upper surface of the film. Because of the action of circulation currents this vapor film collapses and reforms rapidly. According to Rohsenow, the existance of this unstable film "provides additional resistance to heat transfer and reduces the heat transfer rate." In the range of temperature difference between 400°F and 1000°F, (water), the influence of radiation becomes pronounced. In this area "the vapor film is very stable, and the orderly discharge of bubbles suggests that the frequency and location of bubbles origination is controlled by factors operating at the outer surface of the film and that favored spots along

the wire are without effect."

Rohsenow stated that the characteristic boiling curve, Figure 1, is easily obtained if condensing vapor heating is used, but when electrical heating is used, the regime IV is not readily obtainable. "As the electric energy, and hence heat flux is increased, the resulting temperature difference increases in regime III. When the peak value of heat flux is reached, any further increase in electrical energy is accompanied by a lower heat flux. The difference between these two quantities causes a rise in the internal energy of the wire, accompanied by a further decrease in heat flux." Therefore, this system is unstable and will proceed to point (b) where the temperature is very great, unless the electrical input is reduced. Usually, point (b) represents a point beyond the melting point of most metallic heaters. For this reason point (a) is called the burnout point, as stated by Rohsenow.

Now that all regimes on the characteristic curve of heat flux as a function of temperature difference have been generally reviewed, a more detailed examination of factors affecting regimes III, IV, V, and VI is necessary for completeness.

NUCLEATE BOILING INVESTIGATIONS

It is known that many factors influence nucleate boiling. Although all of these factors have not been

fully evaluated they are, as presented by McAdams, (1) nature of the surface, (2) pressure, (3) addition agents, (4) scale deposits, (5) temperature difference, (6) plate, tube or coil size and arrangement, (7) mass flow, (8) nature of liquid, and (9) turbulence. McAdams summarized the findings of many investigators who demonstrated that the boiling film coefficients increase in proportion to each of the following: (1) an increase in surface roughness or grooved surfaces, (2) an increase in pressure above atmospheric, up to one-third of the critical pressure of a particular substance, (3) the addition of certain chemical agents, (4) removing scale deposits since even a thin layer of scale may have a large reducing effect on the overall coefficient, (5) an increase in temperature difference as shown by Figure 1, (6) the number of tubes or coils but not the tube diameter, (7) an increase in mass flow, (8) an increase in turbulence. The boiling film coefficient is also affected by the nature of the liquid. McAdams also presented data, in his Figure 14-15, of Cryder and Finalborgo showing the effect of the nature of the liquid on the boiling film coefficient for a single horizontal tube.

Rohsenow stated that there has been no general correlation of all boiling heat transfer data to obtain a single equation that would correlate the effects of pressure,

type of heater surface and kind of liquid over the entire range of temperature difference. However, correlations have been made within regime III. The effect of pressure has been successfully correlated in this regime and this can be used to obtain a rough estimate of peak heat flux.

When examining the process of nucleate boiling in a vertically heated tube, that is, a natural circulation evaporator, Rohsenow found it to be more complicated than the pool-boiling* process. As liquid near or at its saturation temperature is admitted at the lower end of the natural circulation evaporator, vapor bubbles form on the tube wall and rise in the comparatively narrow space, interfering with the motion of each other as they proceed toward the vapor space at the top. As the fluid flows upward it changes in density and velocity. McAdams discussed this subject further and said that in the upper portion of the tube the fluid is saturated, even if it is initially admitted below the saturation temperature, assuming the temperature difference to be sufficiently large. McAdams stated that, "vaporization occurs progressively,

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*Kreith described pool-boiling as existing in a "a simple system consisting of a heating surface, such as a flat plate or a wire, submerged in a pool of water at saturation temperature without external agitation."

and a two-phase mixture emerges from the top of the tube. The rate of heat transfer at any point in this section of the tube depends upon the local temperature difference and upon the relative flow rates of liquid and vapor." Further, the velocity of the mixture increases as the weight fraction of the vapor becomes larger because of the greater volume occupied by the vapor. Also, according to McAdams, "the pressure gradient is increased because of greater wall drag and because of the force necessary to accelerate the stream to the higher velocities; hence the saturation temperature usually falls more rapidly near the outlet of the tube." Because of the large number of experimental variables which must be included in any evaluation and in spite of the wide-spread use of the natural circulation evaporator the basic heat-transfer and fluid-flow phenomena are not yet fully understood. Several investigators conducted experiments with fluids being admitted to the evaporator below its saturation temperature thereby having a nonboiling section and a boiling section. Their analyses were inconclusive with regard to determining an equation for predicting the boiling film coefficients. Stroebe, Baker, and Badger[6] separated the characteristics of the boiling section from those of the nonboiling section by studying a vertical, 2 inch O.D., twenty foot copper tube with boiling along its entire length. This

condition was obtained by preheating the liquid feed. The following equation was derived:

$$h = \frac{7.8 \times 10^6 \, v^{0.1}}{\left(\frac{c_p \mu}{k}\right)^{0.3} \sigma^2 \, \Delta t^{0.13}}$$

The symbols are defined as follows:

h = boiling film coefficient, BTU/hr-ft^2-$^\circ$F

Δt = temperature difference between the heater surface and the liquid bulk, also termed "excess temperature", $^\circ$F

v = specific volume, ft^3/lbm

μ = viscosity of saturated liquid, lbm/ft-hr

σ = surface tension of the liquid-surface interface, dynes/cm or lbf/ft

k = thermal conductivity, BTU/hr-ft^2-$^\circ$F/ft

c_p = specific heat, BTU/lbm-$^\circ$F

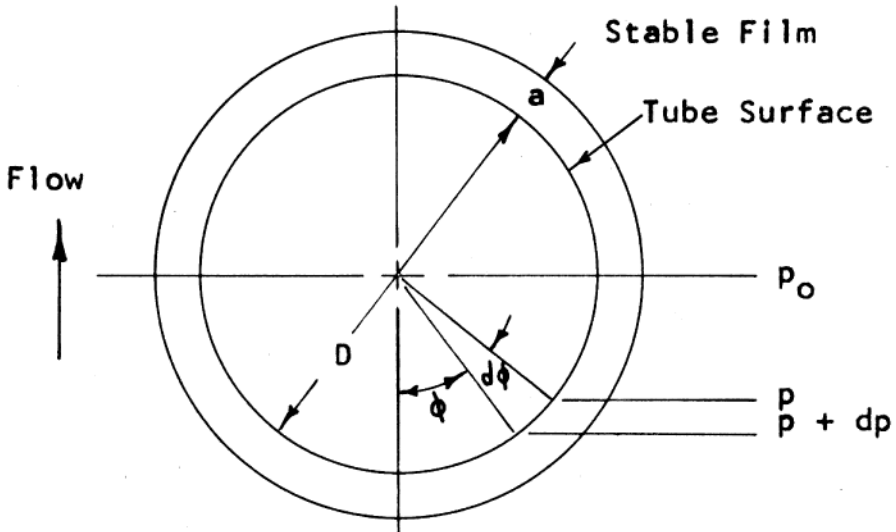
These authors found that 90 per cent of the runs made fell within $\pm$ 20 per cent of the values calculated from this equation. It was pointed out that this equation was entirely empirical and should be used with discretion for conditions that are very different from those in this investigation.

FILM BOILING INVESTIGATIONS

Film boiling exists beyond the point of maximum or peak heat flux. These areas are represented as regimes

IV, V, and VI of Figure 1. In regime IV there is an unstable film formation over the heating surface and large vapor bubbles originate at its outer surface. This film will periodically collapse and form again under the influence of circulating currents.[4] In regime IV there is a decrease in heat flux with an increase in the temperature difference between the bulk liquid and the heater surface. This decrease must be due to an increase of the thickness of the vapor film which is relatively larger than the increase in temperature difference.[8] Regimes V and VI were studied analytically and experimentally by Bromley[7]. To analyze the process for a horizontal tube, Figure 2, he made the following assumptions:

1. A continuous vapor blanket of thickness "a" exists.
2. Vapor rises in viscous flow by buoyant forces
3. Liquid-vapor interface is smooth in the section where most of the heat is transferred
4. Rise of vapor is retarded by viscous drag on tube and on liquid. Contribution of liquid drag is unknown; therefore, it is included in a constant to be determined by experiment
5. Assume q = rate of evaporation times $(h_{fg} + c_v \Delta t/2)$, where c_v = specific heat of saturated vapor, BTU/lbm-^{o}F
6. Neglect kinetic energy and momemtum changes in



The symbols p_o, p and p + dp represent a pressure gradient across the tube.

Figure 2. Sketch of Idealized Stable Film on a Horizontal Tube, by Bromley

vapor film

7. Assume temperature of the wall and saturation temperature of the fluid to be uniform around the tube
8. Evaluate physical properties at mean value between the temperature of the wall and the saturation temperature of the fluid
9. Combined effect of most errors introduced by the above assumptions may be corrected by evaluating a factor experimentally
10. At the liquid-vapor interface, the temperature of the liquid = the saturation temperature of the fluid.

From the laws of hydrostatics and the above assumptions Bromley derived the following equation:

$$h = \text{constant} \left[\frac{k_v^3 \rho_v (\rho_L - \rho_v)\, g \lambda}{D \mu_v \Delta t} \right]^{1/4}$$

The symbols are defined as follows:

$$\text{constant} = \frac{0.9536}{B^{1/4}} = 0.62 \pm 0.04$$

B = experimental constant = 3.65

D = diameter of tube, feet

k_v = thermal conductivity of the vapor, BTU/hr-ft^2-^{o}F/ft

g = acceleration due to gravity, 4.17×10^8 ft/hr^2

ρ_v = density of vapor, lbm/ft^3

ρ_L = density of liquid, lbm/ft^3

$\lambda = h_{fg}$ = latent heat of vaporization, BTU/lbm

All other symbols are defined on page 19. This equation has been copied and explained by Rohsenow, McAdams, and Kreith. "This equation permits the calculation of the conduction part of the heat transfer through the film. Superimposed on this is the contribution of radiation, if it is significant."[1]

BOILING OF BENZENE

McAdams[2] presented data, in his Figure 14-27, of Woods and Bryan showing the behavior of boiling benzene in the regimes of nucleate and partial nucleate boiling. The curve for 100% feed vaporized showed a definite peak at a length-mean temperature difference of 75°F and an average heat flux of 18,000 BTU/hr-ft^2. The range investigated was from a temperature difference of 30°F and a heat flux of 12,000 BTU/hr-ft^2 to a temperature difference of 90°F and a heat flux of 17,000 BTU/hr-ft^2. From these data it was calculated that the overall coefficients decreased from 400 BTU/hr-ft^2-°F at a temperature difference of 30°F to 190 BTU/hr-ft^2-°F at a temperature difference of 90°F. These data were taken from a boiling section of a horizontal four-pass forced-circulation evaporator. These investigators used a steam-jacketed, standard, one inch copper pipe. Low feed velocities were used (0.26 to 1 ft/sec) so that most of the feed could be evaporated as required. It was noted

that with the lower temperature differences, as the fluid is progressively vaporized the local overall coefficient at first increases, goes through a maximum, and then decreases markedly "toward values typical of superheating dry vapor". McAdams attributes this dry-wall vapor binding to insufficient liquid to wet the walls.

SULFUR DIOXIDE BOILING FILM COEFFICIENTS

One of the earliest published works of sulfur dioxide boiling film coefficients in a flooded evaporator was released in 1931 by Stewart.[9] His main objective was to "determine some of the heat transfer characteristics of a given set of coils tested under known conditions and then to determine the effect of the spacing of the extended surfaces." He used extended surface evaporator coils with liquid SO_2 on the inside and air on the outside. Each unit tested had six tubes with three 180° bends forming a coil when attached to a reservior drum, see Figure 3. A liquid level of SO_2 was maintained in the drum. The tubes were made of copper and the extended surfaces were made of aluminum. To determine the heat transfer characteristics of the various coils Stewart varied the mean temperature difference of the SO_2 and air from about 10°F to 25°F. Humidity of the air was maintained at saturated conditions and the SO_2 temperature in the coil was maintained at 30°F

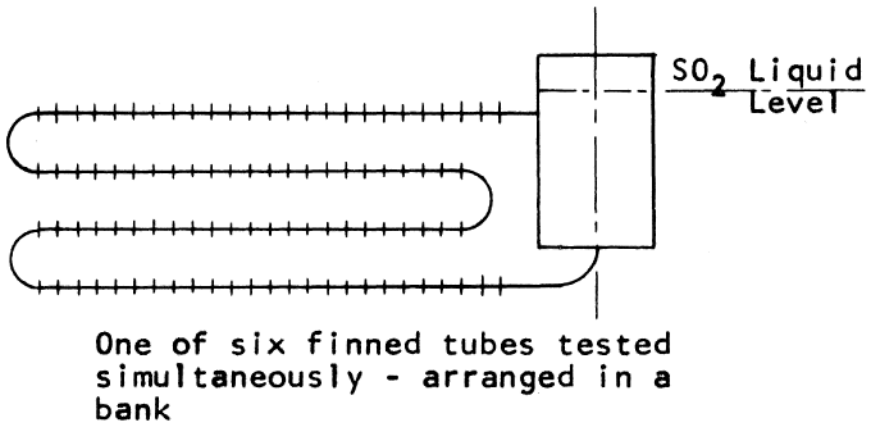


Figure 3. Coils Tested by Stewart

by pressure regulation throughout the experiment. The thermal load was imposed by electric heaters. The results obtained showed a maximum boiling film coefficient of only 0.9 BTU/hr-ft^2-$^\circ$F at a heat flux of 30.6 BTU/hr-ft^2 for a 1-1/2 inch coil. Stewart observed that at irregular intervals the temperature readings showed great variations for apparently steady air temperatures. He noted variations as high as "7°F to 8°F for the 3/4 inch spacing coil operating at 2000 BTU/hr with 30°F SO_2. Greater variations in (bulk liquid) temperature were observed as the SO_2 was reduced in temperature." After windows were installed in the drum, "violent gushing at the outlet of the tubes was observed." Stewart explained that this phenomenon was due to "the surface tension effect of the oil in the 1/2 inch diameter tubes." He called this effect a "geyser action of the refrigerant in the tubes. Stewart also noted that this was an important fault to be overcome in many of the current (1931) designs of coils. Because these temperature readings were erratic they were not used in computing the results already mentioned, but the SO_2 temperatures in the drum were used instead. Stewart was unable to correlate his data into any kind of equation.

King[10] presented results of evaporating SO_2 in a heavy copper cylinder. Since King's primary objective was to present a current (1932) survey of heat transfer, his

treatment of the SO_2 experiment was brief. Although the method and conditions of the experiment were not revealed, several conclusions were drawn and published. King observed that periodic ebullition and temperature cycles occurred and the experimental readings were irregular. He identified these irregularities as a "geyser action" which had been reported by Stewart earlier. King pointed out that Stewart was apparently in error by attributing this "geyser action" to the surface tension of the oil in the tubes. Since King's experiment was free of oil, he surmised that the "geyser action" could be explained by the periodic superheating of the liquid, up to the point where the vapor pressure overcame the internal tension. A liquid is considered to be in a superheated state when its temperature and pressure indicate that it should be in a vapor state or at least should be boiling rapidly. King also conducted some general tests on superheating a pure liquid (methyl formate) by reducing the pressure. He reduced the pressure above a column of methyl formate until it supported a 40 inch column of mercury, which is equivalent to a "tension" of 5 lbf/in^2, without ebullition occurring. He also found that the tendency of a pure liquid to become superheated is reduced substantially by the presence of a dissolved gas, such as air. In addition to this study, he plotted a curve of heat flux versus boiling film coefficient for

SO_2. The plot was on rectangular coordinates and the curve was essentially a straight line. The minimum boiling film coefficient was 230 BTU/hr-ft^2-^oF at a heat flux of 800 BTU/hr-ft^2 while the maximum boiling film coefficient was 530 BTU/hr-ft^2-^oF at a heat flux of 5200 BTU/hr-ft^2. These values indicate that the range of temperature difference was approximately from 4^oF to 10^oF. The slope of his curve indicated boiling in the stable nucleate regime. King concluded that there was "no apparent limit to the extent to which a pure liquid may be superheated" and any reduction in superheating results in higher heat transfer coefficients.

In a work published in 1933, Phillip and Tiffany[11] studied "Ebullition of Refrigerants" in detail. Their main objective was to present observations regarding the "equilibrium of liquid refrigerants and the corresponding vapor for the dynamic process of liquid refrigeration evaporation." These authors began their discussion by pointing out that Stewart was in error when he said that the "geyser action" observed was caused by the surface tension of the oil in the tubes. Stewart's statement was proven wrong by a series of tests involving lubricants and SO_2. The tests revealed that the elevation of the boiling point was not appreciable (approximately 0.2^oF) for a solution of oil-saturated SO_2. "The fact that the temperature of the pure liquid sulfur dioxide was observed

to be on the average about three degrees higher than the temperature corresponding to the pressure for static equilibrium must have been caused by superheating of the liquid." The flooded system tested by Phillip and Tiffany for determination of the correlation of erratic operation and superheating of the liquid refrigerant, SO_2, was a commercial cross fin coil evaporator. It was made of two "hair pin" shaped 3/4 inch O.D. copper tubes fastened to a horizontal cylindrical shell. The "hair pin" loops were set in a vertical plane and carried the copper fins. The liquid SO_2 level was maintained by a float inside the horizontal cylindrical shell (float chamber). Liquid SO_2 was evaporated from inside the "hair pin" loops while air was circulated on the outside of the fins. This investigation was prompted by the unpredictable operation and low capacity of this commercial evaporator. In order to gain a fundamental understanding of SO_2 ebullition these authors constructed a glass "hair pin" loop with a vertical heater. Chromic acid was used to clean the glass evaporator. It was found that a mixture of SO_2 and lubricating oil, which was at atmospheric pressure could be warmed to room temperatures without ebullition occurring. To further demonstrate this phenomenon Phillip and Tiffany carried out a series of ebullition tests with cleaned glass Erlenmeyer flasks. About a quarter of an inch of lubricating oil was poured

on top of the liquid SO_2 in each of these flasks. It can be assumed that initially the temperature of the flasks was below 14°F (the boiling point of SO_2 at atmospheric pressure). As the mixture temperature was allowed to rise to room temperature, some of the mixtures would not boil, while others would rise to nearly room temperature and then evaporate with almost explosive violence. At least half of the mixtures were superheated to 50°F without ebullition. "Pure liquid SO_2 could be superheated as much as 10°F above its boiling point." Stewart's "geyser action" was in reality a case of superheated liquid which had evaporated suddenly with the liberation of a large volume of vapor. Superheating is not only undesirable because of the "geyser action" but also because of the lowering of heat transfer capacities of evaporators. Phillip and Tiffany found that liquid SO_2 had to have some type of catalytic activity to promote the formation of vapor and would decrease the degree of superheating. These authors tried many types of ebullators, that is boiling inducers, and found that when fibrous materials of a vegetable nature were used they were successful. With the ebullators the degree of superheating was kept below 3°F. It was also discovered that air or vapor would act as ebullators if either was admitted as bubbles and allowed to pass up through the liquid SO_2. "These bubbles provided a surface

for the evaporation of the liquid, the condition being similar to the evaporation of pure liquid SO_2 from the plane surface of the liquid." With this method it was found impossible to superheat the liquid over 2°F. A thorough discussion was presented regarding a theoretical analysis of bubbles and their formation and was summarized as follows: in dynamic evaporation of liquid SO_2, a slight superheating of the liquid can be expected after the beginning of a bubble at a solid-liquid interface. The authors concluded that "the degree of superheating is:

1. Directly proportional to the heat of evaporation.
2. Inversely proportional to the coefficient of heat transfer from liquid to vapor.
3. Inversely proportional to the specific volume.
4. Directly proportional to the rate of rise of the bubble.
5. Directly proportional to the size of the bubble."

An article was published in 1933 entitled "Film Heat Transfer Coefficients for SO_2 in a Vertical Evaporator" by Stewart and Hechler.[12] The experimental evaporator, shown in Figure 4, was a standard 3/4 inch copper pipe, 80 inches long. Precautions were taken to prevent radial and end heat losses from the main heater. The evaporator and heating elements were placed in a 4 inch steel pipe

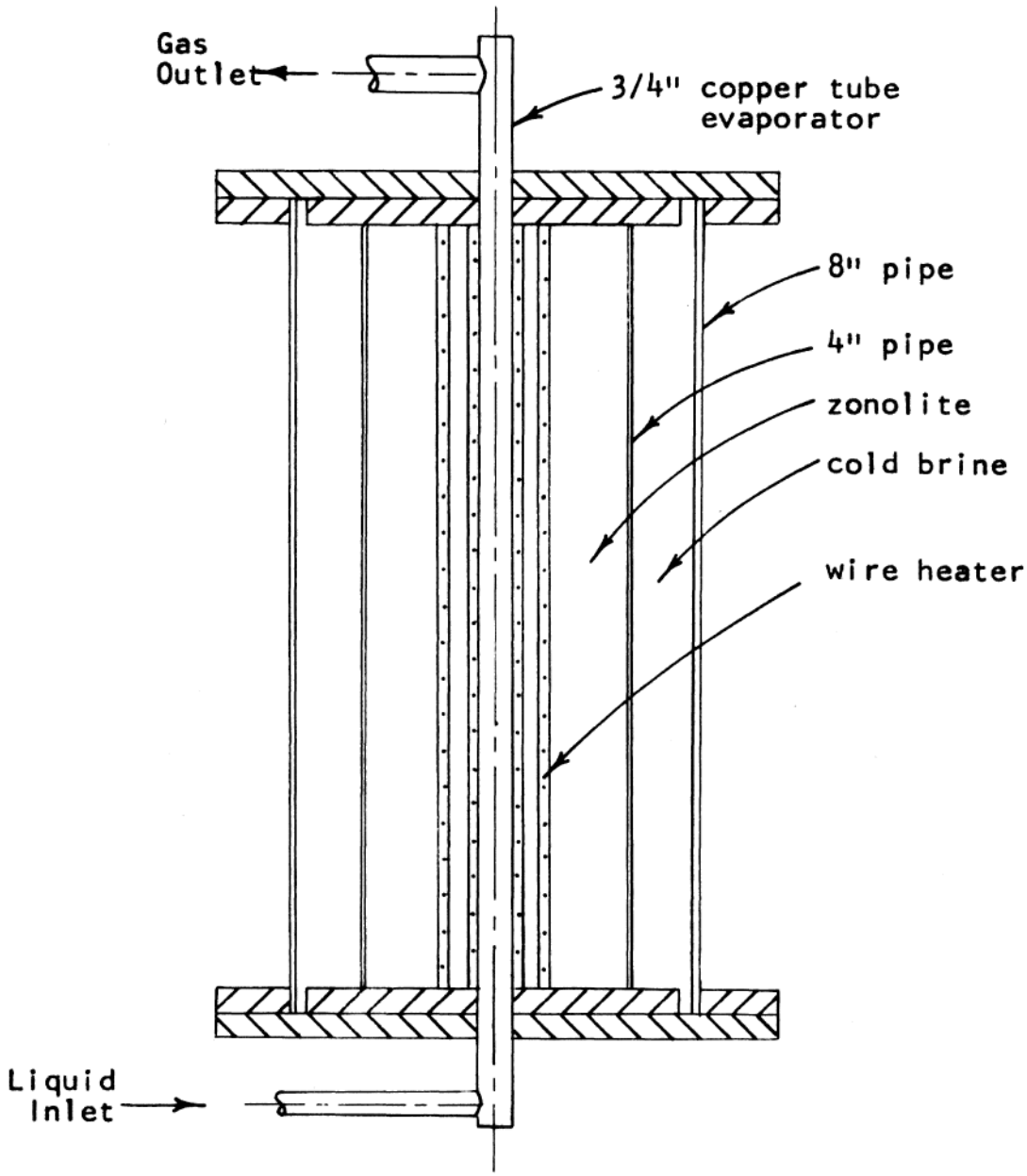


Figure 4. Experimental Evaporator Used by Stewart and Hechler

packed with zonolite. The 4 inch pipe was placed in an 8 inch pipe and cold brine was circulated through the annular space between the two pipes to maintain a constant temperature. Thermocouples were used to take all temperatures of the evaporator wall and the liquid bulk. These investigators encountered unstable operating conditions when they began their experiment, the same "geyser action" as mentioned by previous investigators. Following the recommendation of Phillip and Tiffany, small pieces of maple wood were fastened at intervals inside the evaporator tube to act as ebullators. After this was done stable operating conditions prevailed throughout the remainder of all the tests. The results were shown in the form of eight curves, one of which related the boiling film coefficients to heat fluxes and saturation temperatures. These authors found a maximum heat flux of 4000 BTU/hr-ft^2 and a corresponding value for the boiling film coefficient of 620 BTU/hr-ft^2-^{o}F and a saturation temperature of 10^{o}F. A minimum heat flux of 1000 BTU/hr-ft^2 produced a value for the boiling film coefficient of 580 BTU/hr-ft^2-^{o}F and a saturation temperature of 40^{o}F. The conditions and omissions of this investigation were: (1) forced circulation, (2) not all of the heater surface was exposed to boiling, (3) the liquid level was comparatively high (63 inches), (4) the surface rough-

ness was not recorded, (5) purity of the SO_2 was not recorded, (6) the amount of lubricating oil mixed with the SO_2 was not recorded, (7) the highest saturation temperature used was only $40^{\circ}F$, and (8) the highest heat flux obtained was only 4000 BTU/hr-ft^2.

After 1936, no SO_2 boiling film coefficient data was published for flooded evaporators until 1952. This can be partially explained by the conversion of the refrigeration industry from the SO_2 refrigerant to the new "Freon" refrigerants. However, there has long been an interest in SO_2 as a basic raw material in the production of many other inorganic chemicals.[13]

A more recent article including data on SO_2 boiling film coefficients was published in 1952 in REFRIGERATING ENGINEERING and a more complete release of information on the same experiment appeared in HEAT TRANSFER SYMPOSIUM in 1953.[14] The objective of the authors, Myers and Katz, was to determine "Boiling (Film) Coefficients Outside Horizontal Tubes". Although this work is unquestionably valuable, it is unfortunately brief on the subject of SO_2. The equipment used for this experiment included both plain and finned tubes. In both cases, the tubes were 3/4 inches in diameter, 36 inches long and placed in a vertical row. "Four horizontal tubes were placed one above another on

1-1/2 inch center-to-center spacing inside an evaporator chamber with the boiling liquid outside the tubes." The vapor produced by this evaporator was received by a compressor, compressed and then condensed and the saturated liquid returned to the evaporator. Hot water was circulated through the inside of the tubes and provided the heating for the boiling. These authors examined five different fluids and studied the effects on plain tubes and finned tubes. For SO_2 on plain tubes at a saturation temperature of $65^{\circ}F$, the maximum boiling film coefficient obtained was approximately 750 BTU/hr-ft^2-F at a temperature difference of $16^{\circ}F$ and a minimum boiling film coefficient of 325 BTU/hr-ft^2-F at a temperature difference of 9.5 F. For finned tubes the values obtained for boiling coefficients were almost doubled at the same values of temperature. The basic equation used for calculation of boiling film coefficients on plain tubes was:

$$h = \frac{q}{A\,\Delta t}$$

The symbols are defined as follows:

q = heat transfer rate, BTU/hr

A = surface area in contact with boiling fluid, ft^2

Other symbols are defined on page 19.

Myers and Katz found by dimensional analysis the equation

$$\frac{h}{k}\sqrt{\frac{\sigma}{\rho_L}} = m\left(\frac{k\,\Delta t}{\lambda\mu}\right)^n$$

All symbols have been previously defined on pages 19, 22, and 23. Experimental data was obtained with plain tubes which yielded a value for the constant m of 3.4×10^6 and an exponent n of 2.75 for SO_2. It is significant that this work was done with pilot plant size equipment and not the more usual laboratory size equipment. "It is known that the condition of the surface and its geometry have a marked influence on the boiling coefficient and that it is often not advisable to use boiling coefficients obtained in small-scale laboratory tests for the design of commercial units." Myers and Katz observed two effects of time on the boiling film coefficients. First, "the boiling coefficients decreased over a period of several days when starting with new tubes which had been cleaned with dilute sulfuric acid and rinsed with water." After the apparatus had been operated for some time the coefficients tended to level out and eventually a stable surface was attained on which no changes were observed. The second effect of time was described as hysteresis and was most pronounced with fluids other than SO_2. In short, the heat transfer rate was variable for no apparent reason when the apparatus was started up in the morning but stabilized after several hours operation. The experimenter was able to bring the operation to stable conditions quickly "by first using a

high temperature difference and then reducing the temperature difference to that desired." The explanation of this second effect remains to be found.

The most recent information available for boiling SO_2 was presented in 1958 in a paper entitled "Sulfur Dioxide Evaporator-Assembly Test Report" by Ringler[15] of the Virginia Chemicals and Smelting Company. The main objective of this test was to verify the design capacity of a particular industrial evaporator. Other objectives were to obtain data for future evaporator-assembly designs and to investigate the use of a float switch for preventing flooding of the evaporator. The test system is shown in Figure 5. The most pertinent conclusions drawn from this test were: "(1) The capacity of the system is 1560 pounds of sulfur dioxide per hour (design capacity of 2000 pounds per hour) when vaporizing from an evaporator pressure of 49 psig to a downstream or load pressure of 15 psig while heating with 25 psig saturated steam. (2) The overall heat-transfer coefficient for the submerged section was approximately 368 BTU/hr-ft^2-F, based on the outside area of the bayonet heater. A design value of 300 should be acceptable. The overall coefficient for the superheating section never exceeding 3.9 BTU/hr-ft^2-F, and it would not be practical to design an assembly of this type for more

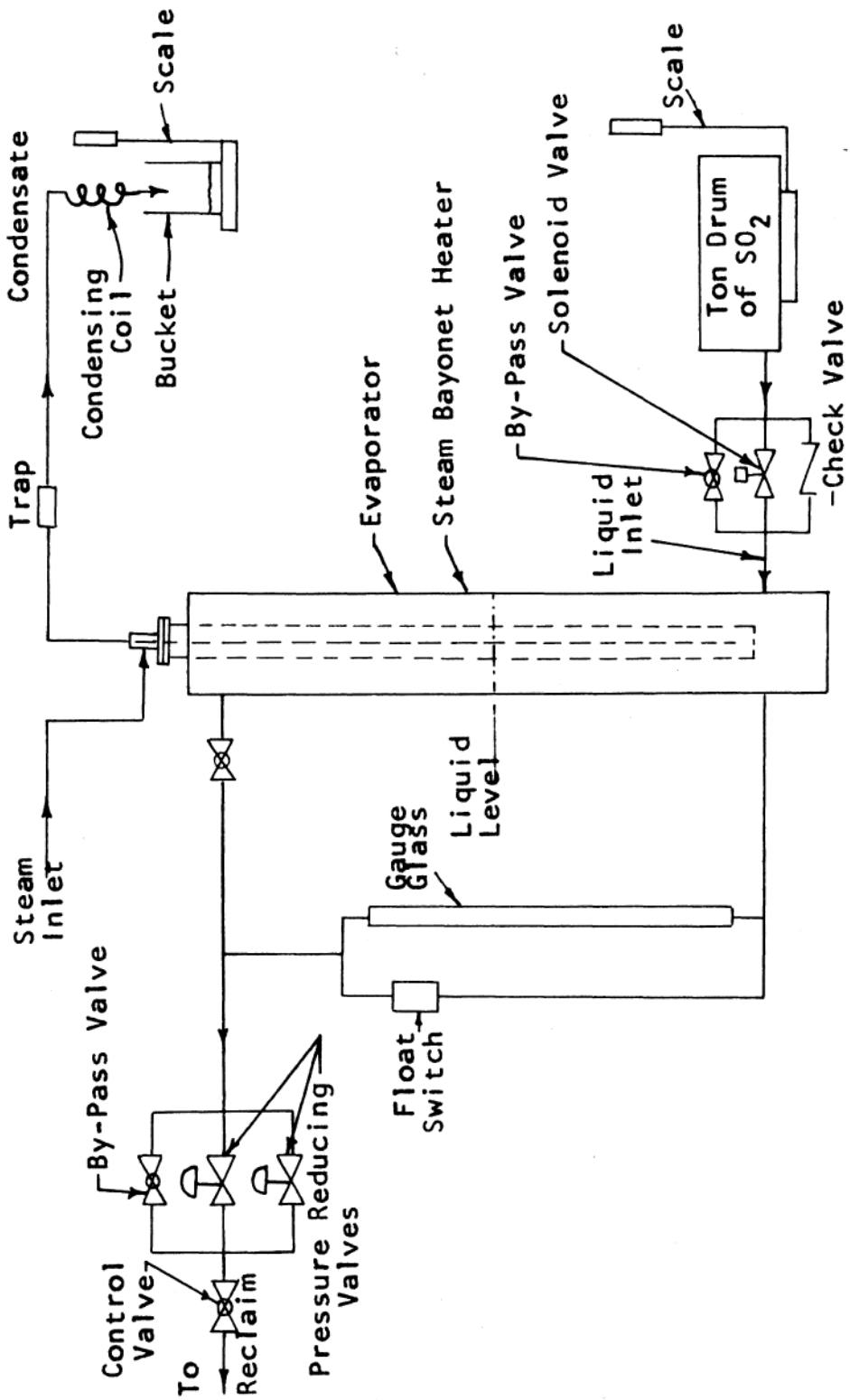


Figure 5. Test Set-Up for 2000 pounds per hour Capacity Evaporator-Assembly by Ringler

than a few degrees of superheat. (3) The float switch would definitely prevent flooding of the evaporator but it was supersensitive and opened and closed rapidly when enveloped in boiling sulfur dioxide. It should not be accepted without further testing. (4) The two parallel Mason-neilan reducing valves represent the limiting components of the system. (5) One 1-inch, Figure 40, Mason-neilan reducing valve with a 3/4 inch orifice will pass 780 pounds of sulfur dioxide gas per hour from 49 psig to 15 psig and 960 pounds per hour from 78.5 psig to 15 psig." This test was conducted with an ambient temperature above 90°F at all times, giving a relatively high pressure (80 psia) in the supply drum.

INDUSTRIAL SPECIFICATIONS FOR SULFUR DIOXIDE

Specifications and specification test procedures for sulfur dioxide were outlined in the Virginia Chemicals and Smelting Company Bulletin 501.[16] Testing of purity, moisture, non-volatile matter, acidity, color and appearance were outlined in detail. As McAdams noted, it is necessary that the experimenter be aware of the exact nature of the liquid to be tested for proper evaluation of its heat transfer characteristics.

A general bulletin was available entitled, "Liquid Sulfur Dioxide"[13] by the Virginia Chemicals and Smelting

Company. It listed the precautions necessary when handling sulfur dioxide. Also, commercial applications were noted and storage details outlined. This was the standard bulletin published for SO_2 customers.

Although sulfur dioxide is toxic, it is rare for anyone to be overcome by its fumes. This is due primarily to the very irritating nature of the gas. Since one can detect a concentration of only 3 to 5 parts per million, it is normal for an affected area to be rapidly evacuated before the concentration becomes dangerous which would be in the range of 400 to 500 parts per million. These facts and others were presented by the Manufacturing Chemists' Association, Inc. with the statement, "Sulfur dioxide gas is irritating to the throat and upper respiratory system. Contact with its liquid form will cause burns of the eyes and the skin. The burning effect on the skin is due to its attendant freezing effect. However, it has good warning properties and the hazards connected with its handling can be minimized."[16] A primary manufacturer of sulfur dioxide stated, "Sulfur dioxide is neither explosive nor flammable and does not present a serious industrial hazard provided adequate precautions are observed."[13]

SUMMARY

Although many investigators have examined the phenomenon of boiling, much remains to be explained. No single method or series of equations have been developed which fully explain or predict all parameters of boiling heat transfer. As a result, many empirical equations have been developed by various experimenters for very particular situations. The three principle areas of study are stable nucleate boiling, partial nucleate boiling, and stable film boiling.

Most investigations have been in the regime of stable nucleate boiling. Many fluids and heat exchanger configurations have been studied and correlated in this regime. This type of boiling is generally well understood although there are still large, unexplored areas remaining.

Although little work has been done in stable film boiling, Bromley's work has gained wide acceptance and has been considered the best pioneer work in this area. He has presented a reliable equation for predicting the boiling film coefficients for stable film boiling.

Since this investigation was in the partial nucleate boiling regime, experiments in this area were of particular interest. Kreith stated that there have been no correla-

tions in this regime. Occasionally work is done in this regime but valid correlations are apparently very difficult to make because of the uniqueness of each heat transfer situation. The majority of investigations utilizing SO_2 were in the stable nucleate boiling regime. Only Ringler's results for SO_2 gave some indication of being in the regime of partial nucleate boiling. The work utilizing benzene also indicated boiling in the partial nucleate regime. The temperature range investigated was nearly the same for benzene as that covered in this experiment. Also, the shape of the heat flux versus temperature difference curve was concave downward which was closely paralleled by the heat flux versus temperature difference curve of this investigation. With the foregoing literature, satisfactory comparisons were possible in the discussion of this thesis.

III. INVESTIGATION

OBJECT OF INVESTIGATION

It has been the intent of this investigation to determine boiling film coefficients of SO_2 at a series of flow rates for a vertical, bayonet type, steam element. Further, it has been intended to design and attain a method of circulating liquid SO_2 and reclaiming SO_2 vapor without the use of either a pump or a compressor.

EXPERIMENTAL APPARATUS

The basic design of the experimental apparatus involved three related systems. They were: (a) the sulfur dioxide system, (b) the steam heating system and (c) the refrigeration system.

SULFUR DIOXIDE SYSTEM

The general outline of the SO_2 flow was: (a) liquid SO_2 left the supply drum at a high pressure, (b) liquid SO_2 entered the evaporator at saturation conditions, (c) a level of SO_2 liquid was maintained in the evaporator shell, covering the steam heater, (d) the steam in the heater gave off heat by condensing and the SO_2 absorbed the heat by boiling, (e) the saturated SO_2 vapor left the

evaporator and passed through a pressure reducing valve, (f) the lower pressure SO_2 passed through a rotameter, (g) the vapor entered the SO_2 condenser and was condensed by the evaporation of Freon-12, (h) the condensed SO_2 entered the supply drum. The SO_2 system was made up of many components which are described in detail in the following paragraphs.

The design of the evaporator and steam bayonet heater consisted of a redesign of an existing industrial evaporator to facilitate laboratory measurements. Since the general configuration of a heat exchanger will influence its heat transfer characteristics, it was necessary to retain the overall dimensions of the industrial evaporator. The original evaporator is shown in Figure 6 and the modified evaporator is shown in Figure 7. The primary difference between these two units was the method of handling the steam. In the original unit the steam was supplied from the top and the condensate taken out by a smaller, concentric pipe also from the top. The laboratory unit was similar, but the steam supply and condensate were fed to, and taken from, the bottom of the unit. The reason for this new arrangement was to avoid the unpredictable condition of having the steam condensate level build up inside the bayonet heater and then be pushed out in irregular cycles.

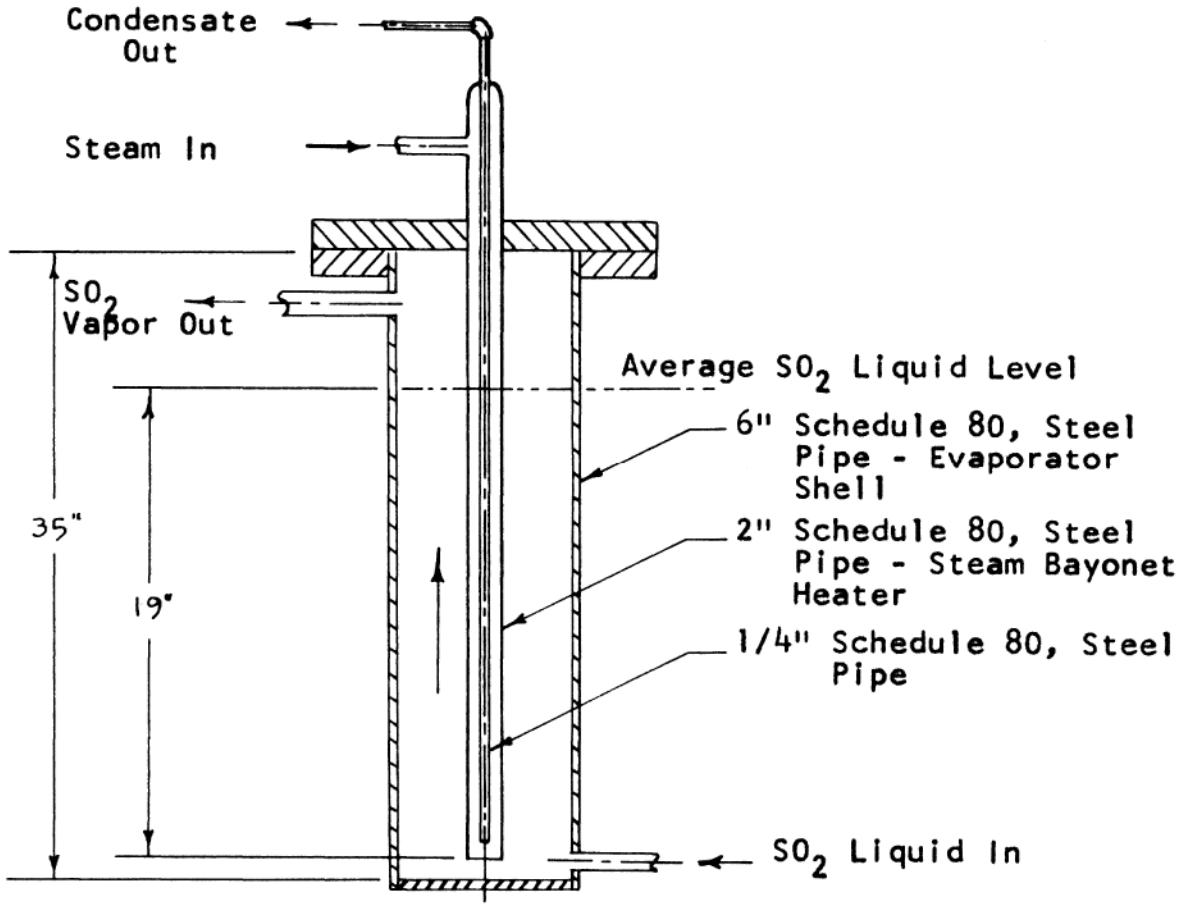


Figure 6. Original Industrial Evaporator
(Courtesy of Virginia Chemicals and Smelting Company)

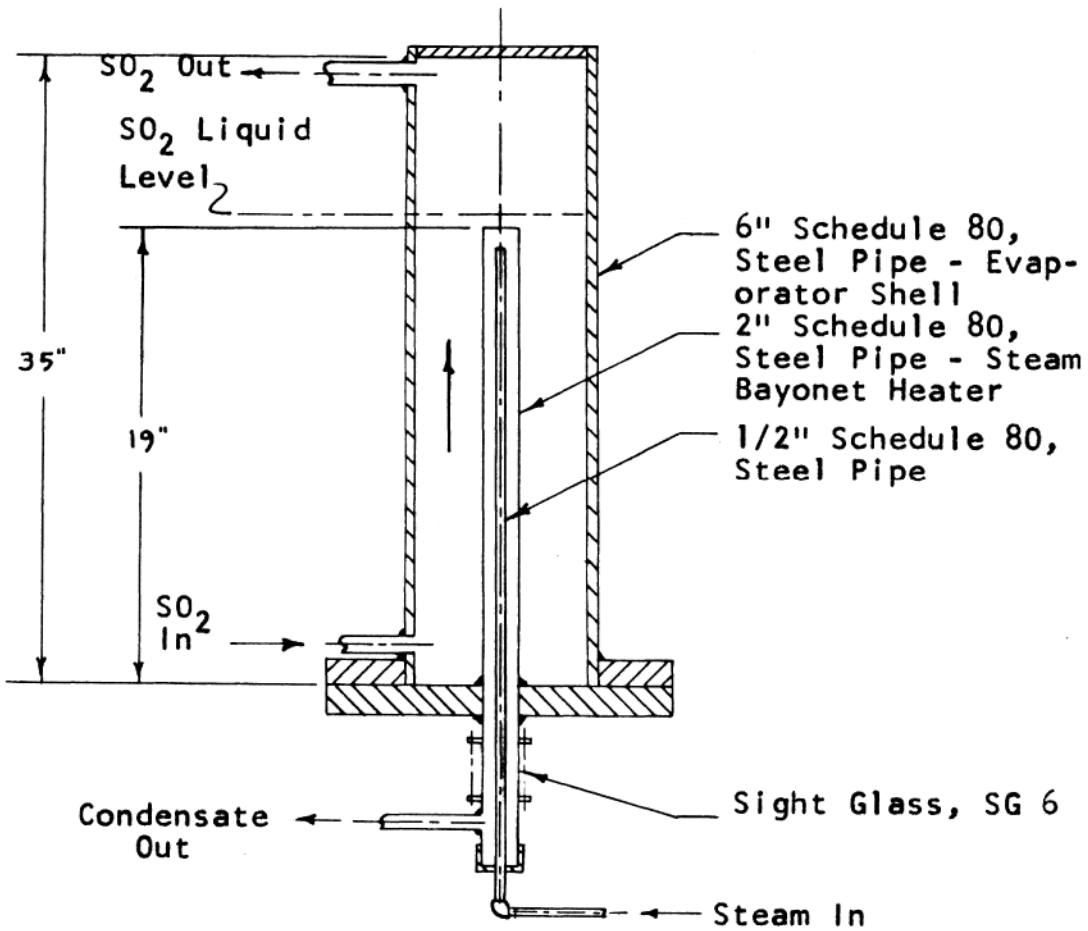


Figure 7. Modified Evaporator For Laboratory Use

The bayonet heater was shortened for the laboratory unit, omitting the part of the heater in the vapor space because this study was confined to evaluating film boiling coefficients and was not to include film coefficients of the dry gas. This modification slightly improved the condensing capacity of the system and removed the need for evaluating an average surface area covered by a surging sulfur dioxide liquid level.

After the evaporator was designed, a system to supply a constant flow of liquid SO_2 at a constant temperature was also designed and built. The system was capable of condensing all of the SO_2 vapor generated by the evaporator. This system is illustrated schematically in Figure 8 and shown in the photograph, Figure 9.

Sight glasses were located in the pipe lines upstream and downstream of the evaporator, see Figures 8 and 9. The sight glass preceding the evaporator, SG 1, was used to visually verify that the liquid was not boiling prior to entering the evaporator. The sight glass in the vapor line, SG 2, was used to visually verify that the vapor had not entrained any liquid SO_2. The sight glass, SG 3, located along the length of the evaporator indicated the liquid level and also gave some indication of the condition of the boiling liquid. The SO_2 liquid level in the drums

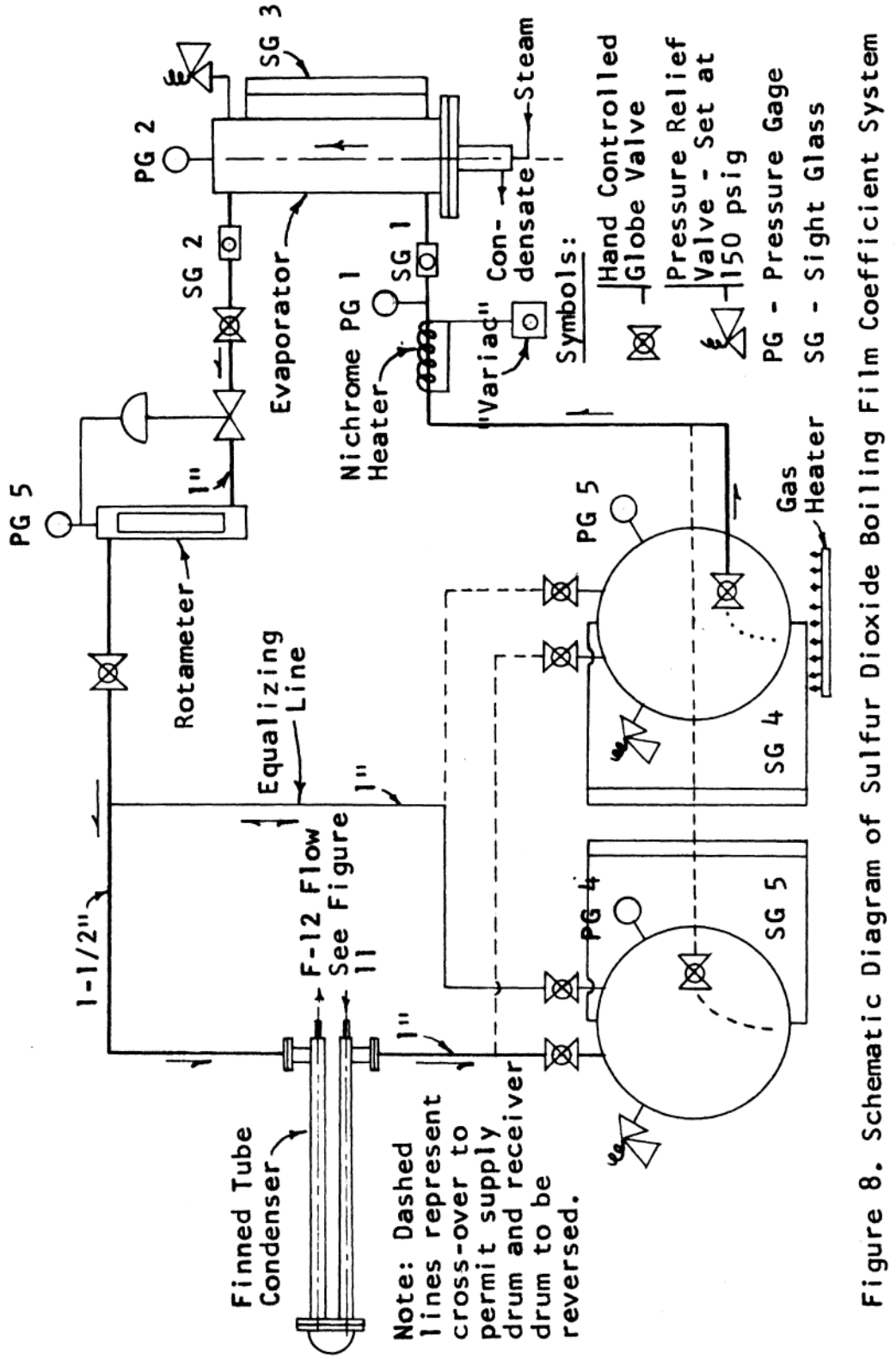


Figure 8. Schematic Diagram of Sulfur Dioxide Boiling Film Coefficient System

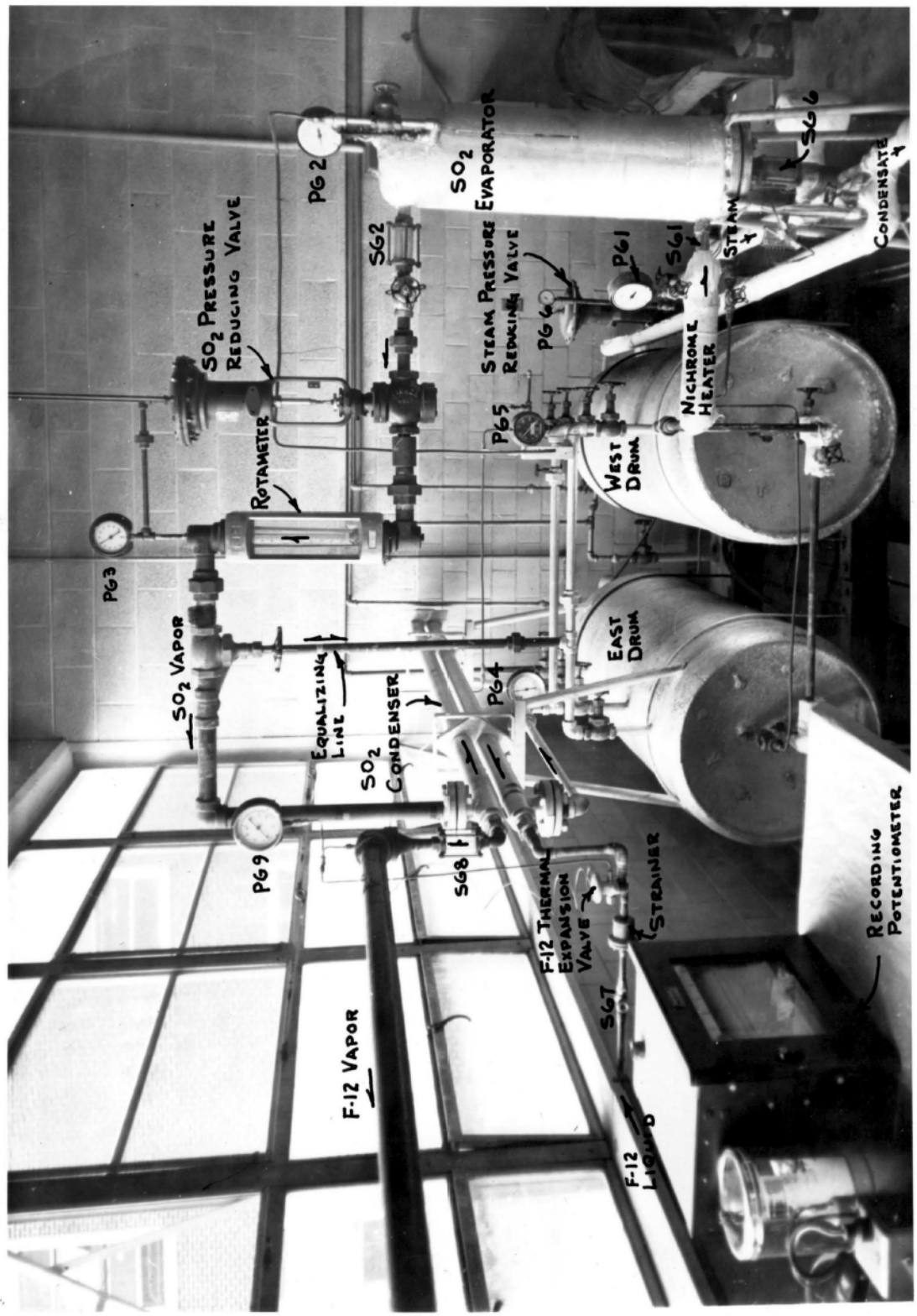


Figure 9. Photograph of Experimental Apparatus

was indicated by sight glasses attached to the end of each of the two SO_2 drums, see Figure 8. In addition these sight glasses were used to check the validity of the readings obtained from the rotameter.

The globe valves used in the SO_2 system were made by the Lunkenhiemer Company. These valves were type 16 PS, packed with Teflon, had type 316 stainless steel discs and seats, and the body material was bronze and rated at 300 psi. Even with these high quality valves some difficulty was encountered when the valve discs and seats became corroded. This condition prevented the valves from completely closing. The situation was corrected by replacing the seats and discs in the affected valves. This was done at some personal risk because of the difficulty of isolating the affected valves, singly or jointly. This maintenance operation was accomplished with a special tool for the valve seats and a gas mask for the investigator. The SO_2 concentration in the room went above 300 parts per million momentarily and could be detected through the gas mask. The possibility of the reoccurrence of the corrosion was minimized by closing all of the valves in the system during the shutdown procedure.

The SO_2 piping was all schedule 80, black steel and the fittings were also steel and rated at 3000 psi. This

heavy duty system was designed primarily to combat the corrosion inherent in a mixture of SO_2 and a trace of water. The design conditions for the piping were as follows:

SO_2 liquid: temperature = 90°F
pressure = 71 psia
flow = 500 pounds per hour
pipe size = 0.742" inside diameter
equivalent length = 10 feet
turbulent flow
clean steel pipe

From Reference 18 - Pipe Flow Chart:
pressure drop = 0.001 psi/ft.
total pressure drop = 0.001 x 10 = 0.01 psi, which is acceptable.

SO_2 vapor: temperature = 90°F
pressure = 50 psia
flow = 500 pounds per hour
pipe size = 1.500" inside diameter
equivalent length = 30 feet
turbulent flow
clean steel pipe

From Reference 18 - Pipe Flow Chart:

$$\text{pressure drop} = \frac{0.015 \text{ psi/ft}}{3.65 \text{ atmospheres}} =$$

$$0.0041 \text{ psi/ft}$$

total pressure drop = 0.0041 x 30 =

0.122 psi, which is acceptable.

Any pressure drop less than one psi would have been acceptable.

The automatic pressure reducing valve was manufactured by the Fisher Governor Company. The valve was type 655A, serial number 3324407, iron body, type 316 stainless steel trim, ASA rating of 250 psi, and had one inch screwed pipe connections. The valve was actuated by the downstream pressure acting on a diaphragm which in turn acted against a spring and the valve plug was opened or closed as required.

The finned tube heat exchanger was purchased from the Griscom-Russell Company on a performance guaranteed basis for the following design conditions:

Condensing SO_2 at the rate of 400 pounds per hour

SO_2 on the shell side at 100°F and 70 psig

Freon - 12 on the tube side at 2°F and 10 psig

Freon - 12 flow = 843 pounds per hour

Heat exchanged = 58,400 BTU/hr

Mean temperature difference = 98°F

The unit supplied had a total surface area of 36.2 square

feet. The material of the heat exchanger was steel. The designation was type AS18S-124, Twin G-Fin Section.

Three pressure relief valves made by the Kerotest Company were included in the SO_2 system and were set to relieve the pressure automatically if it exceeded 150 psig. They were type R with 3/8" nominal pipe thread connections. The design capacity of each pressure relief valve was 102 cfm. Because of the objectionable odor of SO_2, the discharge of each relief valve was piped outside of the building.

The nichrome heater, shown in Figures 8 and 9, was made by wrapping 18 AWG wire around the 3/4" inlet pipe after electrically insulating the pipe. The wire was nichrome V, rated at 0.4062 ohms per foot. There were 48 turns spaced at 1/4 inch intervals along the pipe for a total length of 187 inches. The heater was controlled by an autotransformer with an on-off switch and a knob to vary the voltage. The instrument was a "Variac", made by the General Radio Company, 0-135 volts, with subdivisions of 5 volts, type W5Mt.

A portable gas heater was fabricated to use the laboratory gas and compressed air to maintain a constant temperature in the supply drum. The gas heater was made from a 16 inch long, 3/4 inch, schedule 40, steel pipe with

22 1/8-inch diameter holes drilled along the length as gas jets. This pipe was connected to a 1/2 inch supply pipe in which the gas and compressed air were mixed prior to combustion. The heater was very effective in raising, as well as, maintaining the temperature in the supply drum at the required value.

The insulation used on the SO_2 system consisted of 1-1/2 inch thick 85 per cent magnesia on the evaporator and one inch thick 85 per cent magnesia on the liquid supply line. The thicknesses were based on the insulation manufacturer's recommendations for the temperature differences encountered.

A one-quarter inch copper tube was connected from the high pressure vapor space of the supply drum to the liquid inlet of the evaporator. This was done to remedy a condition of ebullition in the evaporator, should such a condition occur. This practice was recommended by Stewart and Hechler.[12]

The two storage drums were used alternately as supply and receiver for the system. This was accomplished by the valving arrangement shown on Figure 8. The drums were designed for shipping and storage containers for 2000 pounds of SO_2 each. The drums were 30 inches in diameter and 80 inches in length. The drums conformed

to Interstate Commerce Commission specifications for compressed gas drums before modification for laboratory use. In order for these drums to be heated with a flame, as required for this investigation, the fusible plugs, which melt at 165°F, had to be welded over. Also, four one inch nipples were welded along the length of each drum and a sight glass was fabricated for each drum. Each drum was anchored to a skid which served to keep the valves and sight glass in the proper position. The 2000 pounds of SO_2 used for this investigation was shipped in an unmodified drum which conformed to the I.C.C. specifications.

After the SO_2 system was completely assembled, heat was applied to the two drums and the maximum possible vacuum was obtained within the drums and piping by using a laboratory vacuum pump. This was done to remove the non-condensible gases and water vapor. This condition was held for several hours before filling the system with SO_2 from the shipping drum.

STEAM SYSTEM

In order to provide a constant steam flow at a constant temperature and pressure a reliable steam system was designed and built. This system is shown schematically

in Figure 10. Steam was supplied from the main steam header in the laboratory at an average pressure of 40 psig. The steam pressure was reduced by either an automatic pressure reducing valve or a hand operated globe valve. The steam at this reduced pressure was essentially at saturation conditions when it entered the steam bayonet heater. The condensed steam flowed down the walls of the heater and was taken out at the bottom of the unit. This condensate weight was recorded for each test after passing through a condensate cooler. The following paragraphs describe this equipment.

The automatic pressure reducing valve was made by the Fischer Governor Company and was a type 655A, had an ASA rating of 250 psi and had a 1/4 inch orifice. The body was made of iron and the trim was type 316 stainless steel. The valve was actuated by the downstream pressure acting on a diaphragm which controlled the stem travel. The valve serial number was 3347198. This valve was used for reduced pressures down to 5 psig.

A needle globe valve was provided in the by-pass line of the automatic pressure reducing valve. This valve was used to manually control the steam flow when the downstream pressure had to be below the range of the automatic valve. The connections on the needle valve were 3/4 inch standard screwed pipe.

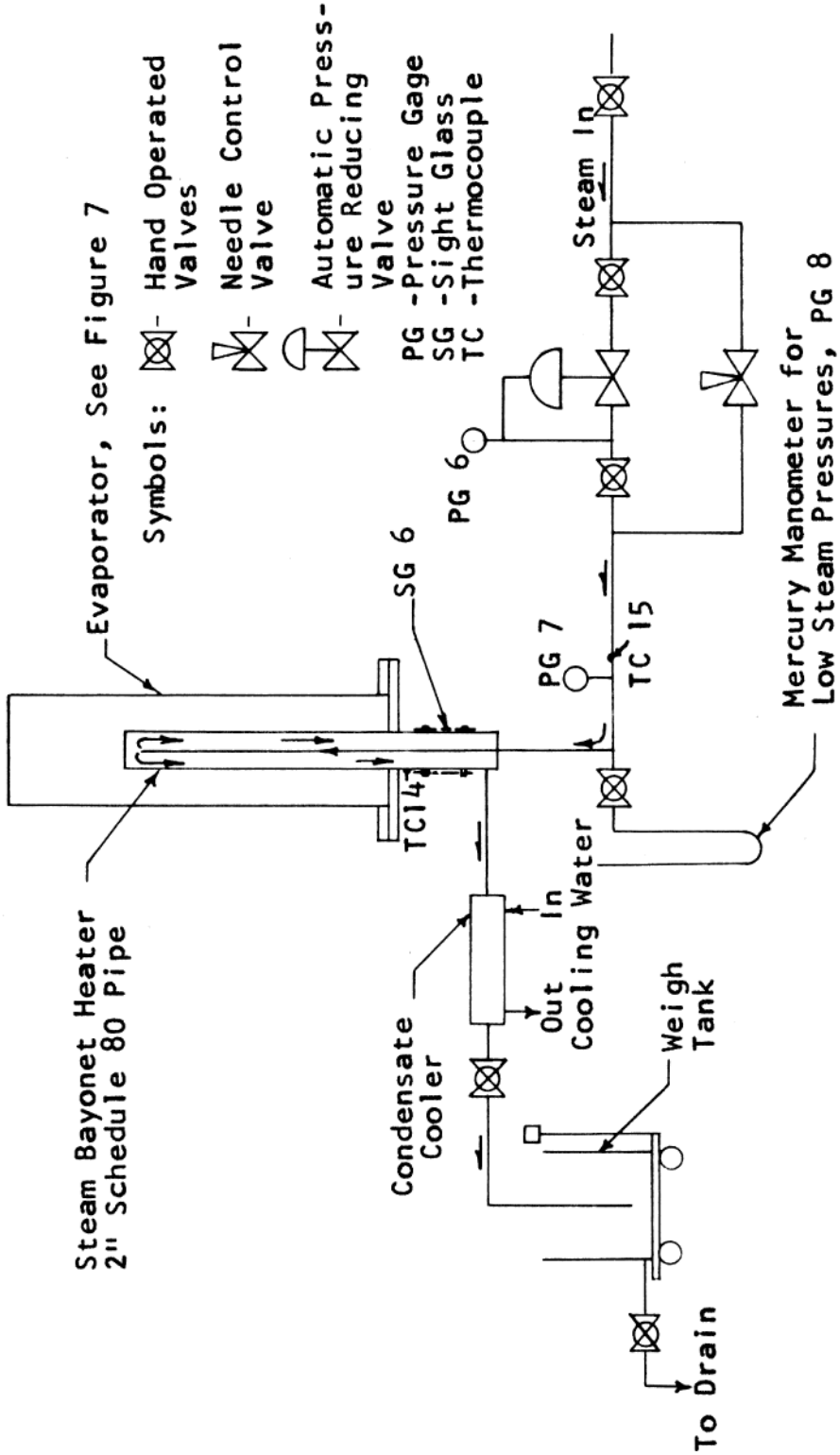


Figure 10. Schematic Diagram of the Steam Supply System for the Sulfur Dioxide Evaporator

The steam supply piping was 3/4 inch, schedule 40 pipe and the condensate piping was 1/2 inch, schedule 40 pipe. These sizes were more than adequate to handle the low flows of 80 pounds per hour and less.

The steam bayonet heater was designed to resemble the industrial heater, although the steam was handled differently as explained previously. The condition of counterflow was maintained, however, since this was considered important. The sight glass, SG 6, was provided at the bottom of the heater to allow visual observation of the condensate level inside of the heater. This also helped to verify that all of the steam was being condensed, see Figure 7.

The condensate cooler was installed to cool the condensate, and thus minimize flashing when it was expanded to atmospheric pressure if such a condition was present. This was a simple concentric pipe heat exchanger with the smaller pipe jacketed by the cooling water. It was found, however, that this unit was unnecessary as the condensate had been sufficiently cooled and did not vaporize when expanded.

REFRIGERATION SYSTEM

The refrigeration system was necessary to condense

the vapor generated by the SO_2 evaporator. A schematic arrangement of this system is shown in Figure 11. The primary component of the system was the finned tube heat exchanger previously described in the section on the SO_2 system. The compressor, condenser, receiver and auxiliary refrigerating equipment was existing laboratory equipment. There were several modifications necessary to develop a working system.

The installation of a 5/8 inch copper tube from the receiver to the F-12 evaporator (SO_2 condenser) was necessary to supply liquid refrigerant. A two inch steel pipe from the F-12 evaporator to the suction of the compressor was installed to return the vapor. The design conditions for the piping were as follows:

F-12 liquid: turbulent flow (Reynolds number > 20,000)
843 pounds per hour
90 psig and 84°F
5/8 inch tubing
equivalent length of 40 feet
clean copper pipe

Pipe Flow Chart Reference 18: Pressure drop = 0.015 psi per ft.

Total pressure drop = 0.015 x 40 = 0.6 psi which is acceptable.

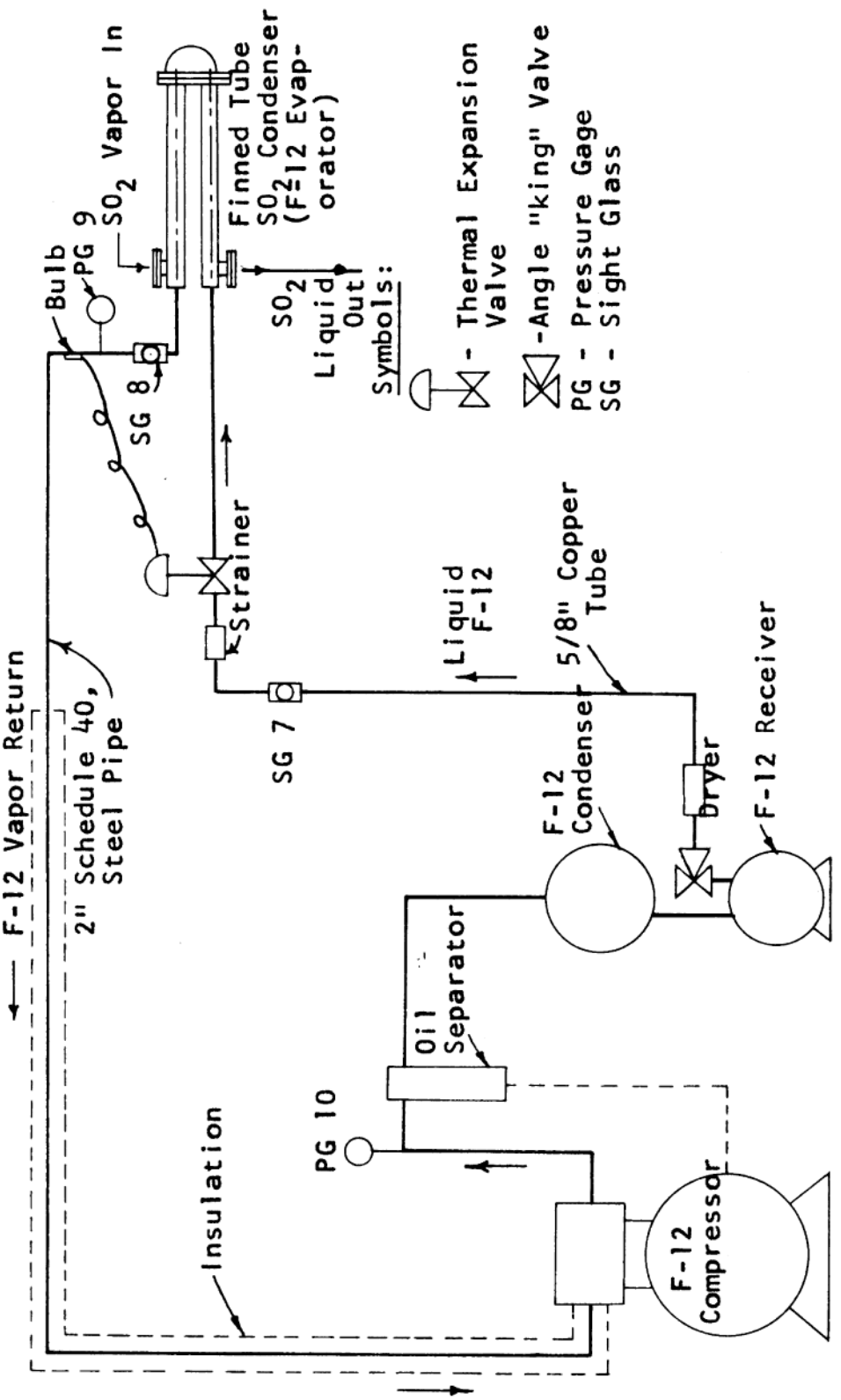


Figure 11. Schematic Diagram of the Refrigeration System for the Sulfur Dioxide Evaporator

F-12 vapor: turbulent flow
843 pounds per hour
10 psig and 2°F
2 inch, schedule 40 pipe
equivalent length of 50 feet

Reference 18: Pressure drop = 0.004/1.72 psi per foot

Total pressure drop = 0.004/1.72 x 50 = 0.116 psi, which is acceptable.

An automatic expansion valve made by the Alco Valve Company was used in the liquid supply line. The valve was type TCL 600 F, thermo expansion, to handle 843 pounds per hour at a pressure reduction of 90 to 10 psig. The connections were soldered with a 5/8 inch inlet and a 7/8 inch outlet. The power assembly number was XB1019F1B. This valve was controlled by the change of volume of a fluid inside a sealed bulb and tube. A change in temperature caused the fluid volume to change and the valve diaphragm would then move the valve stem.

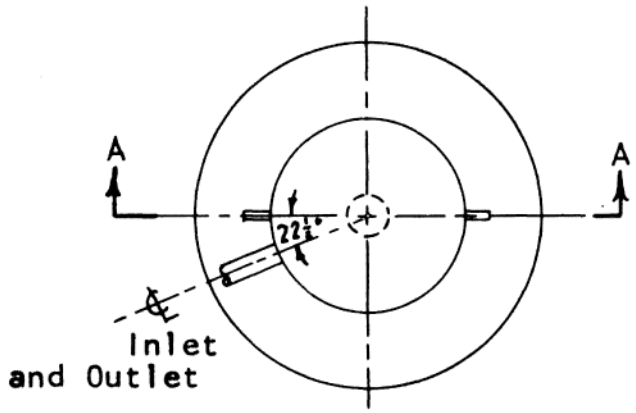
A sight glass, SG 7, was located upstream of the F-12 evaporator and another sight glass, SG 8, located downstream. These glasses allowed a visual inspection of the condition of the refrigerant.

The insulation used on the vapor line was two inches thick, 85 per cent magnesia with polyethylene film for

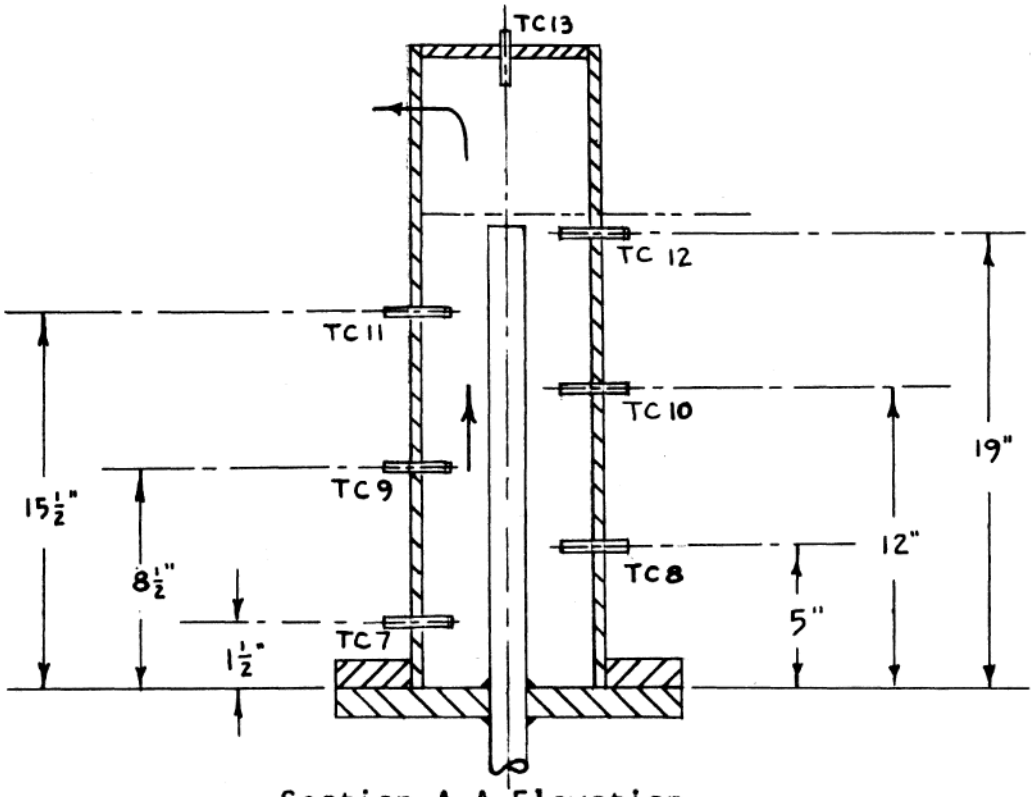
a vapor seal. The insulation covered approximately 60 per cent of the vapor line. Since there was a certain amount of liquid carry-over from the F-12 evaporator it was considered prudent not to insulate the entire vapor line. This omission served as protection for the compressor in this case.

INSTRUMENTATION

The most important instruments in this investigation were the thermocouples positioned to measure the SO_2 liquid temperatures in the evaporator and the surface temperatures on the steam bayonet heater. Any inaccuracies in these readings would have caused the overall results to be in error. The thermocouple arrangement for the SO_2 liquid temperatures is shown in Figure 12 and the thermocouple arrangement for the surface of the steam heater is shown in Figure 13. The thermocouple measuring junctions on the heater surface were made by brazing the copper and constantan to the steel surface of the heater and covering the junction with a small patch of epoxy resin compound to minimize the fin effect of such an attachment. The thermocouple measuring junctions for the liquid temperatures were installed as shown in Figure 14. The heater was positioned in the evaporator



Plan View



Section A-A Elevation

Figure 12. Thermocouple Arrangement for Obtaining the Average Sulfur Dioxide Liquid Temperature In the Evaporator

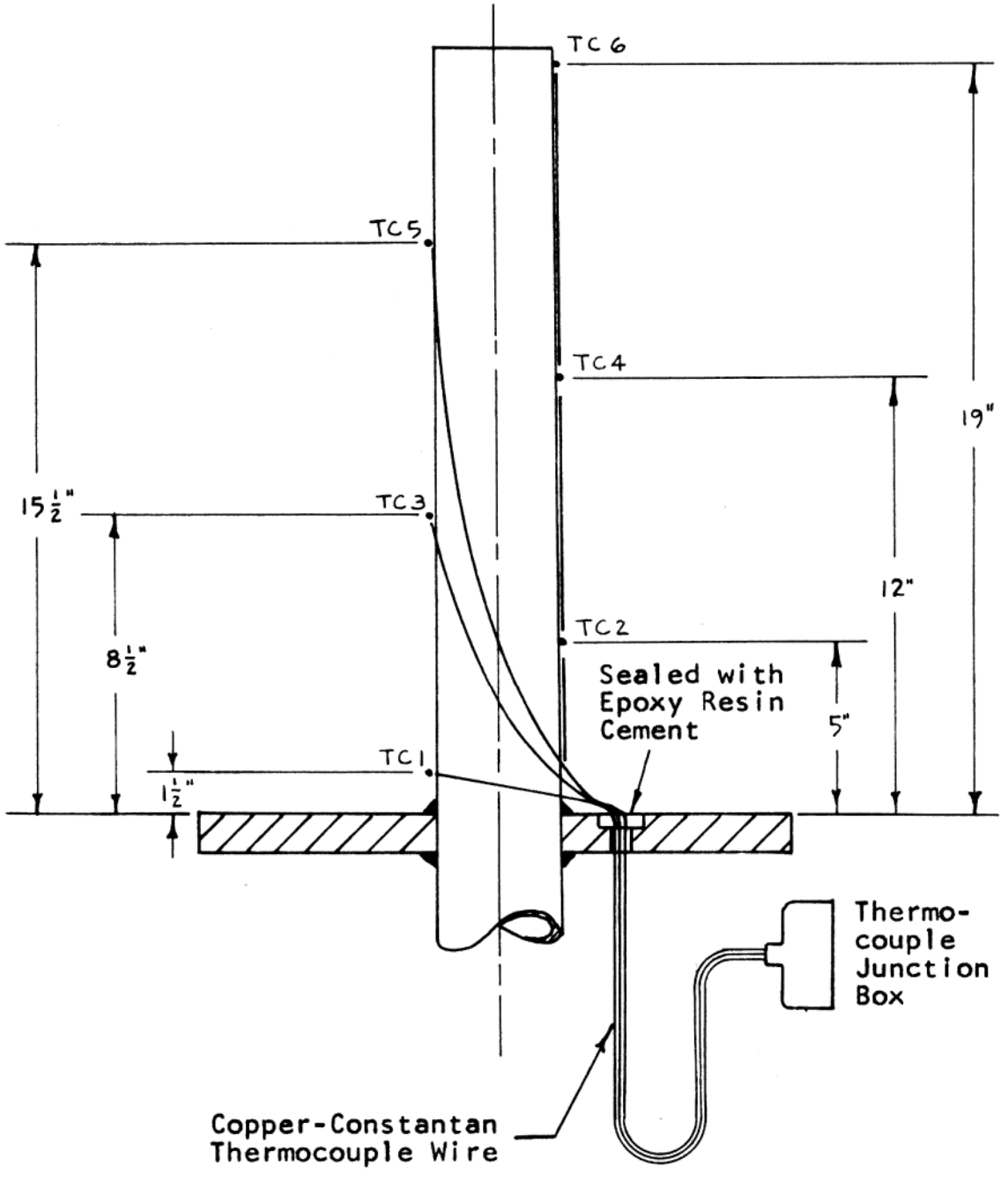


Figure 13. Thermocouple Arrangement for Obtaining The Average Surface Temperature of the Heater

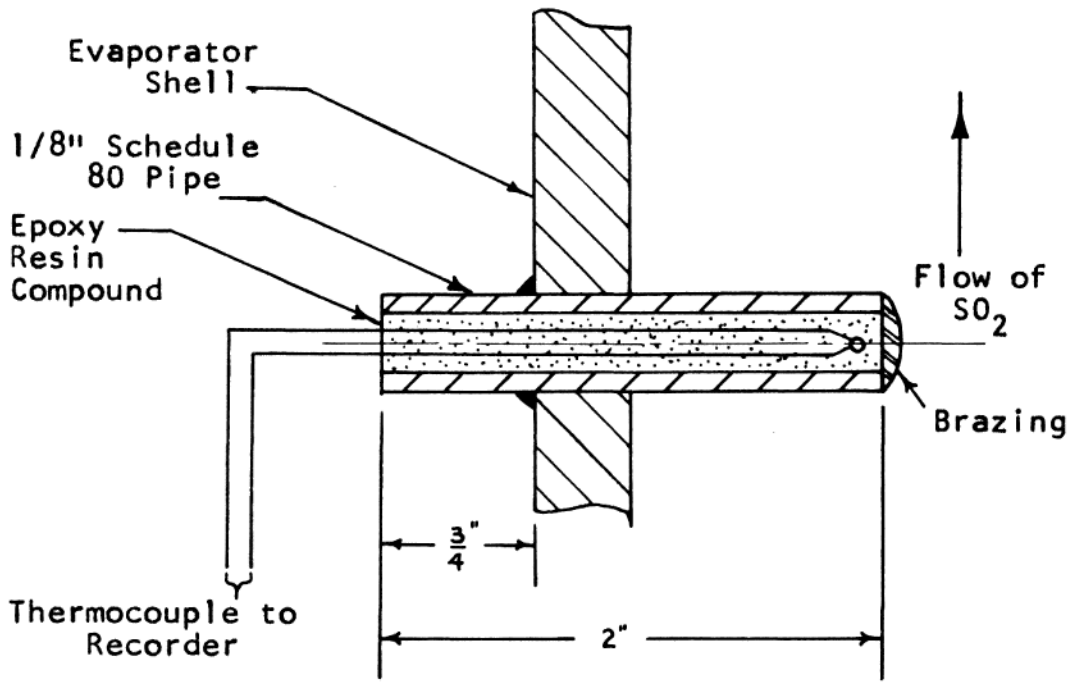


Figure 14. Detail of Thermocouple Well for Obtaining Sulfur Dioxide Liquid Temperatures

so that the thermocouples on the heater surface (TC 1-TC 6) were opposite the thermocouple wells in the evaporator wall (TC 7-TC 12). This arrangement would have allowed calculation of local boiling coefficients if it had become necessary. Thermocouples were also used to record the temperatures at the following locations: SO_2 vapor at the top of the evaporator (TC 13); condensate leaving the steam heater (TC 14); steam entering the steam heater (TC 15); SO_2 vapor at the rotameter (TC 16).

These sixteen temperatures were recorded by a Minneapolis Honeywell recording potentiometer. Model number 153X62-V16-11-111-23, "Electronik", range 1 to 5 millivolts, subdivisions 0.02 millivolts, and serial number 5451.

The pressures were indicated at various positions in the system by nine bourdon type pressure gages and a mercury manometer. These locations are illustrated on Figures 8, 9, and 11. The specifications for the gages are as follows:

PG 1 - Ashcroft; 0 to 160 psi; subdivisions of one psi; 4-1/2 inch diameter; located at the inlet of the SO_2 evaporator.

PG 2 - Ashcroft; 0 to 300 psi, subdivisions of one psi; 4-1/2 inch diameter; located on the top of the SO_2 evaporator.

PG 3 - Ashcroft; 0 to 100 psi; subdivisions of one psi; 3-1/2 inch diameter; located above the rotameter.

PG 4 - Ashcroft; 30 inches of mercury vacuum to 100 psi; subdivisions of one inch of mercury and one psi; 5 inch diameter; located on the east drum.

PG 5 - U. S. Gauge; 30 inches of mercury vacuum to 150 psi; subdivisions of 5 inches of mercury and 5 psi; 4-1/2 inch diameter; located on the west drum.

PG 6 - Ashcroft; 0 to 160 psi; subdivisions of 2 psi; 2-1/2 inch diameter; located at the steam pressure reducing valve.

PG 7 - Merlam; 0 to 31 inches of mercury; subdivisions of 0.1 inch of mercury; located in the steam line at the inlet of the steam heater (for low pressures).

PG 8 - Ashcroft; 0 to 160 psi; subdivisions of 2 psi; 2-1/2 inch diameter; located in the steam line at the inlet of the steam heater for high pressures.

PG 9 - Ashcroft; 30 inches of mercury vacuum to

150 psi; subdivisions of 5 inches of mercury and 5 psi; 4-1/2 inch diameter; located in the F-12 suction line.

PG 10 - Marshalltown; 30 inches of mercury vacuum to 300 psi; subdivisions of 10 inches of mercury and 10 psi; 4-1/2 inch diameter; located in the F-12 compressor discharge line.

The rotameter, made by the Fischer and Porter Company, was graduated from 8 to 100 per cent with 485 pounds per hour of SO_2 at the maximum flow. The meter calibration was for a gas specific gravity of 3.81. A Fischer and Porter correction curve[20] was used to determine the flow whenever the measured gas specific gravity differed from 3.81. The tube number was FP 1-1/2 - 27 - G - 10/80, the serial number was 6010A3776B2, and the meter had 1-1/2 inch standard pipe screwed connections. The meter indicated an instantaneous reading of the flow of SO_2 vapor.

The steam flow was determined by weighing the condensate collected in a tank for a particular period of time. This information was used to determine the heat loss by the steam. These data were obtained for comparison with the heat gained by the SO_2 and discussed later in this thesis. The scales used were made by the

Howe Company, Model number 22, serial number 1558185; 2000 pound capacity, subdivisions of one-half pound.

CALIBRATION OF INSTRUMENTS

The rotameter correction curves supplied by the manufacturer were verified by condensing SO_2 for three, one hour tests and measuring the volume of liquid collected in one of the drums for each test. Each test was made at a different flow. During each of these tests the rotameter was held at a nearly constant value of per cent, pressure, and temperature. In order to determine the volume of liquid SO_2 collected a drawing was made of each drum and the change in the liquid levels observed for each test was sketched on the correct drum drawing. The volume was then determined by measurement from these drawings. Using the drum pressure as the saturation pressure, the total weight of sulfur dioxide collected was determined. Since each test was for a one hour period, the flow was the total weight per hour. A typical calculation of one of these points was as follows:

Drum measurements: 3-5/8" = height increase
65" = average length
23-1/8" = average width

$$\text{Flow of } SO_2 = \frac{\text{Volume collected}}{\text{Specific volume}}$$

$$\text{Flow of } SO_2 = \frac{3.625 \times 65 \times 23.125 \text{ in}^3}{1728 \times 0.01188 \text{ in}^3/\text{lb}}$$

Flow of SO_2 = 267 pounds per hour

Specific volume obtained from Reference 19.

$$\text{Flow' of } SO_2 = \frac{\text{100\% rated flow} \times \text{per cent}}{\text{correction factor}}$$

$$\text{Flow' of } SO_2 = \frac{485 \times 0.363}{6.45} = 273 \text{ pounds per hour}$$

Where the correction factor was obtained from Reference 20.

These values are within 2-1/4% of each other and considered to be of acceptable accuracy.

The other two tests indicated similar results. At a rotameter indicated flow of 340 pound per hour the gravimetric measurement was calculated as 344 pounds per hour. At a rotameter indicated flow of 317 pounds per hour the gravimetric measurement was calculated as 304 pounds per hour. Because of the greater chance of error in measuring a liquid level change, the rotameter reading was regarded as the more accurate of the two methods.

All of the eleven pressure measuring devices were calibrated by using a hydraulic dead weight tester. This compared a known weight acting on one square inch with the pressure gage readings of the gage being calibrated.

All of the calibration readings were within one pound per square inch, therefore, it was considered unnecessary to correct individual readings prior to the calculations.

The calibration of the thermocouples and the potentiometer was made by using a circuit identical to a testing circuit. The measuring junction was immersed in a water bath and the water temperature was increased by adding hot water. The water bath was within a vacuum bottle and each of the test temperatures was therefore relatively constant. The standard of comparison was a precision mercury-in-glass thermometer which was read to one-tenth of a degree Fahrenheit. A series of seven readings was taken and a calibration curve plotted, see Figure 15. It was observed that the deviation along the entire curve was a constant $4^{o}F$. This correction was applied to each temperature that was recorded.

EXPERIMENTAL PROCEDURE

The basic steps employed in the experimental procedure were: start-up of the refrigeration system; start-up of the steam heating system; start-up of the SO_2 system; stabilization of the entire system; collection of the experimental data; and shutdown of the entire system. The detailed procedure for each phase of the operation is given below.

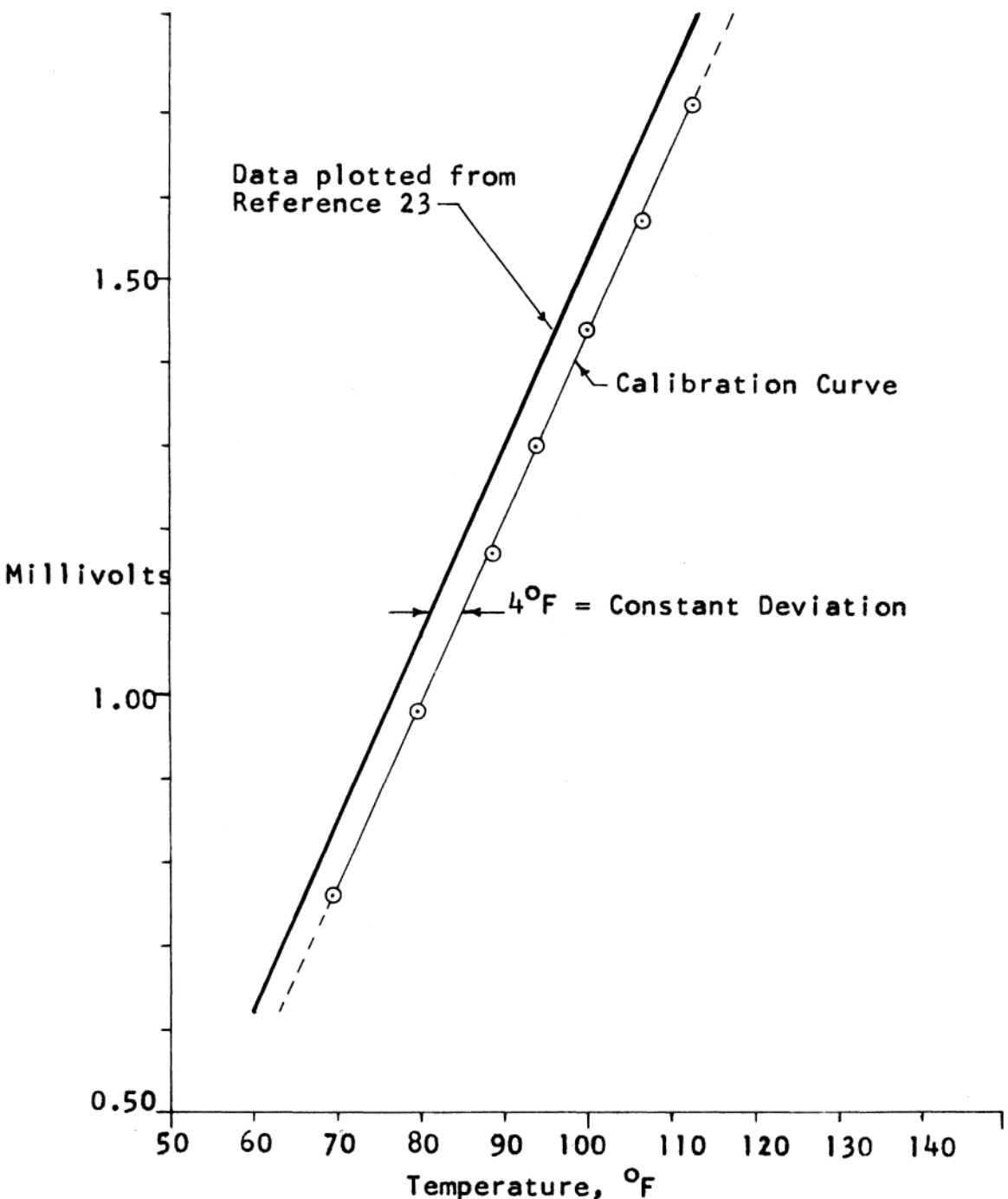


Figure 15. Calibration Curve for Copper-Constantan Thermo-Couples and Minneapolis-Honeywell Temperature Recorder

Start-up of the refrigeration system involved the following steps: (see Figure 11)

(a) The centrifugal water pump was started in the laboratory to provide cooling water for the F-12 condenser and water for the compressor shaft seals. All of the valves in these water lines were opened at this time.

(b) All of the valves in the discharge line of the compressor to the condenser were opened.

(c) The oil level in the compressor was checked.

(d) The F-12 level in the receiver was checked.

(e) The compressor was started by manually closing the switch in the electric motor starting box.

(f) The valves between the condenser and the receiver were opened.

(g) The F-12 liquid outlet valve on the receiver was gradually opened as the load was applied by the condensing SO_2.

(h) The suction and discharge pressures and the liquid and vapor sight glasses were checked to verify that the system was operating properly.

The start-up of the steam heating system involved the following steps: (refer to Figure 10)

(a) Prior to the admission of any steam the SO_2

valves were opened between the evaporator and the condenser. This prevented an excessive pressure build-up in the evaporator as a result of heating any liquid SO_2 present.

(b) All of the valves in the steam line were opened to clear out the condensate.

(c) After the lines had been blown clear the needle valve was closed and the automatic control valve regulated for the required steam pressure. When a steam pressure of less than approximately five pounds per square inch was required the automatic control valve was cut off and the needle control valve was used as the pressure regulator.. Since the needle valve was manually controlled it required much closer attention by the operator. At the lower steam pressures a mercury manometer was used to indicate the pressures instead of the Bourdon-type pressure gages.

(d) After the SO_2 evaporating load was applied, the sight glass below the evaporator was observed to assure that the steam condensate level was visible.

(e) The discharge valve on the weigh drum was opened and the drum emptied prior to each test.

This valve was then closed and an initial reading of the scales was recorded at the beginning of a test. At the end of a test another scale reading was taken. The difference between these readings and the length of time of the test was used to determine the flow of steam in pounds per hour.

The start-up of the SO_2 system was accomplished as follows: (Refer to Figures 8 and 9)

(a) Prior to this start-up the sight glasses for each drum were observed and the drum with the highest liquid level was selected to be the supply drum and the other drum the receiver.

(b) The drum selected as the supply drum had a gas fired heater placed under it. The drum pressure was increased by this flame until it was approximately 55 psig and was held at this pressure throughout all of the testing. The reason for the selection of this particular pressure was to allow the system to be operated at any time at room temperature or above. The saturation temperature for SO_2 at 55 psig is $88^{o}F$.

(c) The valve between the condenser and the re-

ceiver drum and the valve in the equalizer line to the receiver drum was opened. As mentioned in the steam start-up all valves in the SO_2 line between the evaporator and the condenser were open.

(d) The automatic pressure reducing valve was regulated to give the desired flow. This regulation was balanced with the steam flow regulation. As an aid in adjusting this automatic valve either the globe valve upstream or the globe valve downstream of it was first regulated to the required pressure.

(e) The SO_2 liquid cut-off valve on the supply drum was fully opened to permit control of the flow by a manually regulated globe valve, also in this line. This valve was opened very slowly to permit the gradual build-up of a liquid level in the evaporator. The liquid level in the evaporator was kept at a minimum of 19 inches and a maximum of 24 inches above the liquid inlet. Any lower level would have uncovered the steam heater and any higher level would have resulted in some liquid carry-over into the vapor discharge pipe, which could be visually observed in the sight glass

located there. The regulating valve required nearly the constant attention of the operator to assure the proper liquid level.

(f) Occasionally the liquid level became too high and there was some liquid carried into the vapor pipe invalidating that particular test. This high level was quickly reduced by closing the valve in the vapor line nearest the evaporator and allowing the evaporator pressure to be built up by the steam heater until it exceeded the pressure in the supply drum. When this occurred the liquid was pushed back into the supply drum. When the liquid level was restored to its proper position the valve in the vapor line was opened and the evaporator pressure was reduced to its normal operating range. Great care had to be exercised when performing this operation because excessive pressure in the evaporator could have weakened or blown out the epoxy cement plug surrounding the surface thermocouples, see Figure 13. The maximum pressure obtained in the evaporator was 140 psig. This pressure was held for only a few minutes for test purposes. Usually, the

pressure was kept below 100 psig.

Stabilization of the entire system required approximately an hour. The system was then at a condition of thermal equilibrium. This state of balance was obtained by maintaining a constant liquid level in the evaporator, a constant supply of steam, and a constant flow of F-12 refrigerant, and thereby a constant flow of SO_2. To avoid a constant adjusting of steam flow, refrigerant flow, and SO_2 liquid and vapor flows, the steam pressure was set, the refrigerant flow was set, and the sulfur dioxide vapor flow was set. Therefore, the system control passed to a single valve which controlled the SO_2 liquid entering the evaporator. This initial stabilizing proved to be the most difficult of all operational tasks. However, once the system was stable it tended to remain in that condition for relatively long periods of time. For three tests this condition was held for over an hour. The decision to operate the nichrome heater for each test was based on the temperature of the SO_2 liquid entering the evaporator. If the liquid was subcooled at the inlet pressure the nichrome heater was cut on to bring the liquid to a saturated condition. It was important not to have this heater contribute to the boiling however. The sight glass, SG 1, in the pipeline verified that the

liquid was not boiling as it entered the evaporator. After the experimental data was recorded the temperature difference between the boiling SO_2 liquid and the surface of the heater was changed, thereby changing the flow. This change was initiated by changing the steam flow and adjusting the automatic pressure reducing valve in the SO_2 system. The time required to stabilize the entire system under these new conditions required approximately one-half an hour.

Collection of the experimental data was begun after the system was stabilized at a particular temperature difference between the boiling liquid and the heater surface. The recording potentiometer had been previously warmed up for approximately one-half an hour and the mechanism operated for about five minutes prior to the beginning of the test. The first three tests required one hour each and some of the data collected was used to verify the rotameter readings. Fourteen additional tests required ten minutes each with an intervening time of approximately one-half an hour to re-stabilize the system. The experimental data was recorded in the section of this thesis entitled Data and Results.

The shut-down of the system required the following steps:

(a) The SO_2 liquid valve between the supply drum and the evaporator was closed. The liquid in the evaporator was allowed to completely boil out and be condensed.

(b) The steam supply valve was closed.

(c) The flame under the supply drum was extinguished.

(d) The nichrome heater was turned off.

(e) The outlet valve on the F-12 receiver was closed and the refrigeration system allowed to "pump down" for approximately fifteen minutes.

(f) The compressor was stopped by opening the switch in the electric motor starting box.

(g) All valves in the refrigeration system were closed.

(h) All valves in the SO_2 system were closed.

(i) The potentiometer was turned off.

(j) The cooling water for the F-12 condenser was turned off.

(k) The system was inspected for any leakage and repaired as necessary.

SAMPLE CALCULATIONS

Test 7 was selected as typical for all of the calculations performed to obtain the results.

Rotameter: Read 43% at 49.3 psia and 78°F;

$$\text{Specific gravity} = \frac{\text{specific volume of air at standard conditions}}{\text{specific volume of } SO_2 \text{ vapor at test conditions}}$$

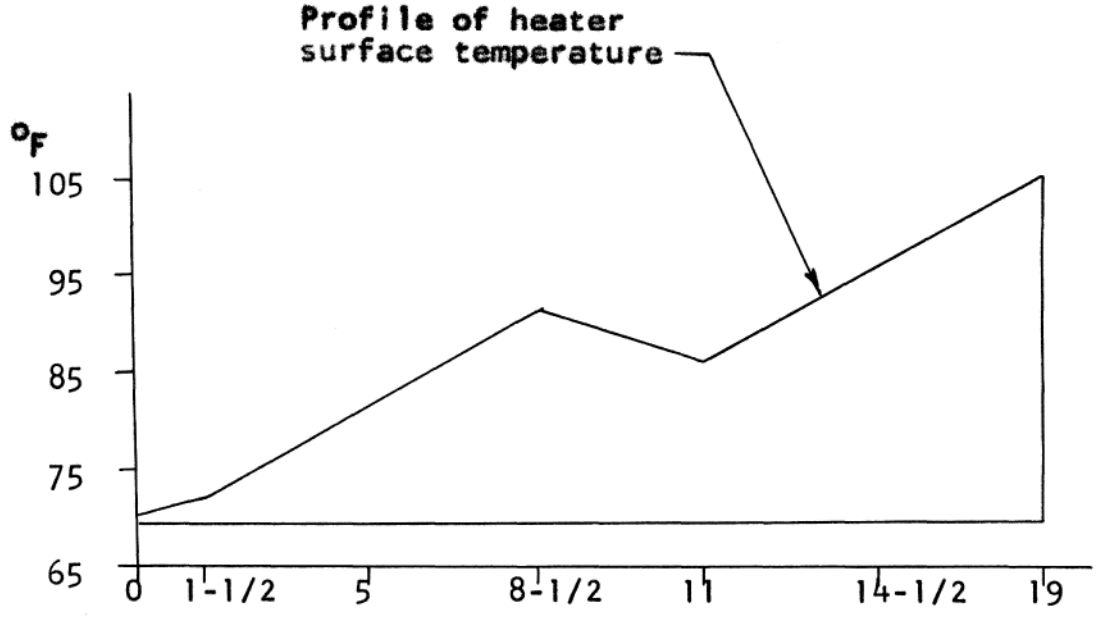
$$\text{Specific gravity} = \frac{13.33 \text{ ft}^3/\text{lb}}{1.69 \text{ ft}^3/\text{lb}} = 7.88$$

Correction factor = 0.70; from chart at a specific gravity of 7.88; Reference 20

$$\text{Flow of } SO_2 = \frac{\text{Rated flow} \times \text{per cent read}}{\text{Correction factor}}$$

$$\text{Flow of } SO_2 = \frac{485 \times 0.43}{0.70} = 298 \text{ pounds per hour}$$

Temperature Distribution:



Scale of temperature distribution plot: 0.0625 in^2 = 5 in -^oF.

$$\text{Average } \Delta T = \frac{\text{Area between temperature profiles} \times \text{scale factor}}{\text{length of heater}}$$

$$\text{Average } \Delta T = \frac{64.5 \times 5}{19} = 17^oF$$

Heat gained by SO_2, ΔQ_{so_2} = flow x latent heat at test conditions

where latent heat = 155.0 BTU/lb from Reference 19

ΔQ_{so_2} = 298 x 155.0 = 46,200 BTU/hr

Using the general convection equation: $\Delta Q = h A \Delta T$, and solving for $h = \frac{\Delta Q so_2}{A \Delta T} = \frac{46,200}{1.0035 \times 17} = 2710$

where h = boiling film coefficient, BTU/hr-ft^2-^oF

A = surface area exposed to boiling, ft^2

ΔT = average surface-liquid temperature difference between boiling SO_2 liquid and the heater surface, ^oF

ΔQ_{so_2} = heat gained by the boiling liquid SO_2, BTU/hr

Verification of ΔQ_{so_2}:

ΔQ_{so_2} = flow x difference of enthalpy between liquid into evaporator and vapor at rotameter

$$\Delta Q_{SO_2} = 298 \times (212.3 - 56.3) = 46,500 \text{ BTU/hr}$$

Therefore, $\Delta Q_{SO_2} = 46,200$ BTU/hr was used as the most accurate value.

Calculation of overall heat transfer coefficient, U, BTU/hr-ft^2-^{o}F by Ringler's method:

$$U = \frac{\Delta Q_{SO_2}}{A\ \Delta T'} = \frac{46,200}{1.0035 \times 125} = 370 \text{ BTU/hr-ft}^2\text{-}^o\text{F}$$

where $\Delta T'$ = temperature difference between saturated SO_2 in evaporator and inlet steam temperature at saturation conditions

The recorded data used for these calculations are presented in Table 1 and the tabulated experimental results are presented in Table 2.

DATA AND RESULTS

Table 1. Recorded Data

Test No.	Date	Heater Surface Temperatures, °F						SO_2 Liquid Temperatures, °F						
		TC1	TC2	TC3	TC4	TC5	TC6	TC7	TC8	TC9	TC10	TC11	TC12	TC13
1	4/3	119	124	140	143	138	193	81	82	82	83	83	83	83
2	4/5	125	127	147	148	147	200	87	85	86	86	88	88	88
3	4/5	123	128	148	152	158	206	86	86	86	86	88	88	88
4	5/20	108	104	114	106	113	118	84	84	84	84	84	84	84
5	5/20	96	98	109	100	105	111	82	82	82	82	82	82	82
6	5/20	81	92	101	94	100	108	78	78	78	78	78	78	78
7	5/20	76	85	96	90	100	109	73	73	73	73	73	73	73
8	5/20	83	95	103	96	105	116	78	78	78	78	78	78	78
9	5/20	84	101	108	102	106	121	83	83	83	83	83	83	83
10	5/20	77	86	94	90	98	110	74	74	74	74	74	74	74
11	5/20	82	88	95	98	104	118	80	80	80	80	80	80	80
12	6/18	122	125	129	120	140	144	102	102	102	102	102	102	102
13	6/18	108	120	128	117	146	156	96	96	96	96	96	96	96
14	6/18	134	136	140	134	165	168	118	118	118	118	118	118	118
15	6/18	140	142	147	142	175	181	118	118	118	118	118	118	118
16	6/20	103	108	110	106	116	126	86	86	86	86	86	86	86
17	6/20	105	113	113	110	120	127	87	87	87	87	87	87	87

Table 1. Recorded Data (continued)

Test No.	Rotameter			Evaporator		Barometric Pressure	Room Temperature	Steam Heater		Steam Flow	Steam Pressure
	PG3	TC16	%	PG1	PG2			TC14	TC15		
1	55.8	79	46	62.4	62.4	13.97	72.0	-	-	58	29.0
2	61.9	83	42	67.1	67.1	13.97	71.5	204	240	59	28.9
3	61.5	82	36	67.0	66.6	13.97	71.5	143	235	48	23.9
4	54.8	80	49	61.8	61.8	13.80	81.0	180	214	81	19.8
5	55.8	82	47	60.8	60.8	13.80	81.0	154	200	63	18.0
6	53.3	81	45	56.8	56.8	13.80	81.0	105	201	56	17.5
7	49.3	78	43	50.8	50.8	13.80	81.0	96	198	34	17.0
8	46.8	72	41	53.3	53.3	13.80	81.0	110	198	39	16.5
9	56.8	81	37	60.3	60.3	13.80	81.0	108	197	54	16.0
10	54.8	76	39	53.8	53.8	13.80	81.0	104	198	51	15.5
11	56.8	75	35	59.3	59.3	13.80	81.0	119	200	36	15.0
12	61.3	93	41	84.3	84.3	13.78	82.0	176	209	60	15.2
13	58.8	92	40	71.8	71.8	13.78	82.0	118	208	54	14.8
14	61.8	94	41	101.8	101.8	13.78	82.0	176	212	60	15.9
15	64.8	100	41	106.8	106.8	13.78	82.0	172	210	65	17.7
16	55.8	81	48	68.8	68.8	13.80	78.5	167	208	54	14.9
17	56.8	81	46	68.8	68.8	13.80	78.5	169	212	52	15.9

Note: For locations of TC14 and TC15, see Figure 10.

All pressures are absolute, all temperatures are ^{o}F, and the flow is in pounds per hour.

Table 2. Experimental Results

Test No.	Rotameter			Average ΔT °F	Area of Heater ft^2	Film Coefficient, h BTU/hr-ft^2-°F	Heat Flux, Q/A BTU/hr-ft^2	ΔQ_{SO_2} BTU/hr
	Gas S.G.	meter correction	flow lbs/hr					
1	8.95	0.665	339	49.5	0.932	1128	55,800	52,000
2	9.62	0.640	317	51.7	0.932	1000	51,800	48,300
3	9.33	0.645	273	61.0	0.932	735	44,900	41,800
4	8.45	0.680	349	25.1	1.0035	2110	53,000	53,000
5	8.50	0.680	335	20.0	1.0035	2550	51,000	51,000
6	8.33	0.685	319	15.7	1.0035	3130	48,800	48,800
7	7.88	0.700	298	17.0	1.0035	2710	46,200	46,200
8	7.33	0.725	274	20.5	1.0035	2050	42,100	42,100
9	8.50	0.680	264	15.8	1.0035	2540	40,200	40,200
10	8.90	0.665	284	19.2	1.0035	2280	43,800	43,800
11	8.61	0.680	246	16.0	1.0035	2350	37,600	37,600
12	9.33	0.655	300	27.4	1.0035	1610	44,200	44,200
13	9.06	0.660	290	33.3	1.0035	1300	43,200	43,200
14	9.33	0.655	306	29.2	1.0035	1505	44,000	44,000
15	9.52	0.640	307	37.3	1.0035	1182	44,100	44,100
16	8.33	0.680	342	21.6	1.0035	2390	51,600	51,600
17	8.30	0.660	352	26.5	1.0035	2000	51,100	51,100

Note: Gas S.G. = gas specific gravity and is based on standard air.

VI. DISCUSSION

The discussion of results of this investigation is presented as outlined below:

Experimental Accuracy

Results of this Investigation

Comparisons with Previous Investigations

Summary

EXPERIMENTAL ACCURACY

The experimental accuracy of the copper-constantan thermocouples combined with the recording potentiometer was checked. After plotting a calibration curve, Figure 15, the accuracy of this unit was considered acceptable. The boiling temperature of SO_2 at the indicated pressure was always at the saturated value according to the thermodynamic properties of SO_2.[19]

Wherever a thermocouple well was used there was an inherent fin effect with an accompanying temperature gradient along the well. This affect was believed to be at a minimum in this test because the thermocouples in these wells continually indicated the saturation temperature of the boiling SO_2.

The experimental accuracy of the heater surface thermocouples was the most difficult to evaluate. These thermocouples were soldered to the steel heater surface

and slightly insulated from the SO_2 liquid with an epoxy resin compound. The fin effect on these thermocouples was believed to be very slight because of their extremely short length. This method of attaching thermocouples was not unique and is generally regarded as an acceptable method of measuring surface temperatures. The remaining instruments were calibrated as explained in the investigation and found to be within acceptable limits.

RESULTS FOR THIS INVESTIGATION

This experiment produced consistent, predictable data of boiling coefficients for various temperature differences, Δt, between the heater surface and the boiling SO_2 liquid. Four tests were conducted to verify the reproducibility of the curve shown on Figure 16 and 17. These values were within ten per cent of the values indicated by the curve which was considered acceptable.

The surface-liquid temperature difference was varied from $15.7^{\circ}F$ to $61.0^{\circ}F$. This represented the complete range of the laboratory experimental system. The evaporator was incapable of attaining a lower Δt because the lowest local Δt was only $2^{\circ}F$ - $3^{\circ}F$, which is considered a minimum for most heat transfer equipment. The highest Δt obtained was limited by the temperature of the available

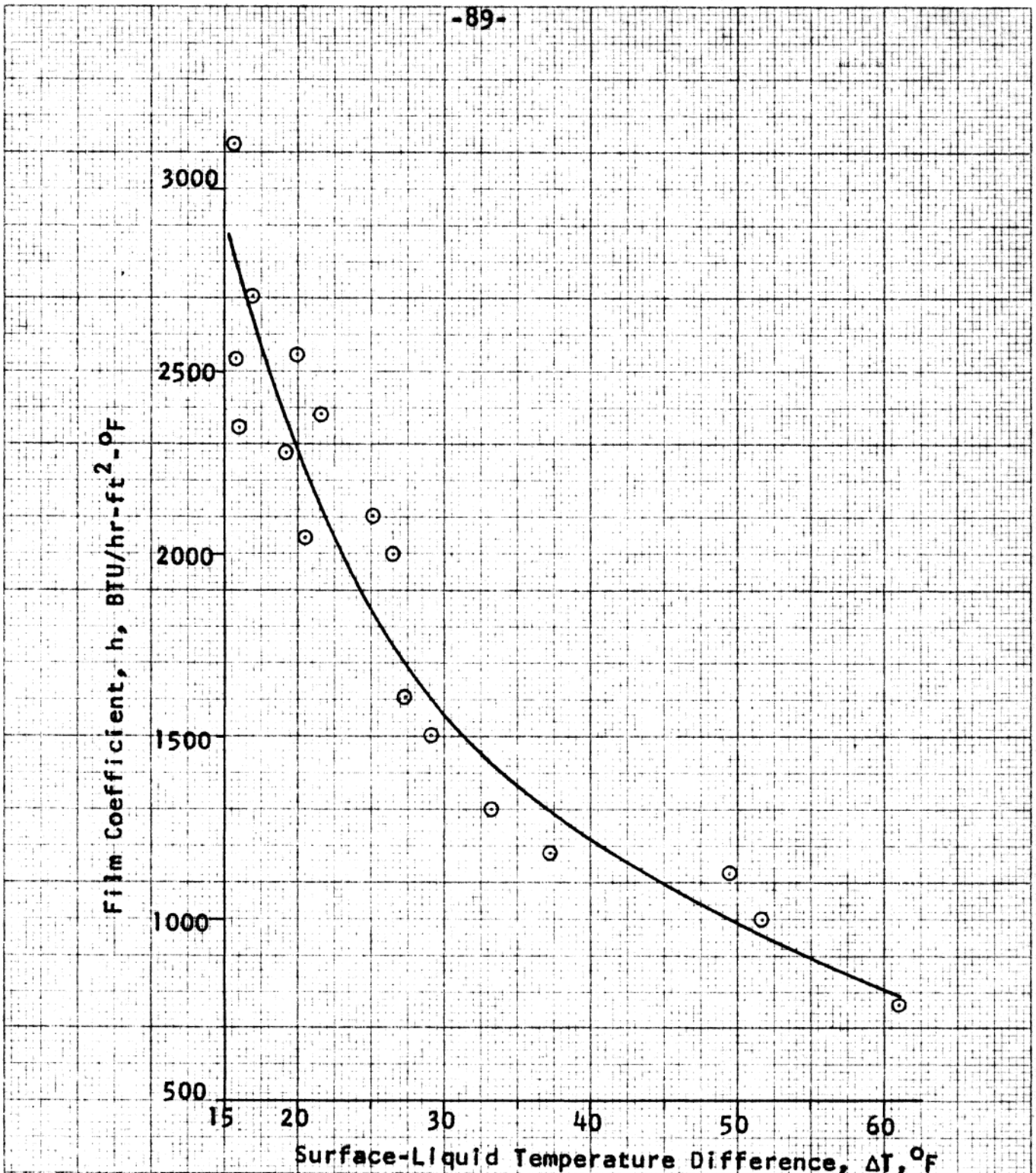


Figure 16. Curve of Boiling Film Coefficients for Sulfur Dioxide

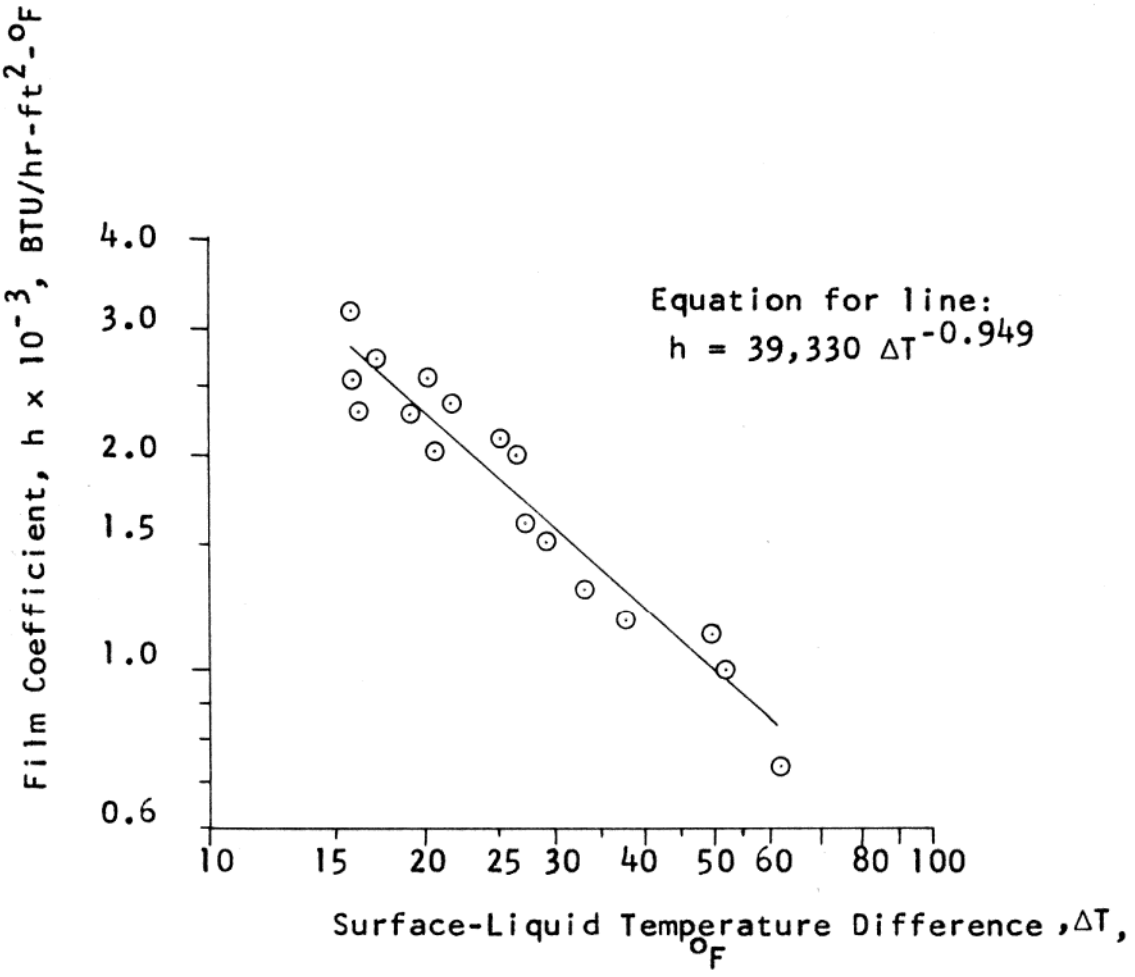


Figure 17. Logarithmic Plot of Boiling Film Coefficients for Sulfur Dioxide

steam supply. This range represented the widest range of Δt for SO_2 investigated by anyone to date. The probable reasons for this previous lack of data was explained in the introduction.

The minimum heat flux obtained was 37,600 BTU/hr-ft^2-$^\circ$F at a surface-liquid temperature difference of 16°F. A maximum heat flux of 55,800 BTU/hr-ft^2 was obtained at a Δt of 49.5°F. It is believed that the limiting factor for the test evaporator was the condensing capacity of the laboratory refrigeration system. This was evidenced by a very gradual rise of the pressure in the receiver drum at the maximum flow rate. However, the laboratory unit was tested to 352 pounds per hour, which was substantially above the 300 pounds per hour rating of the industrial unit. Had the refrigeration capacity been somewhat greater, it was estimated that the test evaporator would have produced 400 pounds per hour of SO_2.

By examining Figures 16 and 17, boiling film coefficient versus surface-liquid temperature difference, it was established that a condition of partial nucleate boiling existed during the experiment. This conclusion was made because of the negative slope of this plot, that is, with an increase in surface-liquid temperature difference there was a corresponding decrease in boiling film coefficient.

Two important factors are assumed to have contributed to the relatively high values of boiling film coefficients: (a) the disturbance of the boundry layer by the rising bubbles on the heater surface, as pointed out by McAdams, and (b) the pressure of over four atmospheres maintained in the evaporator during the tests, as confirmed by Myers and Katz.

By examining Figure 18, heat flux versus surface-liquid temperature difference, it was seen that there was an indication of a peak at a Δt of $50^{\circ}F$ and heat flux of 48,500 BTU/hr-ft^2. This showed that with an additional increase in Δt there would be a decrease in Q/A. This indicated the point of optimum operation of the evaporator with regard to the Δt between the heater surface and the boiling SO_2 liquid.

Initially the heater surface was clean steel pipe. When it was examined after many hours of operation a thin, greenish powder had covered the surface. This powder had the appearance of iron sulfate with a trace of water. If this coating affected the boiling film coefficient evaluation in the laboratory, undoubtedly it would have also affected the field operation of the evaporator. Therefore, the laboratory tests were realistic as they closely approached normal operating conditions of this heat exchanger.

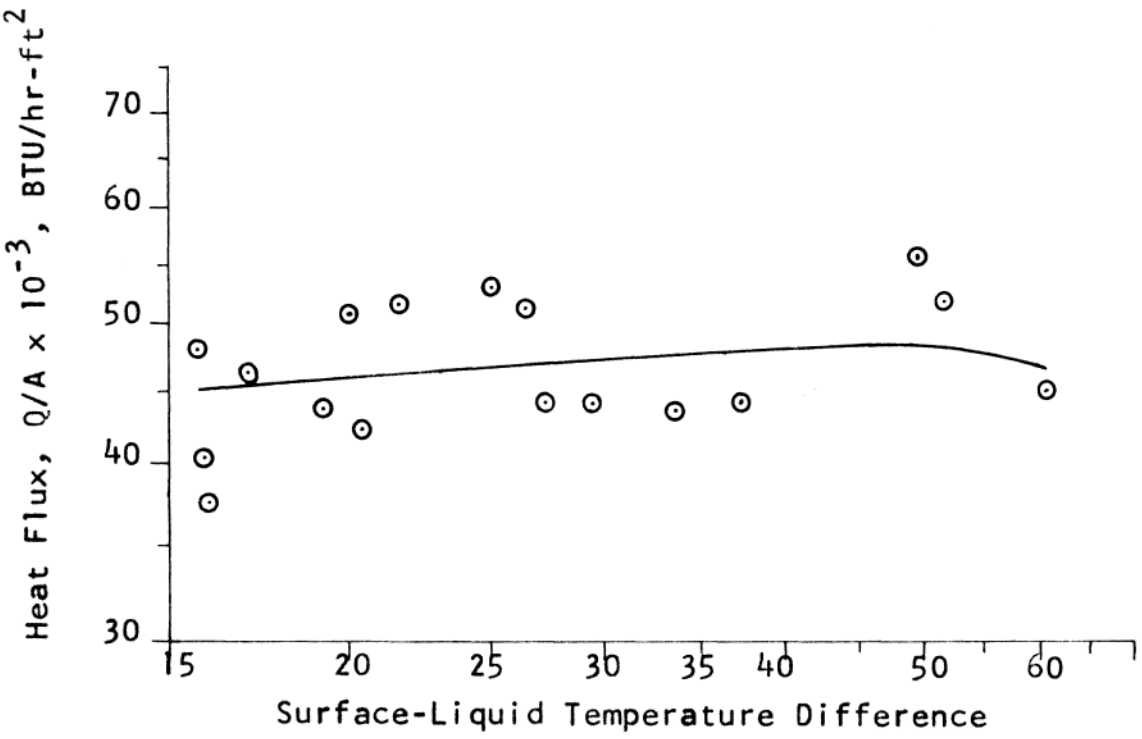


Figure 18. Logarithmic Plot of the Effect of Temperature Difference on Heat Flux for Sulfur Dioxide

It was observed that the SO_2 was in the partial nucleate regime at relatively low values of Δt as compared with water. However, other fluids do boil at low values of Δt as presented by McAdams in his Figure 14-13. It was seen from these curves that n-pentane (90 per cent pure) entered the threshold of film boiling at a Δt as low as $8^{\circ}F$ at a pressure of 415 psia and heat flux of 70,000 BTU/hr-ft^2.

The plot of heat flux versus SO_2 flow was presented as Figure 19. This curve was essentially a straight line. An equation was determined by the method of averages and was found to be : $W = 0.00673\ Q/A - 9.25$ where W = flow of SO_2 in pounds per hour. This curve, as well as, the other curves presented, will be useful to designers of SO_2 evaporators of the type used in this experiment.

COMPARISONS WITH PREVIOUS INVESTIGATORS

Due to several limitations, no comparisons could be made with the investigation of Stewart. The results of SO_2 film boiling coefficients by this author were for relatively low values of heat flux. A maximum heat flux of only 30.6 BTU/hr-ft^2 was obtained and a corresponding maximum film boiling coefficient of only 0.9 BTU/hr-ft^2-$^{\circ}F$. Values obtained in this investigation were much higher.

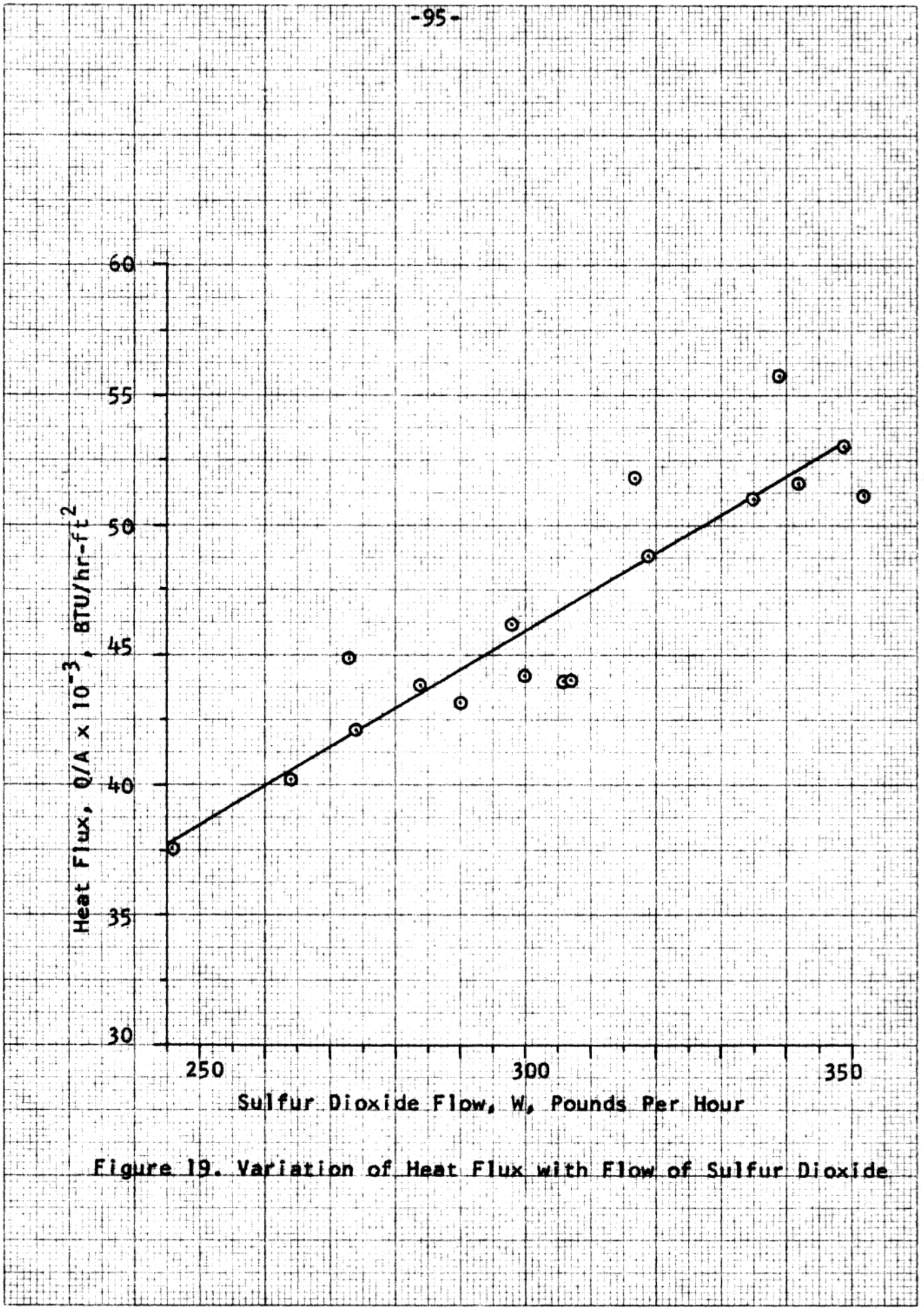


Figure 19. Variation of Heat Flux with Flow of Sulfur Dioxide

Further, it was concluded that the accuracy of Stewart's information was doubtful since some of his conclusions were later proven erroneous by King and Phillip and Tiffany.

King's plot of SO_2 film boiling coefficient versus heat flux resulted in a straight line in spite of some periodic temperature cycling and ebullition which occurred during his tests. King presented his plot as a broken straight line because of this condition. The range of surface-liquid temperature difference investigated was from $4^{\circ}F$ to $10^{\circ}F$. It was interesting to note that when King's straight line was extended to a heat flux of 46,000 BTU/hr-ft^2, the corresponding value of film coefficient was 2700 BTU/hr-ft^2-$^{\circ}F$. This was in close agreement with the value of 2500 BTU/hr-ft^2-$^{\circ}F$ obtained by this investigation at the same heat flux. The slope of the line apparently changes from positive to negative in this range because King's line had a positive slope while the curve in this investigation had a negative slope. This indicated a change from nucleate to partial nucleate boiling at the higher heat fluxes.

Phillips and Tiffany investigated the ebullition of SO_2 but did not present any boiling film coefficients. Their results were particularily valuable for this investi-

gation, however, by pointing out the difficulties encountered if ebullition was allowed to occur and remain unchecked. Great care was taken in this investigation to avoid this condition of superheating of the liquid and thereby produce ebullition. As mentioned previously, a vapor line of high pressure SO_2 was connected to the evaporator for use as an "ebullator" in the event of ebullition of the SO_2 in the evaporator. This SO_2 vapor was frequently used during the warm-up of the apparatus but was needed for only one test. The superheating of the boiling liquid mentioned by these authors did not occur at any other time during this investigation. Little difficulty was encountered with ebullition due to the fact that the degree of superheating of SO_2 liquid varies inversely with the specific volume, as stated by these authors. The specific volume of the liquid in this investigation was approximately ten per cent greater than the specific volumes that were investigated by Phillips and Tiffany.

Stewart and Hechler studied boiling film coefficients in a vertical SO_2 evaporator of considerable height (80 inches). Their results were quite limited in that the highest heat flux attained was only 4000 BTU/hr-ft^2 as compared with 85^{o}F and 55,800 BTU/hr-ft^2 for this investigation. At a given saturation temperature, an increase

in heat flux above 3000 BTU/hr-^{o}F caused an increase in the boiling film coefficient. It was observed, however, that the rate of increase of the film coefficient became slightly less at higher heat fluxes. Apparently, at even higher values of heat flux the curve slope may be reversed. Such was the case in this investigation. The values of heat flux were over ten times greater, the values of film coefficients were three to four times greater and the surface-liquid temperature difference range was wider, 16^{o}F to 61^{o}F compared with Stewart's and Hechler's Δt range of 3^{o}F to approximately 10^{o}F. Since a compressor was used by these investigators, a certain amount of lubricating oil was undoubtedly deposited on the surface of the heating element and this added another, unevaluated, resistance to the transfer of heat. These authors noted that with an increase in saturation temperature the boiling film coefficient will increase for a given heat flux. These factors may explain the difference between the magnitudes of boiling film coefficients in their investigation and this investigation. Because of the large differences in operating conditions between these two investigations a comprehensive comparison of results was not possible.

The work of Myers and Katz was concerned with boiling of SO_2 in the stable nucleate regime. For this reason

there was little opportunity for comparison with this investigation. However, these authors confirmed that boiling film coefficients increase with increased pressure. This may place the boiling film coefficients found by Myers and Katz within the range of those found by this investigation at corresponding values of surface-liquid temperature difference. However, the slope of the boiling coefficients versus surface-liquid temperature difference plot by Myers and Katz was positive while the slope of this plot was negative for this investigation, as previously mentioned. This was the correct relationship between these curves since the former was in the stable nucleate boiling regime and the latter was in the partial nucleate boiling regime. Since a compressor was used to circulate the SO_2 in the Myers and Katz system there undoubtedly was some lubricating oil mixed with the fluid. Although the affect of this oil was not evaluated, it probably affected the film boiling coefficients adversely.

There seemed to be very close agreement between This investigation and Ringler's investigation, although his apparatus was designed for a greater flow of SO_2. It was also the most recent information available regarding SO_2. Values for overall coefficients, U, of heat transfer

were determined by Ringler rather than boiling film coefficients. Since his tests involved an SO_2 evaporator of a similar type as the one tested in this investigation, the results were easily compared. Ringler's method for determination of U values was applied to this investigation and it was found that the range of U was from 480 to 313 BTU/hr-^{o}F. These values decreased slightly as the temperature difference between the steam and the liquid SO_2 was increased. This followed the trend of boiling film coefficients versus surface-liquid temperature difference of this investigation. Ringler's values for U varied from 368 to 256 BTU/hr-ft^2-^{o}F. In light of the higher values of boiling film coefficients found in this investigation, it was concluded that the steam condensate film coefficient must be the controlling factor which established an overall heat transfer coefficient. Referring to Stover's[22] chart for determining values for the steam film coefficients, it was found that for this investigation the variation was from 285 to 446 BTU/hr-ft^2-^{o}F. It can be clearly seen that these lower values for steam film coefficients would be the predominant factors when combined with the heat transfer resistance of the steel wall and the boiling film coefficients. These latter two values would be over 1000 BTU/hr-ft^2-^{o}F. It can be

concluded that designers of SO_2 evaporators could use the values for steam film coefficients less approximately ten per cent to get a reliable value for U. This reduction of ten per cent should compensate for the influence of the other two resistances mentioned previously.

There was a marked similarity of the results of Woods and Bryan, as presented by McAdams, and the results of this investigation. The fluid investigated by these authors was benzene. The shape of the heat flux versus surface-liquid temperature difference curve was nearly identical, the curve was within a similar range of surface-liquid temperature difference and it exhibited a peak as did the curve presented in this thesis as Figure 18. From the reference curve, overall values of coefficients were calculated and found to decrease in a manner similar to the plot of boiling film coefficients versus surface-liquid temperature difference in this thesis, see Figure 17. Although the range of surface-liquid temperature difference was similar the absolute values of heat flux were lower as would be expected for a different fluid. However, the overall coefficients were comparable as the reference values varied from 400 to 300 BTU/hr-ft^2-^{o}F for benzene and this investigation yielded overall coefficients of 467 to 313 BTU/hr-ft^2-^{o}F over the same range

of surface-liquid temperature difference.

SUMMARY

It has been shown that the experimental accuracy was within acceptable limits and that the data presented can be used by others with a reasonable degree of confidence.

The results of this investigation were concerned with boiling of SO_2 liquid in the partial nucleate regime. For this reason, correlation with other investigations was limited. The best agreements were made with the work of Ringler for SO_2 and Woods and Bryan for benzene. The results of this experiment have partially filled a gap left for many years by researchers of SO_2 heat transfer. As stated previously, this gap existed because the applications of SO_2 as a refrigerant in the 1930's required boiling evaluations only in the low temperature ranges.

VII. CONCLUSIONS

The objective of this investigation was accomplished, in that values for boiling film coefficients were determined and a practical method of circulating SO_2 without a compressor was attained.

The boiling film coefficients decreased with an increase in temperature difference between the steam heater surface and the boiling SO_2 liquid. This negative slope of the curve denoted boiling in the partial nucleate regime. In spite of the comparatively little information of boiling in this regime, comparisons were successfully made as discussed in the previous section.

Although the operation of the equipment proved to be very difficult at times, once a stable condition was attained it remained stable and required very little adjustment by the operator.

The most practical conclusion of this work was that the steam film coefficients were the predominant factors in determining the overall film coefficients. An SO_2 evaporator designer need be concerned only with the proper evaluation of a steam coefficient when his heat exchanger employs steam as the heating medium. The curves presented in this thesis should be studied carefully

and the designer made cognizant of the limitations of this proposed procedure before applying it too generally.

VIII. RECOMMENDATIONS

It is recommended for future investigators of SO_2 boiling coefficients that:

1-A series of heaters of different configurations be tested,

2-Different size evaporators be evaluated,

3-A heating medium other than steam be utilized. If electricity is used, care should be taken to avoid damage to the heating element as the boiling goes from partial nucleate boiling to stable film boiling with a rapid rise in heat flux.

4-The effect of evaporator pressure at several comparatively high values be studied,

5-Ebullition of SO_2 at higher pressures be investigated,

6-The system used in this investigation be further instrumented to be more self-governing. This would aid in the earlier stabilization of the system.

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X. APPENDIX

DERIVATION OF EQUATIONS FOR CURVES

A logarithmic plot of boiling film coefficient versus surface-liquid temperature difference yielded a straight line, therefore the equation was the form, $h = a\Delta t^b$, where a and b are experimental constants. The equation was re-written as: $\log h = \log a + b \log \Delta t$. The experimental data was divided into two groups and the logarithms found for each value of h and Δt. The summation of the groups and the solution for the constants follow:

Group 1: $\lesssim \log h = \lesssim \log a + b \lesssim \log \Delta t$

$$30.47782 = 9 \log a + 11.45933\ b$$

Group 2: $\lesssim \log h = \lesssim \log a + b \lesssim \log \Delta t$

$$24.79056 = 8 \log a + 12.61003\ b$$

Solving these two equations simultaneously for a and b resulted in: $a = 39{,}330$ and $b = -0.949$. Therefore, the equation for the curve, Figure 17, is: $h = 39{,}330\ \Delta t^{-0.949}$. The procedure used was set forth in Reference 21 and termed "the method of averages for logarithmic form". A substitution of all experimental values of h into the above equation yielded an average per cent deviation of 9.29.

A Cartesian plot of heat flux, Q/A, versus W

yielded a straight line, therefore, the equation was of the form, $W = c + d\ (Q/A)$, where W was the flow of SO_2 and c and d were experimental constants. The experimental data was divided into two groups. The summation of the groups and the solution for the constants follow:

Group 1: $\lesssim W = c + d \lesssim (Q/A)$

$2229 = 8c + 342{,}200\ d$

Group 2: $\lesssim W = c + d \lesssim (Q/A)$

$2966 = c + 451{,}700\ d$

Solving these two equations simultaneously for c and d resulted in: $c = -9.25$ and $d = 0.00673$. Therefore, the equation for the curve, Figure 20, is: $W = 0.00673\ Q/A - 9.25$. The procedure used was set forth in Reference 21 and termed "the method of averages for linear form". A substitution of all experimental values of Q/A into the above equation yielded an average per cent deviation of 3.65.

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ABSTRACT

It was the intent of this investigation to determine boiling film coefficients of SO_2 in a temperature range not previously investigated. Further, a practical method of circulating liquid SO_2 and reclaiming SO_2 vapor without a compressor or pump was attained.

For the range investigated the boiling film coefficients decreased with an increase in temperature difference between the boiling SO_2 liquid and the surface of the heater. This occurred in the regime of partial nucleate boiling. Favorable comparisons were made with the limited information available for this boiling regime.

It was found that the controlling film coefficient was on the steam side of the test evaporator. This was due to the comparatively large values obtained for SO_2 boiling film coefficients. Within the limits of this investigation, the overall heat transfer coefficient may be taken as approximately equal to the steam film coefficient.

Much more work needs to be done in the regime of partial nucleate boiling as the available literature is far from complete.