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THE EFFECTIVE HEATING SURFACE IN A BASKET TYPE EVAPORATOR

by

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and
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A Thesis Submitted in Partial Fulfillment
for the Degree of
Bachelor of Science in Chemical Engineering

The University of Tennessee

August, 1938

Acknowledgment

The authors wish to express their appreciation to the following persons:

To Dr. R. M. Boarts Jr., for his advice and personal aid in this investigation.

To F. M. Chance and J. E. Gantt for their helpful suggestions.

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THE EFFECTIVE HEATING SURFACE IN A BASKET TYPE EVAPORATOR

Introduction

The heating element in an evaporator is composed of two heat transferring surfaces. These surfaces are the area of the tubes and the area of the drum surrounding the tubes. The entire heating surface, composed of these two surfaces is commonly referred to as the basket in the type of evaporator studied here. In the simple capacity equation which is used in evaporator calculations, the effective area (A) enters as a term along with the overall heat transfer coefficient (U), the rate of heat transfer ($\frac{Q}{\Theta}$), and the difference in temperature on the two sides of the heating surface (ΔT). The capacity equation is given as:

$$\frac{Q}{\Theta} = U\ A\ \Delta T \qquad (1)$$

It is the purpose of this thesis to determine with the aid of the above equation the effective heating surface in the experimental evaporator. The ΔT term of the equation is varied for a series of runs in which all of the tubes of the evaporator are open. A corresponding set of runs are made varying the ΔT, and in this series half of the tubes of the evaporator are closed, rendering them ineffective as heating surfaces. By correlation of the data obtained in this manner, it is possible to solve for the effective heating surface over that of the tubes in the evaporator under study.

General Survey of the Field

The subject of evaporation is quite old, and evaporators have been used in this country as well as abroad for a number of years. "In the middle ages the water bath was supposed to have been discovered by Maria the Jewess, an alchemist of the first century. A. D. Lippman points out that this is due to a mistaken connection between her and the expression "bain Marie" and that the water bath was actually used and known to Hippocrates centuries before.[1]"

"Steam heated open pans of various designs were in use in the sugar industry about 1800. The first real advance was the invention of the vacuum pan by Howard, who took out patents in 1812 and 1813. He used a jet condenser and an air pump, and reached a boiling point of 60 degrees Centigrade. His pan was immediately adopted in English refineries but found very slow acceptance on the Continent, apparently because of the general disinclination to use air pumps. Degrande, in 1824, improved Howard's pan by adding a heating coil, thus increasing its capacity. Roth's pan, which used no air pump but depended on large volumes of cooling water flowing into a large receiver, was more popular, but less efficient.[2]"

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(1) Badger, W.L., Heat Transfer and Evaporation, N.Y., Chemical Catalog Co., 1926, pp. 93-4.

(2) ibid.

"About 1848, Cail, of Cail and Cie., Berlin, began to build vertical tube evaporators, under patents received indirectly from Rillieux, according to statements made to this effect by Horsin-Deon. Robert, director of a sugar factory at Seelowitz, Austria, is usually given credit for the vertical tube evaporator, but apparently this idea of upright tubes was Rillieux's. [1]"

"In 1877 Riedel built an evaporator with an annular downtake and the heating surface in the form of a drum. This drum is usually carried on brackets; and steam may be supplied by a nipple connecting with the wall of the machine, or by a central pipe coming in from the top. The latter construction is easier to disconnect for repairs. These evaporators, usually known as the "basket type" are built by almost all makers in the United States.[2]"

Evaporation today is a fundamental part of many process industries, with special importance being placed on the operation in the sugar and salt industries. "An evaporator is a highly developed apparatus, based on a mass of theoretical and practical knowledge which can not be ignored if efficient results are desired. It is not simply a boiling pot, and he who proceeds on any such assumption will have to pay the penalty of his folly.[3]"

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(1) Badger,W.L., Heat Transfer and Evaporation,N.Y. Chemical Catalog Co.,1926,p.103.

(2) ibid.

(3) Webre,A.L.,Robinson,C.S.,Evaporation,N.Y.,Chemical Catalog Co.,1926,p.9.

In most cases today, evaporation is performed by using waste steam from some steam engine or prime mover of some type. This, of course, affects a saving in the use of heat in any given set-up, since this means that the power used for the evaporation is practically free.

The basket type evaporator of today is designed with the complete heating surface in one piece in order to insure easy removal of the heating surface and its replacement. By this method much time and money is saved in large scale production where it is necessary to stop evaporation and clean the heating surface from time to time in order to insure a better coefficient of heat transfer.

In the chemical industries, this apparatus has found very useful application in the concentration of crystallizing solutions, for it has a large, free annular downtake, permitting uninterrupted liquor circulation so essential to this class of work. The design in vogue is the familar cylindrical basket with steam admission into the top head.[1]

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(1) Webre,A.L.,Robinson,C.S.,Evaporation,N.Y.,Chemical Catalog Co.,1926,p.9.

Relation of Work to Previous Work

There has probably been work done experimentally and on a large scale on the effective heating surface in evaporators, but there is no evidence of such in the literature. Since there are only possibly five or six large manufacturers of evaporators, experimental work leading to design is kept as a trade secret. In the design of evaporators today, the manufacturers have empirical knowledge as to the variation of U (the overall heat transfer coefficient) with ΔT, in their different types of design. From the results of small scale tests and from operating data on previous installations, the effective heating surface necessary for any new evaporator is determined. This is purely a matter of experience, and so far as is known, no rational formula for the determination of the effective area of evaporators exists.

General Theory

The basic equation upon which most heat transfer calculations are based is known as the capacity equation:

$$\frac{Q}{\theta} = U A \Delta T$$

where;

$\frac{Q}{\theta}$ = the rate of heat transfer in Btu. per hour.

U = the overall heat transfer coefficient in Btu. /°F.Hr.sq.ft.

A = the area of the heating surface in sq.ft.

ΔT = the difference between the temperature of the steam and the bulk liquid temperature of the water in the evaporator in °F.

It is upon this equation that the calculations of this thesis are based, especially with reference to the area which is of primary interest.

In all ordinary cases of heat transfer in an evaporator, there are at least three resistances; the film resistance on the liquor side, the film resistance on the steam side, and the resistance of the metal wall through which the heat is being transferred. Where there is scale formation, this constitutes a fourth resistance. The overall coefficient of heat transfer is the reciprocal of the sum of all the resistances which may occur in a particular case.

The chief factors affecting the steam film coefficient are: steam pressure, non-condensible gases, quality or superheat in the steam, and condensate drainage.

Steam pressure will affect the heat transfer in the evaporator, since an increase in the steam pressure would in turn increase the ΔT, and hence the steam film coefficient would be raised by an increase in steam pressure.

The removal of non-condensed gases is a subject which can not be neglected in considering the steam film coefficient. These gases are usually looked upon as air in the steam, air in the feed liquor, and air through leaks. The effect of air is to form an additional resistance which will of course lower the steam film coefficient. According to Badger, experimental data obtained from a basket type evaporator showed that the lines of flow of air were so definitely down the basket that air vents at the top were practically useless.[1] This is, however, not true in the case of experimental evaporator under consideration here, since the steam inlet is over one half of the distance to the bottom of the steam chest causing a flow to the top of the chest which would expel any non-condensible gases in the chest. A vent is also used at the bottom of the evaporator under consideration here.

Superheat in the steam affects both the mean temperature drop and the heat transfer coefficient. The relation of temperature with distance along the tubes can be imagined

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(1) Badger,W.L.,Heat Transfer and Evaporation,N.Y., Chemical Catalog Co.,1926,p.92.

to start at a temperature corresponding to some degree of superheat, dropping off after a small length of tube, proceeding on a constant temperature line until the very end of the tube, where the temperature drops off to about as much below the constant line temperature as the superheat temperature was above it. Superheat, even to 40°, however, would be of very little effect on the heat transfer coefficient since the specific heat of steam is about .48 and thus the heat lost in becoming saturated would be practically neglible in relation to the heat of condensation. With consideration to this, it can be seen that it would take a considerable degree of superheat to effect the steam film coefficient.

The type of condensation and its removal is another factor which must have a bearing on the steam film coefficient, but this is not susceptible to control. When film type condensation on a vertical tube exists, the condensate must all drain the whole length of the tubes, and hence the mean thickness of the film on the tube will be greater on a long tube than on a short one. When dropwise condensation exists, a much greater steam film coefficient is obtained, and no means of calculation for this case exists. Because of the constancy of the experimental runs, and because the steam supply used was relatively free from oil, the condensation was believed to be entirely film type.

Scale formation constitutes a fourth resistance. This formation may easily become the controlling factor with reference to the heat transfer coefficient. Soft water was used in the present study in order to guard against scale formation.

With reference to the liquid film coefficient, an increase in velocity means in general an increase in heat transfer or an increase in viscosity means a decrease in heat transfer, other factors remaining constant. However, no data are in existence at present in which velocity of natural circulation has been measured or approximately estimated. Since in the basket evaporator the circulation is caused by the steam evolved rising through the tubes and carrying the liquid with it, it is to be expected that with longer tubes it would be possible to generate a stronger pumping action. This led designers to use longer and longer tubes, but tubes over six feet long in this type of evaporator are rare due to the difficulty of cleaning. No systematic study has, however, been made of tube proportion. Probably the velocity of the liquid is the major factor that controls the heat transfer from steam to boiling liquids. The result of the presence of a layer of foreign material on the heating surface can be seen with reference to the preceding discussion on scale formation, but a better comprehension of the quantitative effect of such deposits may be had by remembering that the thermal

conductivity of most scale forming materials is approximately the same as that of water. No lengthy discussion of such effects is considered to be necessary.

Description of the Method and Apparatus Used

Reasons for the Present Method

For two men operating the experimental evaporator, the method chosen seems to be the only feasible one. Under the conditions of the method chosen, one man was placed at the main body of the evaporator in order to control the vapor and steam temperatures and pressures, the temperature of inlet feed by control of the supply of steam through the pre-heater, and the liquid level around the basket of the evaporator. The other man was placed below at the condensate tanks in order to take readings of vapor condensate and steam condensate. Manometers were used to obtain steam and vapor temperatures, since they were more accurate than thermometers and less sluggish. Steam from the University power house, was throttled from about 35 lbs.sq.in. gauge pressure to the pressure corresponding to the temperature desired in the evaporator. This caused the steam to become superheated. The degree of superheat was measured with an industrial thermometer placed at the steam inlet. Feed water was kept constant at 160°F., since this would simplify calculations and the feed pump would carry the water at this temperature without vapor binding. A six pass tubular condenser was used as a source of vacuum supplemented by a vacuum pump to pull out the non-condensible gases. Soft water was used to insure little scale

formation. A small amount of sodium dichromate was put into the feed solution to minimize corrosion. Steam was bled through the steam chest to insure the best possible removal of non-condensible gases. The liquid level in the evaporator was controlled visually since this was the only available method. Staggered readings were taken on the condensate tanks in order to enable one man to take all the readings in the unit time of the run and at the intervals specified. The steam condensate is used as the basis for heat transfer calculations here, and therefore, the level of condensate in the steam tanks was read more often than the level in the vapor tanks.

Apparatus Used

Basket Evaporator:

Shell: 3 sections, each 3/16" Pl. rolled to 20"O.D.height 30-3/4"

Steam Chest: 3/16" Pl.-16"O.D. height 28"

Tubes: 54-1"O.D.-18 BWG, 1-5/8" between centers

Deflector set on 3" std. pipe 6" above top sheet.

Condenser: 6 pass-6' long.

Tubes:-48-5/8"O.D. 20 BWG

Manometers: 3 standard type,

One for feed orifice

One for steam pressure

One for vapor pressure.

Pre-heater for feed:

Tank 36" high-23"Diameter

With closed steam coil

Storage tank: 78" high-53" Diameter

Pumps:

One positive action Viking for pumping feed

One centrifugal for pumping back to storage tank from the condensate tanks.

Figure 4 shows a layout of the apparatus.

Discussion of Operation

In operating the evaporator on a given run, the following data were taken:

Quantity of steam condensate

Quantity of vapor condensate

Steam temperature (varied from 180°F.to 266°F.)

Vapor temperature (varied from 176°F.to 230°F.)

Steam inlet temperature (for getting superheat)

Feed temperature (maintained at 160°F. approximately)

Complete control of the evaporator with respect to temperature drop was exercised by adjusting the vapor and steam pressures. The heat transferred was measured by the condensate taken out at the condensate tanks per unit time of a run. Applying to small corrections of superheat, feed temperature, barometric pressure, etc. to this heat transfer and knowing the ΔT from the difference in pressure on the vapor and steam sides, this leaves only the two unknowns U and A in the capacity equation to be determined.

The vacuum available was not sufficient for the 194°F.(steam)- 176°F. (B.P.) run.

Frothing was encountered on the 212-230°F. run. This was assumed to be due to the presence of organic matter and disappeared after the evaporator body had been drained and washed.

A 54° ΔT run was found to exceed the capacity of the evaporator. However, this run was made, but it was obvious that there was a high percentage of entrainment. This difficulty arose from the construction of the evaporator and could possibly be removed by raising the deflector and enlarging the annular space between basket and the shell.

Sources of Error

No account was taken of boiling point elevation due to hydrostatic head. It was practically impossible to keep the liquor level constant. The feed was not introduced exactly at a constant rate. Vapor condensate had a tendency to come out in slugs. Steam was bled through the steam chest to blow out non-condensible gases. This means that the superheat of the steam so used was transmitted to the liquor without any account being made of it in the heat balance. No account was taken of the radiation losses from the shell.

Precautions Taken to Avoid Errors

Superheat was measured by the use of a thermometer. Thermometers were used to check the temperature calculated from manometer readings. Runs at atmospheric pressure were made slightly above atmospheric pressure to insure non-condensible gas removal and that no air was sucked back into vapor space. Distilled water was used in preference to city water in order to keep down scale formation during test.

Nomenclature

The nomenclature used in this paper is as follows:

Subscript 1 : applies to the data taken with half of the tubes open.

Subscript 2 : applies to data taken with all tubes open.

Q = amount of heat transferred in Btu. (based on steam condensate)

Θ = time of runs in hours

A = area of tubes in sq.ft.

U = overall heat transfer coefficient in Btu./ hr. ^{o} F. sq.ft.

X = basket effect in sq.ft. (over and above the tube area.)

ΔT= temperature of steam - temperature of vapor in ^{o}F.

Interpretations of headings on Tables 1 and 2

Steam temperature = temperature of steam in steam chest in ^{o}F.

Vapor temperature = temperature of boiling liquid in ^{o}F.

Steam temperature (inlet) = temperature of the inlet steam (used to measure superheat.)

Total heat-Btu.steam = total heat in Btu. /hr. based on heat calculated from steam condensate.

Total heat-Btu.vapor = total heat in Btu. /hr. based on heat required to evaporate the feed.

Steam condensate-Cm. = the rise in tank level of steam condensate per 20 minute run, in centimeters.

% Discrepancy = the % of heat unaccounted for based on steam condensate.

Figure 1 - refers to graph of rate of heat transfer versus ΔT.

Figure 2 - refers to graph of ΔT versus overall coefficient.

Figure 3 - refers to graph of ΔT versus overall coefficient.

For further information on the curve sheets see correlation of data section.

Correlation of Data

If the graph of rate of heat transfer versus ΔT, Figure 1, is studied carefully, it is observed that the effect of change in boiling point of the liquid in the evaporator is not as large as the experimental error. Since no conclusive data were obtained on the effect of boiling point on heat transferred, the effect of boiling point is neglected and K_1 and K_2 were drawn as averaged slopes through the experimental points of Figure 1. Lines K_1 and K_2 are straight lines since the experimental data did not warrant any curve. The lines are extrapolated out to zero heat transfer, the intercept at zero heat transfer being at 5° ΔT.

Theoretically, there is zero heat transfer at zero ΔT which means that lines K_1 and K_2 should meet at zero ΔT and zero heat transfer. However, this error becomes smaller with an increase in ΔT so that it is believed that the straight line relationship is approximately correct in the range of experimental data shown.

If ΔT is taken as ordinate and heat transfer taken as abscissa, then the cotangent of the angle between the line and the abscissa is 0.16 for K_1 and 0.28 for line K_2. The intercept is at $\Delta T = 5^\circ$.

It is desired to treat lines K_1 and K_2 analytically to derive a simpler relationship between the two unknowns; the total effective area and the overall heat transfer coefficient.

Writing the equation for lines K_1 and K_2:

$$\frac{Q_1}{\theta} = U_1 A_1 \Delta T = (\Delta T - 5)\ 16000 \qquad (2)$$

$$\frac{Q_2}{\theta} = U_2 A_2 \Delta T = (\Delta T - 5)\ 28000. \qquad (3)$$

Rewriting and simplifying:

$$U_1 A_1 \Delta T = 16000\ \Delta T - 80{,}000 \qquad (4)$$

$$U_2 A_2 \Delta T = 28000\ \Delta T - 140{,}000 \qquad (5)$$

Solving (4) and (5) simultaneously:

$$1.75\ U_1 A_1 \Delta T = 28000\ \Delta T - 140{,}000, \qquad (6)$$

or:

$$1.75\ \frac{Q_1}{\theta} = \frac{Q_2}{\theta}. \qquad (7)$$

$$1.75\ U_1 A_1 \Delta T = U_2 A_2 \Delta T \qquad (8)$$

For equal ΔT:

$$1.75\ U_1 A_1 = U_2 A_2 \qquad (9)$$

Now, since the inside area of the 54 tubes in the evaporator is 29.7 sq.ft. and the area of half of the tubes is 14.85 sq. ft.;

A_2 = 29.7 sq.ft. + X_2 (basket effect)

A_1 = 14.85 sq.ft. + X_1 (effect of basket).

Rewriting equation (9):

$$1.75\ U_1\ (14.85 + X_1) = U2(29.7 + X_2) \qquad (10)$$

Rewriting equation (9):

$$1.75 = \frac{U_2 A_2}{U_1 A_1} \qquad (11)$$

Equation (10)and (11) means that if the total effective area is constant the coefficient must also be constant, or if the overall heat transfer coefficients change, the area must be changed enough in the opposite direction to keep the ratio between U_2A_2 and U_1A_1 equal to the constant 1.75. Equation (11) has more unknowns than can be solved with the information at hand. Therefore assumptions were made to delimit the range of overall coefficient that must be present.

Assume $X_1 = X_2 = 0$

This assumption gives the maximum overall coefficient that can be expected-slightly below 1000 for half tubes open and about 860 for all tubes open. These coefficients are probably high since overall coefficients of 1000 are very seldom obtained except in systems where the liquid film coefficient attains a high value due to forced convection. This indicates that there is an appreciable effective area other than the tube area.

Assume $X_1 = X_2 =$ constant

Then such a value of X_1 and X_2 was chosen so that U_1 and U_2 were approximately equal as is shown by Figure 3. This value of X was 5.25 sq.ft., the quantity of heat being transferred being read off of Figure 1 for all calculations of Figure 2 and Figure 3.

This procedure does not necessarily prove that the effective heating surface of the basket over and above the tube area is a constant. It does correlate the data obtained in these runs. It is desirable to regard the overall coefficient as acting independently of the area of the evaporator. Then, it can be seen that assigning a constant value for the heat transferring capacity of the basket apart from the tubes is justified on the grounds that such a factor would aid in the design of basket type evaporators.

The present work is not an answer to this problem of design. It does point the way to an extension and corroboration of these data for the purpose of obtaining more logical design methods.

Conclusions

The conclusions drawn from this study are as follows:

(1) From the data taken it can be concluded that there is more heat transferred at the higher boiling points, other factors being equal. The present data are not quite complete enough for a conclusive statement, but there is enough indication of the fact to make reference to it.

(2) The capacity of the evaporator was found to be exceeded at a heat transfer of 1,376,000 Btu./hr., this heat being based on the steam condensate. The velocity of vapor passing the deflector was so great on this run that it could easily be seen that entrainment was present. To further the evidence that entrainment was present on this run, the vapor condensate was drained and found to contain enough sodium dichromate to color the water. At a ΔT corresponding to a slightly smaller heat transfer than in the run above capacity, the evaporator behaved normally. This ΔT was about 54°F., and the boiling point was 176°F. There was, however, a very distinct line of demarkation as the capacity was exceeded.

(3) The overall coefficient of heat transfer was found to increase with ΔT, but to increase at a decreasing rate; being about 580 (based on A_1 = 20.10 sq.ft. and A_2 = 34.95 sq.ft.)

at a ΔT of $20^{\circ}F$. At a ΔT of about $70^{\circ}F$., the overall heat transfer coefficient was found to be 740. This is shown by Figure 3. If the overall heat transfer coefficient is based on the area of the tubes only, the coefficients are much larger and the curves tend to separate farther with a larger ΔT. (See Figure 2)

(4) It was found that if the effective area of the basket over and above the tube area were equal to 5.25 sq.ft. and a plot of coefficients was made versus ΔT, that the coefficient curves for all tubes open and for half tubes open were practically identical. (See Figure 3)

(5) The effect of the basket area in a basket type evaporator is practically constant, being 53.2% as efficient as an equivalent tube area in the evaporator which was operated.

Bibliography

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Webre,A.L.,Robinson,C.S.,Evaporation,N.Y.,ChemicalCatalog Co.,1926.

The following references are cited since they were indirectly used for comparison and checking of some calculations in this thesis:

Badger,W.L.,McCabe,W.L.,Elements of Chemical Engineering, N.Y.,McGraw-Hill Book Company,1936.

McAdams, W.H.,Heat Transmission,N.Y.,McGraw-Hill Book Co., 1933.

Boarts,R.M.Jr.,Liquid Film Heat Transfer for Long Vertical Tubes,University of Michigan,1936.

APPENDIX

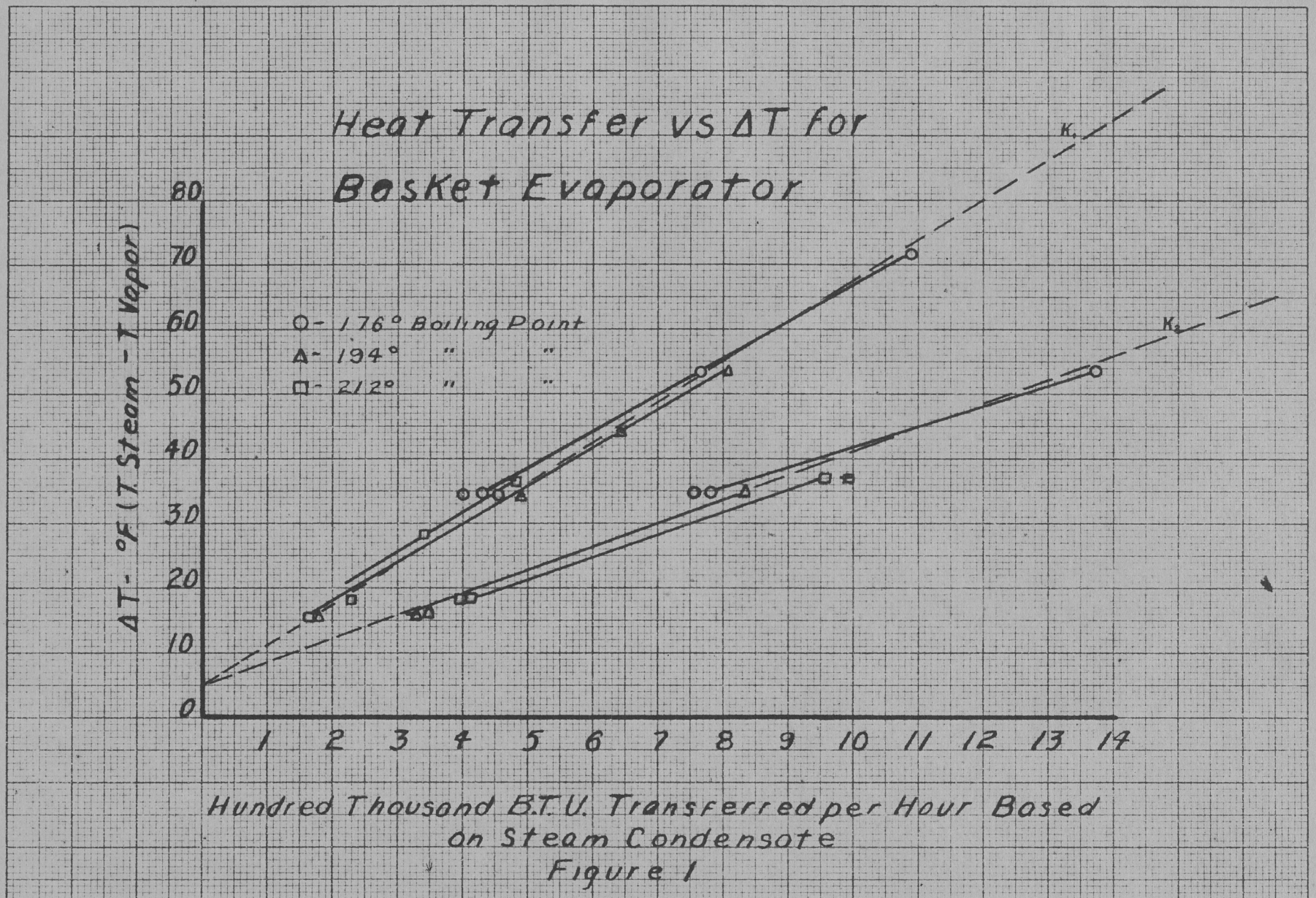

Heat Transfer vs ΔT for
Basket Evaporator
ΔT - °F (T Steam - T Vapor)
80
70
60
50
40
30
20
10
0
O - 176° Boiling Point
Δ - 194° " "
□ - 212° " "
K1
K2
1 2 3 4 5 6 7 8 9 10 11 12 13 14
Hundred Thousand B.T.U. Transferred per Hour Based on Steam Condensate

Figure 1

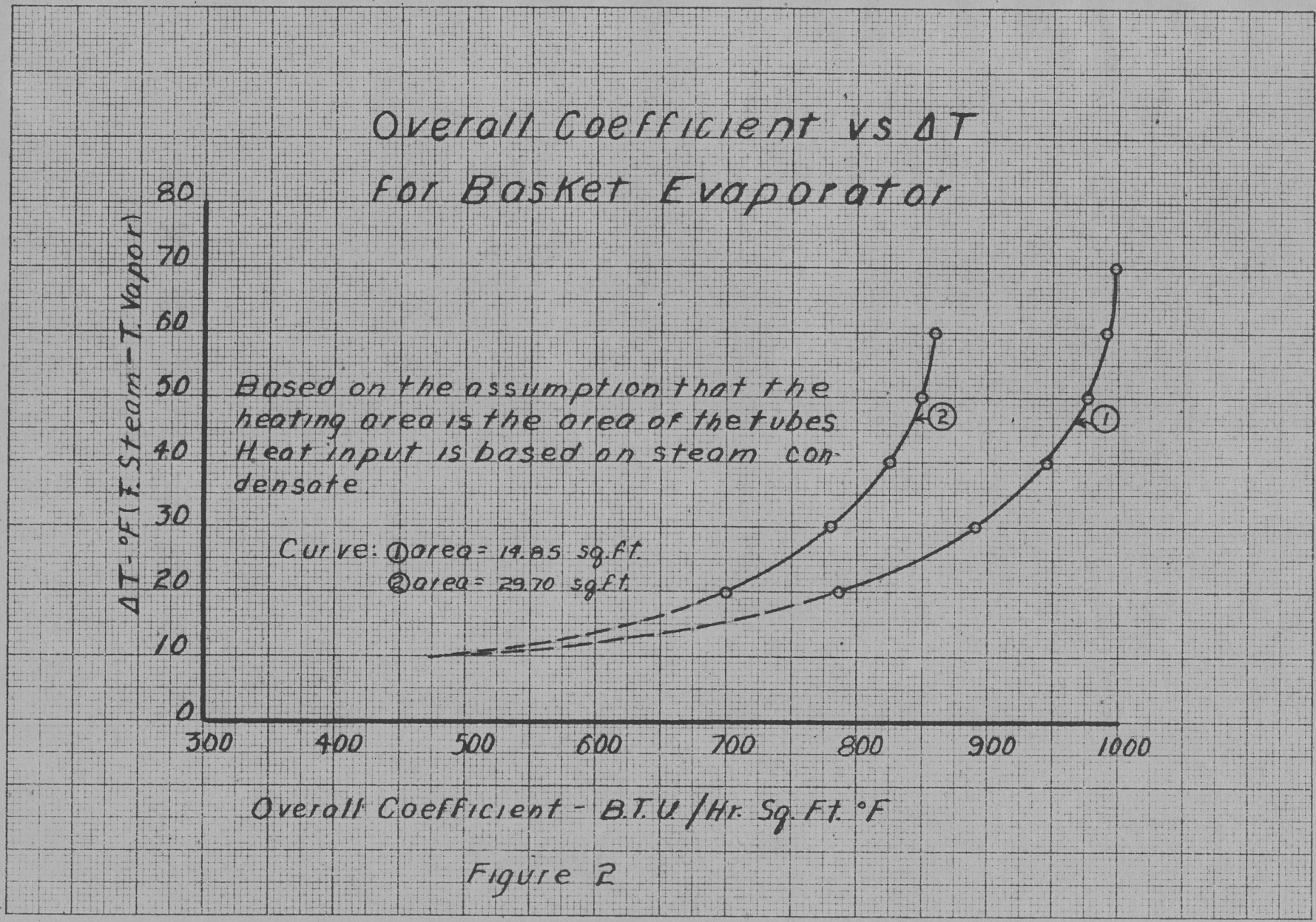
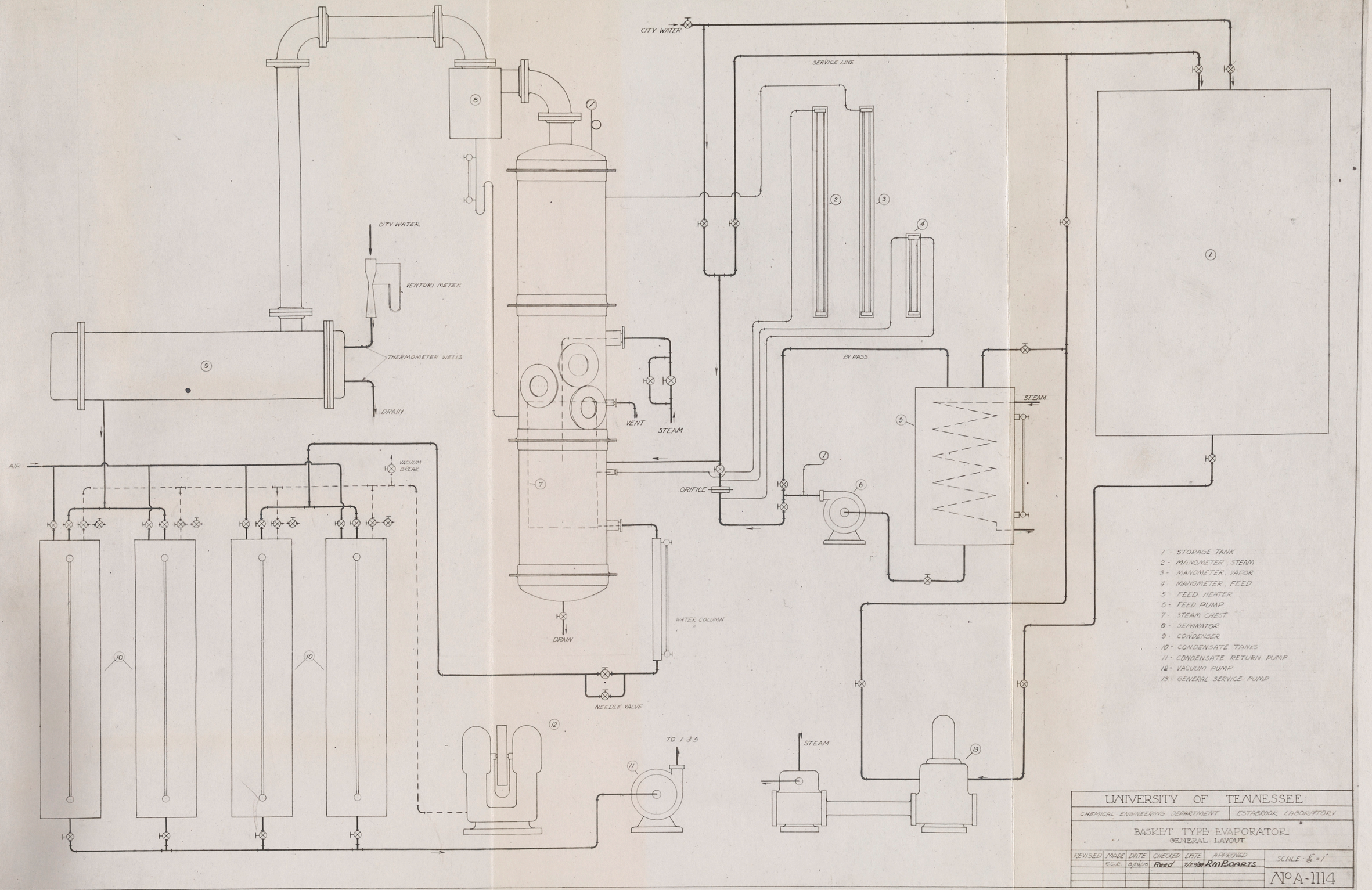

Overall Coefficient vs ΔT
for Basket Evaporator
ΔT - °F (T. Steam - T. Vapor)
80
70
60
50
40
30
20
10
0
300
400
500
600
700
800
900
1000
Based on the assumption that the heating area is the area of the tubes
Heat input is based on steam condensate
Curve: ① area = 14.85 sq. ft.
② area = 29.70 sq. ft.
Overall Coefficient - B.T.U./Hr. Sq. Ft. °F

Figure 2



UNIVERSITY OF TENNESSEE

CHEMICAL ENGINEERING DEPARTMENT | ESTABROOK LABORATORY

BASKET TYPE EVAPORATOR
GENERAL LAYOUT

REVISED	MADE	DATE	CHECKED	DATE	APPROVED	SCALE - 1/8" = 1'
	R.G.R.	8/20/46	Reed	7/29/47	R.M. BOARTS	

No A-1114

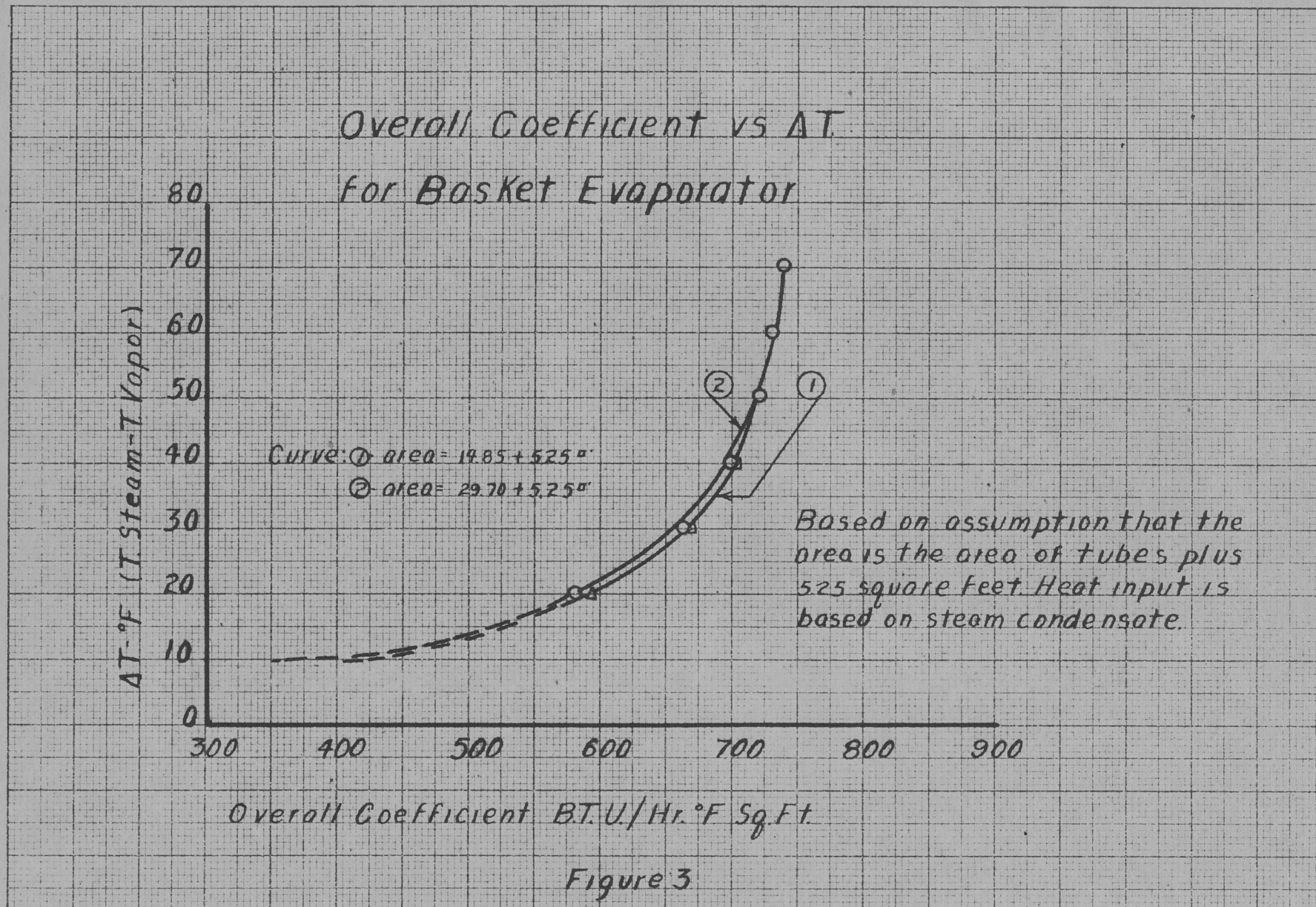
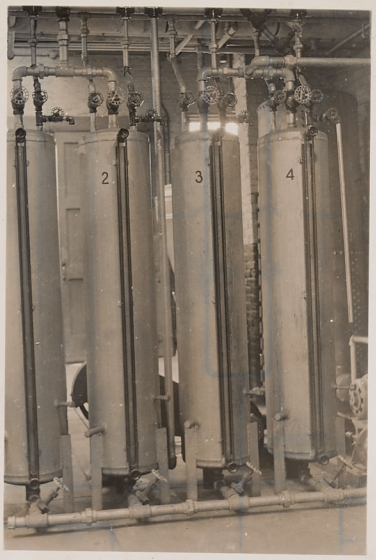
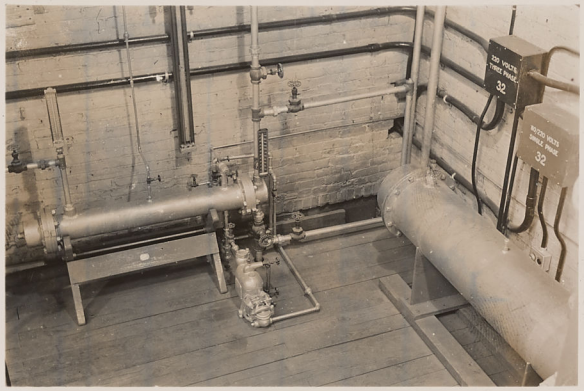

Overall Coefficient vs ΔT
for Basket Evaporator
80
70
60
50
40
30
20
10
0
ΔT - °F (T Steam - T Vapor)
Curve: ① area = 19.85 + 5.25 a''
② area = 29.70 + 5.25 a''
Based on assumption that the area is the area of tubes plus 525 square feet. Heat input is based on steam condensate.
300
400
500
600
700
800
900
Overall Coefficient BTU/Hr. °F Sq. Ft.

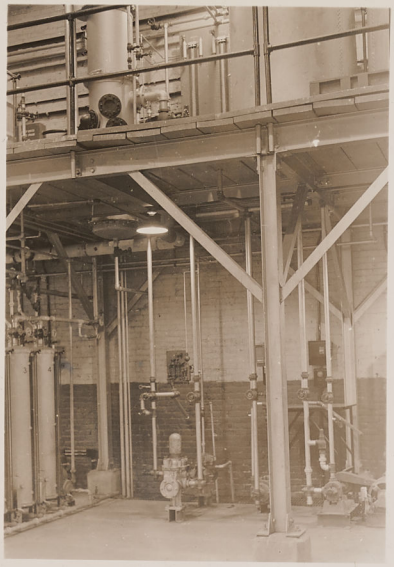
Figure 3



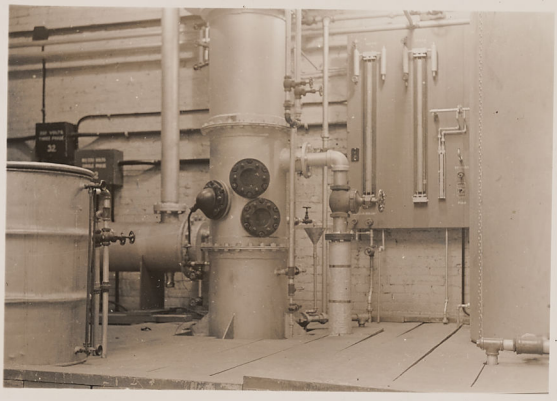
Drip Tanks



Condenser



General Layout



Evaporator Shell

TABLE 1

DATA FOR HEAT TRANSFER WITH HALF OF THE TUBES IN THE BASKET IN USE

Series	Run Number	Steam Temp.	Vapor Temp.	ΔT	Steam Temp. Inlet	Steam Condensate Cm.	Vapor Condensate Cm.	Total Heat Btu.Steam	Total Heat Btu.Vapor	% Discrepency %	Overall Coefficient Btu./°F.Hr.Sq.Ft.
5	1	211.6	177.1	34.5	246	74.7	70.5	456,040	440,790	0.73	657
3	1	211.6	177.1	34.5	243	65.6	71.2	399,760	443,910	-9.04	578
2	3	211.5	177.1	34.5	248	70.8	77.0	430,330	480,070	-8.65	623
4	2	230.1	176.7	53.4	232	129.1	126.0	760,130	785,420	-3.97	709
4	3	248.3	176.7	71.6	252	186.0	183.3	1,089,900	1,142,800	-2.06	756
3	2	211.6	195.5	16.1	274	26.0	22.3	160,960	138,550	7.19	497
2	2	211.5	195.4	16.1	272	28.0	24.9	173,160	156,320	1.03	535
3	4	230.3	195.5	34.8	252	82.6	78.5	494,770	492,950	0.27	706
4	1	230.1	195.2	34.9	252	86.0	80.0	515,220	502,210	0.40	736
1	2	239.6	195.0	44.6	241	108.8	107.2	641,210	672,970	-4.90	713
1	5	248.2	195.0	53.2	250	137.0	131.8	802,090	827,490	-3.15	750
3	5	230.3	211.7	18.6	265	38.3	32.6	230,190	205,870	9.50	615
3	3	230.3	211.7	18.6	265	37,9	33.9	227,050	214,090	1.75	608
1	3	239.6	211.4	28.2	241	57.3	62.4	337,520	394,000	-5.97	594
1	4	248.2	211.4	36.8	250	81.0	88.4	474,250	564,150	-5.27	640
2	1	248.3	223.3	25.0	252	53.5	47.7	325,460	301,440	1.35	647
2	4	248.4	224.1	24.3	252	54.4	45.6	333,810	289,270	7.48	684

TABLE 2

DATA FOR HEAT TRANSFER WITH ALL OF THE TUBES IN THE BASKET IN USE

Series	Run Number	Steam Temp.	Vapor Temp.	ΔT	Steam Temp. Inlet	Steam Condensate Cm.	Vapor Condensate Cm.	Total Heat Btu.Steam	Total Heat Btu.Vapor	% Discrepancy	Overall Coefficient Btu./°F.Hr.Sq.Ft.
6	2	211.6	176.7	34.9	214	126.5	121.9	758,500	758,700	0.06	623
7	1	211.7	176.8	34.9	214	130.6	124.4	783,970	775,350	0.91	641
7	3	230.3	176.8	53.5	255	228.3	248.3	1,376,700	1,547,900	-8.02	736
6	3	211.6	195.1	16.5	224	54.5	49.3	328,940	309,460	1.69	568
6	1	211.6	195.1	16.5	240	55.6	51.9	338,370	325,840	2.70	586
9	3	230.3	195.1	35.2	232	139.0	136.2	823,950	855,090	-2.64	669
9	5	230.3	195.1	35.2	232	140.3	133.0	831,720	854,290	-3.61	677
7	4	230.3	211.6	18.7	254	65.9	58.8	395,190	371,360	1.66	604
8	1	230.3	211.4	18.9	232	69.2	62.6	410,200	395,280	2.78	620
8	2	248.5	211.4	37.1	250	162.5	149.0	950,830	940,800	0.95	733
9	1	248.5	211.4	37.1	254	151.0	139.9	885,730	883,490	0.40	682
9	2	248.5	230.6	17.9	258	55.8	46.0	327,950	292,490	9.50	523
9	4	248.5	230.4	18.1	254	54.4	46.6	318,710	381,690	-5.06	504

TEST ON BASKET EVAPORATOR

Sample Calculation Sheet No. I

Date of run	7/25/38		
Series	8		
Run Number	2		
Room Temperature °F.	83		
Steam press.R.side Cm.Hg.	87.0	Vapor press.L.side Cm.Hg.	49.1
" " L. " " "	10.9		
Gauge pressure " "	76.1	Vapor Press.R.side Cm.Hg.	50.0
Barometric Pressure " "	74.1		
Absolute Pressure " "	150.2	Gauge press.Cm.Hg.	0.9
Steam Temp.inbasket °C.	120.27	Bar.press.Cm.Hg.	74.1
" " " " °F.	248.5	Absolute press.Cm.Hg.	75.0
Temp.in vapor space °F.	211.4	Temp.in vapor space °C.	99.63
T °F.	37.1		
Steam Temp. (inlet) °F.	250	Feed temp.°F.	160.0
L.H.of steam cond. Btu/#	946.5	L.H. vapor cond.Btu/#	970.6
Superheat steam Btu/#	0.7	Sen.heat of feed Btu /#	51.4
Heat available Btu/#	947.2	Heat trans. Btu /#	1022.0
Steam condensate Cm.	162.5	Vapor Cond. Cm.	149.0
		Log Cm.vapor cond.	2.1732
Log Cm.steam cond.	2.2108	Log factor	0.7909
*Log factor	0.7909	Log heat trans.	3.0094
Log heat available	2.9764	Log total heat trans.	5.9735
Log total heat available	5.9781	Total heat trans.	940,800
Total heat available Btu/hr.	950,830		
U	733		
**% Discrepancy	0.95		

* Factor - $\frac{.253 \cdot 60.9 \cdot 3}{7.48} = 6.18$

Where: .253 = tank calibration in gal./Cm.
60.9 = #H_2O/Cu.ft. at 165°F.
7.48 = Gal/ Cu.ft.
3.0 = Factor to convert 20 min. runs to hour runs.

**% Discrepancy = $100 \left(1 - \frac{\text{heat transferred}}{\text{heat available}}\right)$

APPROVAL

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

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