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**AN EXAMINATION OF BUBBLE SIZES FOUND IN LOCAL  
BOILING HEAT TRANSFER EXPERIMENTS**

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**Andrew D. Whitehead**

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AN EXAMINATION OF BUBBLE SIZES FOUND  
IN LOCAL BOILING  
HEAT TRANSFER EXPERIMENTS

by

Andrew D. Whitehead

B.S. Chem. (1944) Brown University

B.S. E. E. (1948) Massachusetts Institute of Technology

Submitted in partial fulfillment of the  
Requirements for the degree of

MASTER OF SCIENCE

at the

Massachusetts Institute of Technology

(1955)

Signature of Author \_\_\_\_\_

Department of Chemical Engineering, July 22, 1955

Signatures of Professors  
in Charge of Research \_\_\_\_\_

Signature of Chairman of  
Department - Committee on  
Graduate Students \_\_\_\_\_



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An Examination of Bubble Sizes Found in  
Local Boiling Heat Transfer Experiments

by

Andrew D. Whitehead

Submitted to the Department of Chemical Engineering on  
22 July, 1955 in partial fulfillment of the requirements  
for the degree of Master of Science in Nuclear Engineering.

ABSTRACT

Photographic examination of the bubbles occurring in  
Local Boiling Heat Transfer experiments has led to the con-  
clusion that for distilled, degassed water and a stainless  
steel surface, the bubbles prefer one particular size, -2.86  
mils diameter. A model or mechanism for vapor formation is  
introduced and it is shown that this model supports the ob-  
served phenomena of a preferred bubble size. Use is made of  
the result of the analysis as a condition for boiling, and a  
possible explanation of the burnout process.

Thesis Supervisors

Warren M. Rohsenow  
Associate Professor  
Mechanical Engineering

John A. Clark  
Assistant Professor  
Mechanical Engineering

Volume 10, Number 1

Submitted to the Department of Chemical Engineering on  
25 July, 1962 in partial fulfillment of the requirements  
for the degree of Master of Science in Chemical Engineering.

ABSTRACT

Photographic standards of low density occurring in  
local films that transfer exposures due to the pro-  
cessing of film for medical, dental and scientific  
uses are discussed. The authors have determined that  
this process is not as uniform as was previously  
assumed and it is shown that the usual exposure of 0.5-  
1.0 mAs is not sufficient to produce a uniform  
exposure of a photographic film. The authors  
investigate the variation of the optical density

Author's Name  
Address  
Institution

Thesis Number

John A. Clark  
Assistant Professor  
Department of Chemical Engineering

## ACKNOWLEDGEMENTS

This thesis was carried out under the supervision of Professor W. M. Rohsenow and Professor J. A. Clark, to whom I am indebted for their advice, intelligent criticism, and encouragement. I wish to thank Peter Griffith for the many hours he spent with me in discussion of the very complex problems of bubble formation.

THE UNIVERSITY OF CHICAGO  
DEPARTMENT OF CHEMISTRY

MEMORANDUM

DATE: \_\_\_\_\_

This matter was referred to me by the Department of  
Physics at the University of Chicago, and I have  
been asked to advise on the feasibility of the  
proposed experiment. I have given the matter  
some thought and in view of the very  
serious nature of the problem, I have  
consulted with the Department of Physics.

The proposed experiment is of a very  
interesting nature and it is  
certainly worth the effort  
involved in carrying out the  
experiment. However, the  
Department of Physics is  
not in a position to  
undertake this experiment  
at the present time.

Very truly yours,  
[Signature]

[Signature]

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## 1. Introduction

For some years now there have been considerable efforts made to take advantage of a heat transfer process known as Local Boiling Heat Transfer. It is characterized by a large difference in temperature between the heated surface and the bulk of the fluid. In general, the temperature of the fluid is well below the saturation temperature for the fluid pressure while the heated surface is a few degrees above the saturation temperature. The boiling is confined to the vicinity of the surface. The bubbles that do form condense in the cooler bulk fluid and there is not net vapor generation. The process can immensely increase the heat transfer rate without any appreciable increase in temperature difference between heated surface and bulk fluid.

In the course of some experimental investigations to determine density variations due to local boiling, some several thousand pictures were made of the vapor bubbles formed on the heated surfaces, under a wide range of conditions. At considerable effort, the bubbles on many of these pictures were counted and classified as to size.

The object of this paper is to examine these 'bubble counts' to obtain some further understanding of 'Local Boiling Heat Transfer'.

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well below the saturation temperature for the liquid pressure

with the liquid surface is a few degrees above the saturation

temperature. The boiling is confined to the vicinity of the

surface. The bubbles that do form - rise in the liquid with

fluid and there is not any vapor generation. The process can

readily be distinguished from film boiling which occurs without any

liquid movement in temperature difference between heated sur-

face and bulk fluid.

In the course of some experimental investigations to de-

termine density variations due to local boiling, some several

thousand pictures were made of the water surface under an in-

frared camera, under a wide range of conditions. As a result

of this effort, the picture on any of these pictures was analyzed

and classified as to size.

The object of this paper is to describe these picture counts

to obtain some further understanding of local boiling heat

transfer.

## 2. Presentation of 'bubble count' data.

### a. Source of data.

As described in reference (H-1), George Henry, in 1953, was able to photograph boiling on an electrically heated surface over which distilled, degassed water was flowing at various combinations of pressures - (500, 1000, 1500 psia) -, velocities - (20, 30 ft. per second) -, subcoolings - ( $0^{\circ}$ -  $150^{\circ}$  F) -, and rates of heat transfer - (approximately 10 - 50% required for burnout). The flow channel was a square section .500 inches on a side. The water flowed vertically upward through this channel. The heated surface was of stainless steel and 3.00 inches long and 3/8 inch wide. These conditions resulted in fully developed turbulent flow in the region of interest. The boiling that occurred was observed to be confined to a layer next to the heated strip. Bubbles were observed to form on the surface and to collapse without apparently moving from the surface. This type of boiling is called 'local boiling' and is characteristic of boiling under conditions of highly turbulent flow of subcooled water over heated surfaces.

A 4 by 5 inch laboratory still camera was focused on the heater strip in conjunction with a single flash light source whose period of illumination was about 2 microseconds.

A complete description of the apparatus used and the experimental techniques employed is given in references (H-1). References (H-1) and (H-2) made further use of the apparatus and refined the techniques used in (H-1). Since in this paper little exact use is made of any data, it is believed unnecessary to re-



count the experimental procedures in detail.

For a particular set of variables, system pressure, bulk temperature, velocity, and heat flux density, ten pictures were taken. Four of these ten were selected as representative and analyzed. To reduce the time for analyzing the pictures, only a small portion of the heater strip near its upstream end was used. The size of this section was 1/2 inch in length by 1/2 inch in width. Gauge marks in the channel were used to determine the picture magnification for each picture. This area was subdivided and the number of bubbles counted in each subdivision. Then each bubble of picture size of 0.01 inch diameter and greater was measured to the nearest 0.01 inch. All remaining bubbles were grouped at a picture size of 0.005 inches.

The actual dimensions of the bubble were determined from the magnification factor (approximately 5), and a factor to account for the relation between apparent bubble size and true bubble size (experimentally determined to be .834). This is worked out in reference (K-1) and corrected in reference (K-1). With measurements made to the nearest 0.01 inch, a magnification factor of 5 and a conversion factor of .834, it is apparent that the resolution in determining the actual size of the bubble is of the order of 2.5 mils.

The results are tabulated in appendix I as numbers of bubbles at each bubble size for the particular conditions in force.

#### b. Discussion of Data.

Several representative sets of data from appendix I have been plotted in figure (1). One of the four pictures that was used to obtain each set of data is shown in figures (2) through (7).

about the appropriate treatment in detail.

The first part of the report is devoted to a

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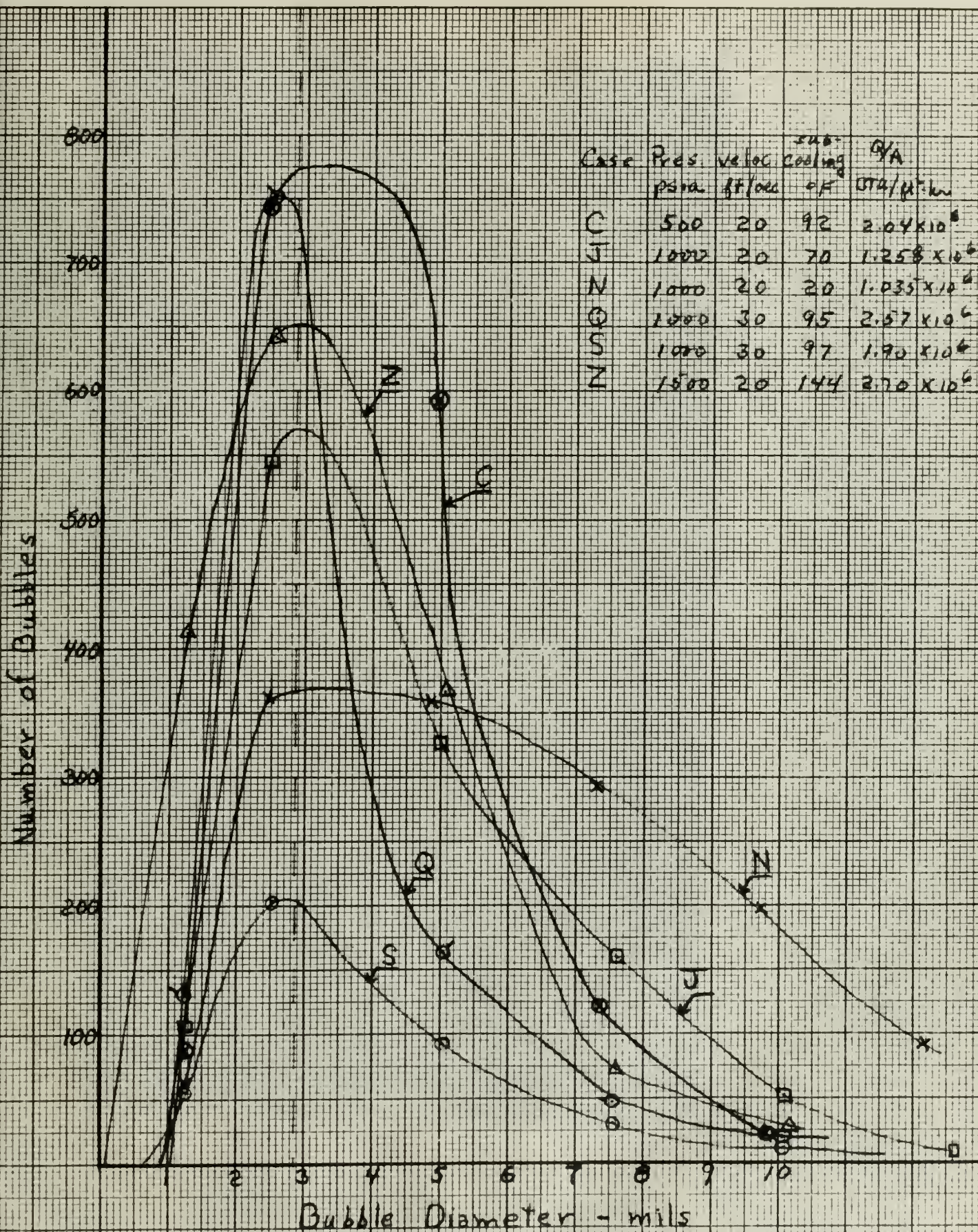
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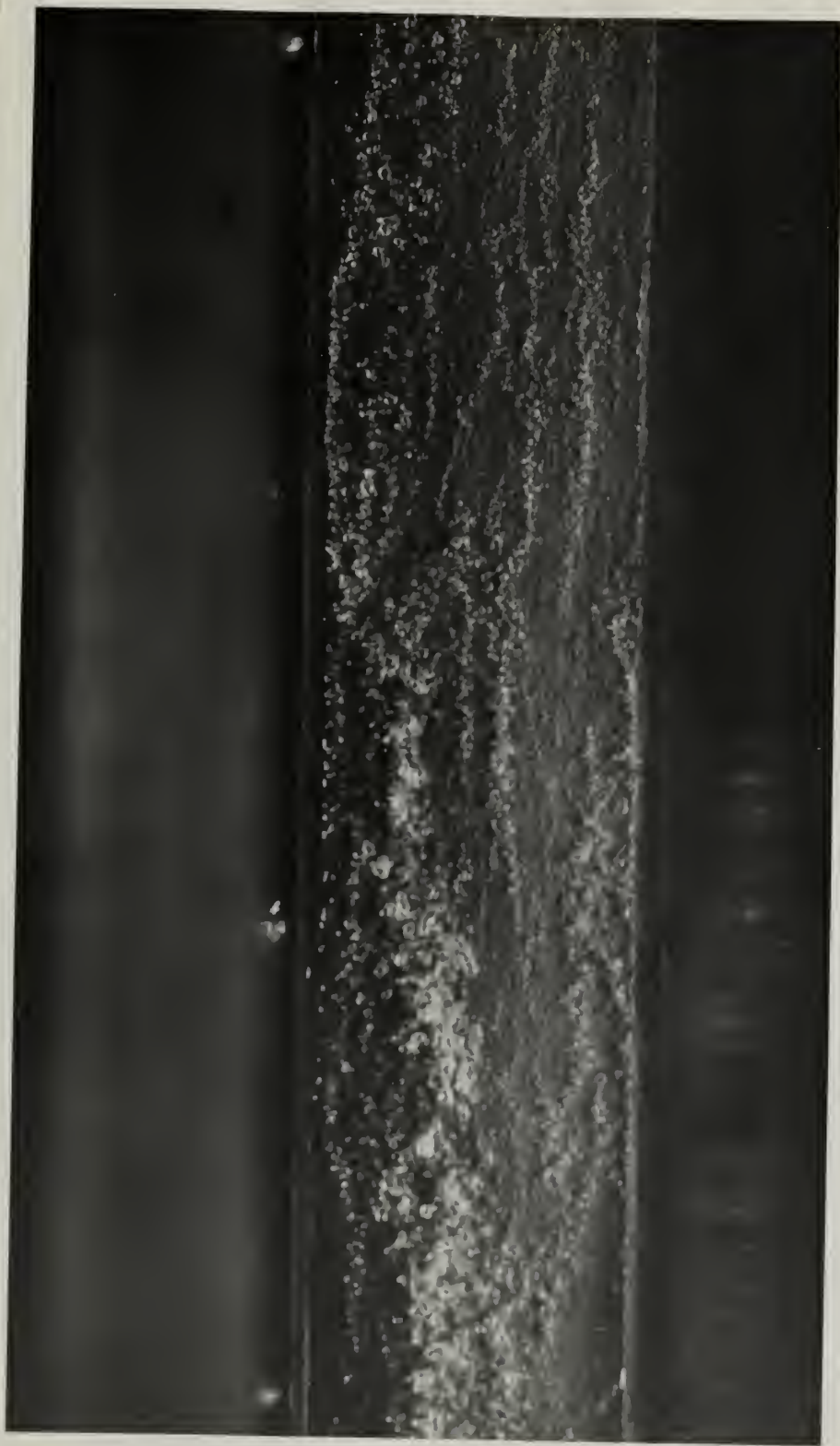


Case	Pres. psia	veloc ft/sec	sub-cooling °F	$\Delta T_A$ / $\mu^2 \cdot \text{min}$
C	500	20	92	$2.04 \times 10^6$
J	1000	20	70	$1.258 \times 10^6$
N	1000	20	20	$1.035 \times 10^6$
Q	1000	30	95	$2.57 \times 10^6$
S	1000	30	97	$1.90 \times 10^6$
Z	1500	20	144	$3.70 \times 10^6$

Figure (1)







Case	Pressure	Velocity	Subcooling	$q/A$
C	500 psia	20 ft/sec.	92 °F	$2.04 \times 10^6$ BTU/hr-ft <sup>2</sup>

Figure (2)

BTR 1/23 - Jt 5

2.00 = 100

40° F

207/rec.

200 pair

0

1/23 (2)

Temperature

Velocity

Pressure

Flow



Case	Pressure	Velocity	Subcooling	Q/A
J	1000 psia	20 ft/sec.	70°F	$1.258 \times 10^6$ BTU/hr-ft <sup>2</sup>

Figure (3)



1  
 2000 psia  
 10 ft/sec  
 100 F  
 BTU/hr-ft<sup>2</sup>

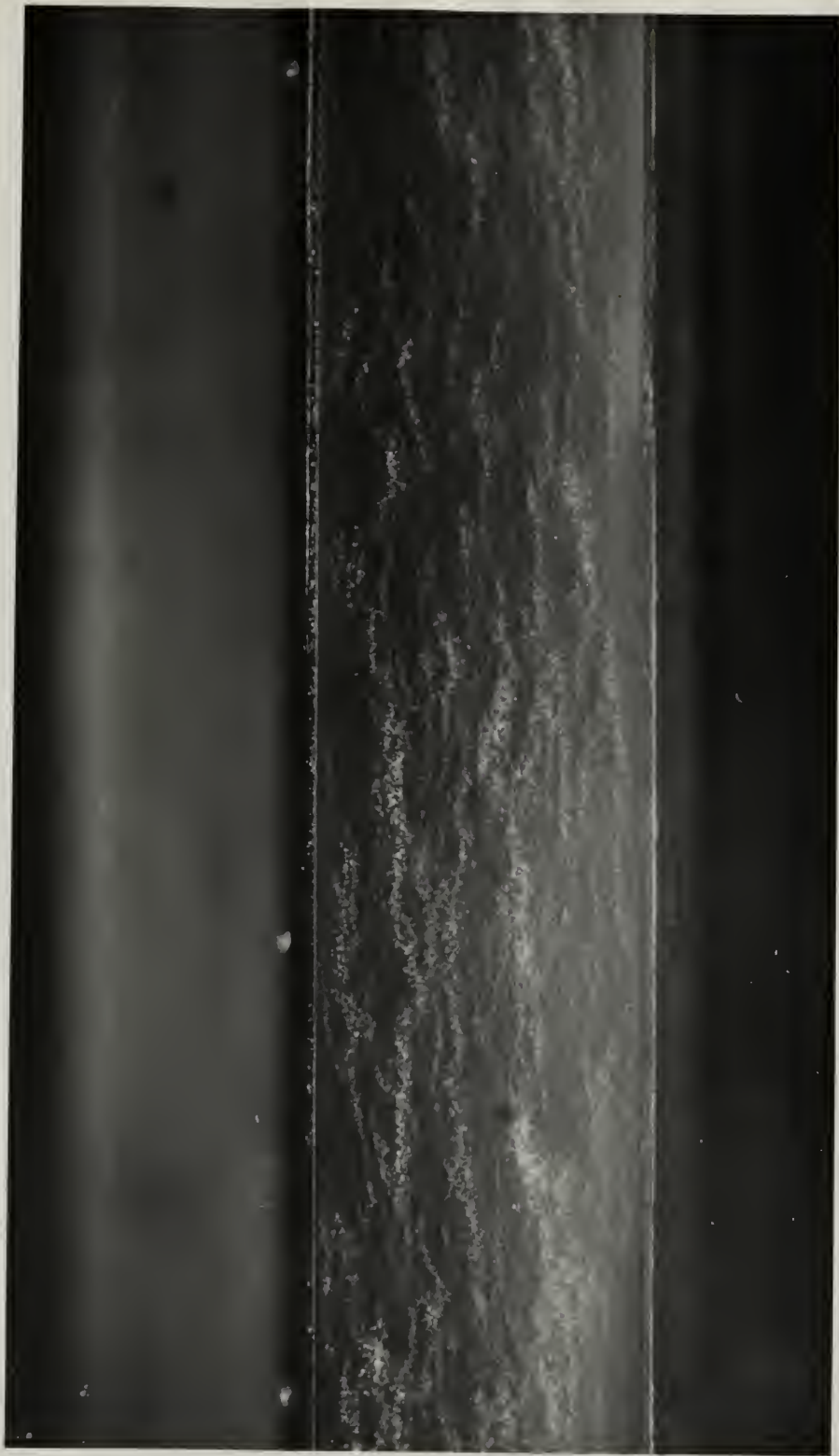


Case	Pressure	Velocity	Subcooling	$\dot{q}$
H	1000 <i>psia</i>	20 <i>ft/sec.</i>	20° <i>F</i>	$1.035 \times 10^6$ <i>BTU/hr-ft<sup>2</sup></i>

Figure (4)



BTM/ln - ft  
1000 ft  
1000 ft/sec  
1000 ft/sec  
1000 ft/sec



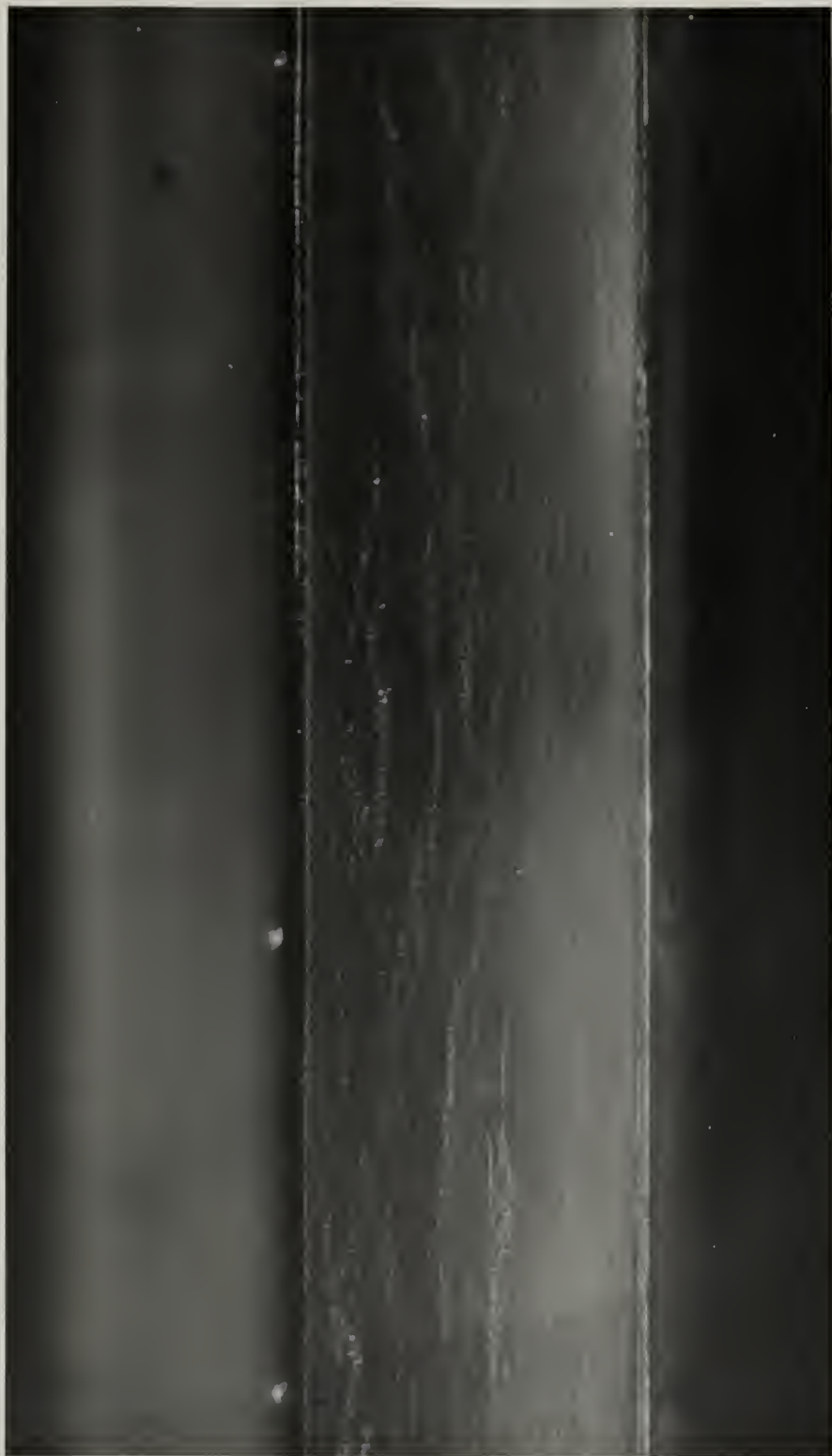
Case	Pressure	Velocity	Subcooling	$q/A$
2	1000 psia	30 ft/sec.	95 °F	$2.57 \times 10^6$ BTU/hr-ft <sup>2</sup>

Figure (5)





1000 ft/sec  
 30 ft/sec  
 30° F  
 BTU/hr-ft<sup>2</sup>

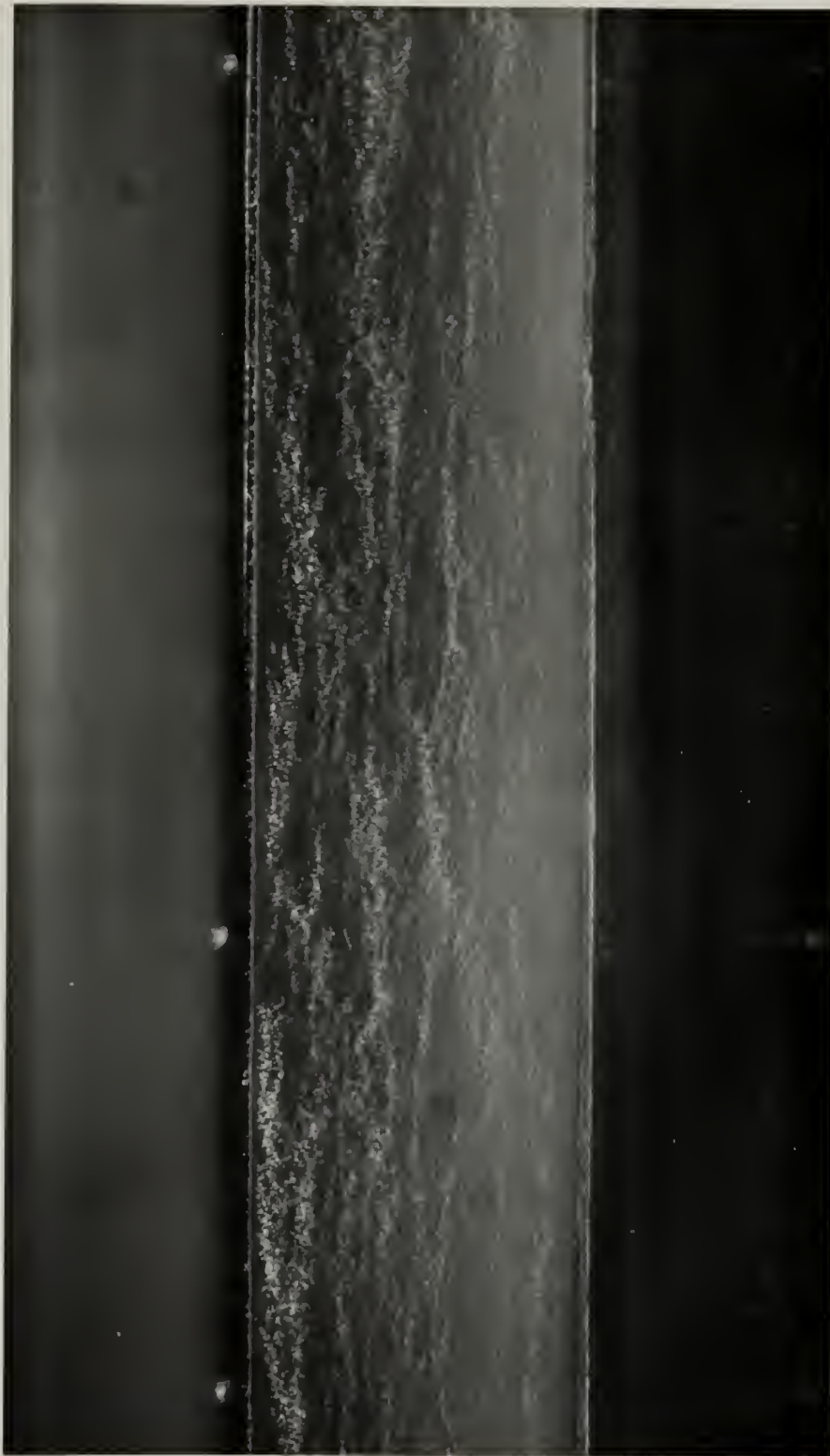


Case	Pressure	Velocity	Subcooling	$q/A$
S	1000 psia	30 ft/sec.	97 °F	$1.90 \times 10^6$ BTU / Hr-ft <sup>2</sup>

Figure (6)



Date	Location	Velocity	Direction	Wind
3	1000 ft	20 ft/sec	SE	1-10 m 100
				BTR / 100 ft



Case	Pressure	Velocity	Subcooling	$\dot{Q}/A$
2	1500 psia	20 ft/sec.	144 °F	$2.70 \times 10^6$ BTU/lv-ft <sup>2</sup>

Figure (7)



BTW / m - ft  
2010 - 2011  
11.0 F  
ft/sec  
1000  
1000

The pictures appear to show considerable variations in size and numbers of bubbles with changes in pressure, velocity, subcooling, and rate of heat transfer. At first glance, the curves of figure (1) show similar degrees of variation. However, the really striking thing about the curves is that they all tend to peak at about 2.86 mils diameter. All of the data in appendix I has been plotted and the diameter at which the peak occurs noted. This is presented in Table I, along with the deviations from a mean value of 2.86 mils. Bearing in mind that the resolution used in measuring bubble diameter was about 2.5 mils, it appears that, for the experimental cases investigated, the majority of bubbles at any instant are in the approximate range of 2.1 to 3.6 mils diameter.

In reference (G-1), Gunther made a photographic study of boiling under forced convection conditions at various flow rates, degrees of subcooling, heat fluxes, but at essentially atmospheric pressure. He concluded that bubble size increased with decreasing velocity, with decreasing subcooling, and moderately increased with decreasing heat transfer rate. His conclusions were made on the measurement of relatively few bubbles. His conclusions do not appear to be borne out by the more exhaustive measurements presented in this paper.

Gunther, in the same paper (G-1), provided several graphs of bubble size as a function of time. A curve of the general shape of Gunther's is given in figure (8). Blocking it off in equal increments of size (which is the way the bubbles were measured in the data presented) makes it quite obvious that a

The present report is a preliminary investigation in  
 this and nature of copper with changes in pressure, velocity,  
 and rate of heat transfer. It is the purpose of this  
 series of papers (I) to show similar changes of pressure,  
 velocity and heat transfer being shown in this paper  
 all data are taken at about 1.50 m/sec. All of the data  
 in this paper have been taken at the same rate of heat  
 transfer. This is because in this I, along with  
 the velocity from a mean value of 1.50 m/sec. Being in this  
 that the velocity was in increasing order of pressure and heat  
 transfer. It is shown that, for the experimental conditions  
 listed, the majority of copper at any instant are in the  
 same range of 1.5 to 2.0 m/sec.

In reference (1-1), Goulet and a preliminary study of  
 boiling under forced convection conditions at various flow rates,  
 degree of subcooling, heat transfer, but an essentially same  
 degree of subcooling. It is concluded that boiling also increased with  
 increasing velocity, with decreasing subcooling, and subcooling  
 increased with decreasing heat transfer rate. The conditions  
 are made on the measurement of velocity, the boiling. The  
 conditions do not appear to be given by the data shown.  
 Five measurements presented in this report.

Finally, in the same paper (1-1), Goulet and a preliminary study  
 of boiling in a function of flow. I give of the results  
 shown of Goulet's is given in figure (3). It is shown in all in  
 equal increments of flow (which is the case in this paper)  
 measured in the present report. It is shown that a

TABLE 1

Case	Pressure psia	Velocity ft/sec	$Q/A$ BTU/ft <sup>2</sup> -hr	Subcooling °F	Diameter mils	Deviation mils
A	500	20	$2.34 \times 10^6$	117	2.8	-.06
B	500	20	$2.02 \times 10^6$	117	2.8	-.06
C	500	20	$2.04 \times 10^6$	92	3.2	+ .34
D	500	20	$1.705 \times 10^6$	92	3.0	+ .14
E	500	20	$1.43 \times 10^6$	92	2.9	+ .04
F	1000	20	$2.155 \times 10^6$	95	3.4	+ .54
G	1000	20	$1.821 \times 10^6$	95	3.0	+ .14
H	1000	20	$1.516 \times 10^6$	95	2.7	-.16
I	1000	20	$1.54 \times 10^6$	70	3.5	+ .64
J	1000	20	$1.258 \times 10^6$	70	2.8	-.06
K	1000	20	$1.285 \times 10^6$	45	3.0	+ .14
L	1000	20	$1.022 \times 10^6$	45	4.0	+1.14
M	1000	20	$.799 \times 10^6$	45	3.2	+ .34
N	1000	20	$1.035 \times 10^6$	20	3.2	+ .34
O	1000	20	$.810 \times 10^6$	20	3.0	+ .14
P	1000	20	$.610 \times 10^6$	20	3.0	+ .14
Q	1000	30	$2.57 \times 10^6$	95	2.5	-.36
R	1000	30	$2.22 \times 10^6$	97	2.0	-.86
S	1000	30	$1.90 \times 10^6$	97	3.0	+ .14
T	1000	30	$1.832 \times 10^6$	70	2.4	-.46
U	1000	30	$1.545 \times 10^6$	70	2.6	-.26
V	1000	30	$.846 \times 10^6$	20	2.6	-.26
W	1000	30	$.614 \times 10^6$	20	2.6	-.26
X	1000	30	$.452 \times 10^6$	10	2.6	-.26
Y	1000	30	$.321 \times 10^6$	7	2.7	-.16



TABLE 1

Case	Velocity Feet/sec	Distance Feet	Time Sec	Velocity Feet/sec	Distance Feet	Time Sec
1	1000	100	0.1	1000	100	0.1
2	1000	200	0.2	1000	200	0.2
3	1000	300	0.3	1000	300	0.3
4	1000	400	0.4	1000	400	0.4
5	1000	500	0.5	1000	500	0.5
6	1000	600	0.6	1000	600	0.6
7	1000	700	0.7	1000	700	0.7
8	1000	800	0.8	1000	800	0.8
9	1000	900	0.9	1000	900	0.9
10	1000	1000	1.0	1000	1000	1.0
11	1000	1100	1.1	1000	1100	1.1
12	1000	1200	1.2	1000	1200	1.2
13	1000	1300	1.3	1000	1300	1.3
14	1000	1400	1.4	1000	1400	1.4
15	1000	1500	1.5	1000	1500	1.5
16	1000	1600	1.6	1000	1600	1.6
17	1000	1700	1.7	1000	1700	1.7
18	1000	1800	1.8	1000	1800	1.8
19	1000	1900	1.9	1000	1900	1.9
20	1000	2000	2.0	1000	2000	2.0
21	1000	2100	2.1	1000	2100	2.1
22	1000	2200	2.2	1000	2200	2.2
23	1000	2300	2.3	1000	2300	2.3
24	1000	2400	2.4	1000	2400	2.4
25	1000	2500	2.5	1000	2500	2.5
26	1000	2600	2.6	1000	2600	2.6
27	1000	2700	2.7	1000	2700	2.7
28	1000	2800	2.8	1000	2800	2.8
29	1000	2900	2.9	1000	2900	2.9
30	1000	3000	3.0	1000	3000	3.0

TABLE I (continued)

Case	Pressure psia	Velocity ft/sec	$Q/A$ BTU/ft <sup>2</sup> -hr	Subcooling °F	Diameter mils	Deviation mils
Z	1500	20	$2.70 \times 10^6$	144	2.7	-.16
AA	1500	20	$2.27 \times 10^6$	146	2.5	-.36
BB	1500	20	$1.97 \times 10^6$	94	2.6	-.26
CC	1500	20	$1.669 \times 10^6$	69	2.7	-.16



### Typical Bubble Growth Curve

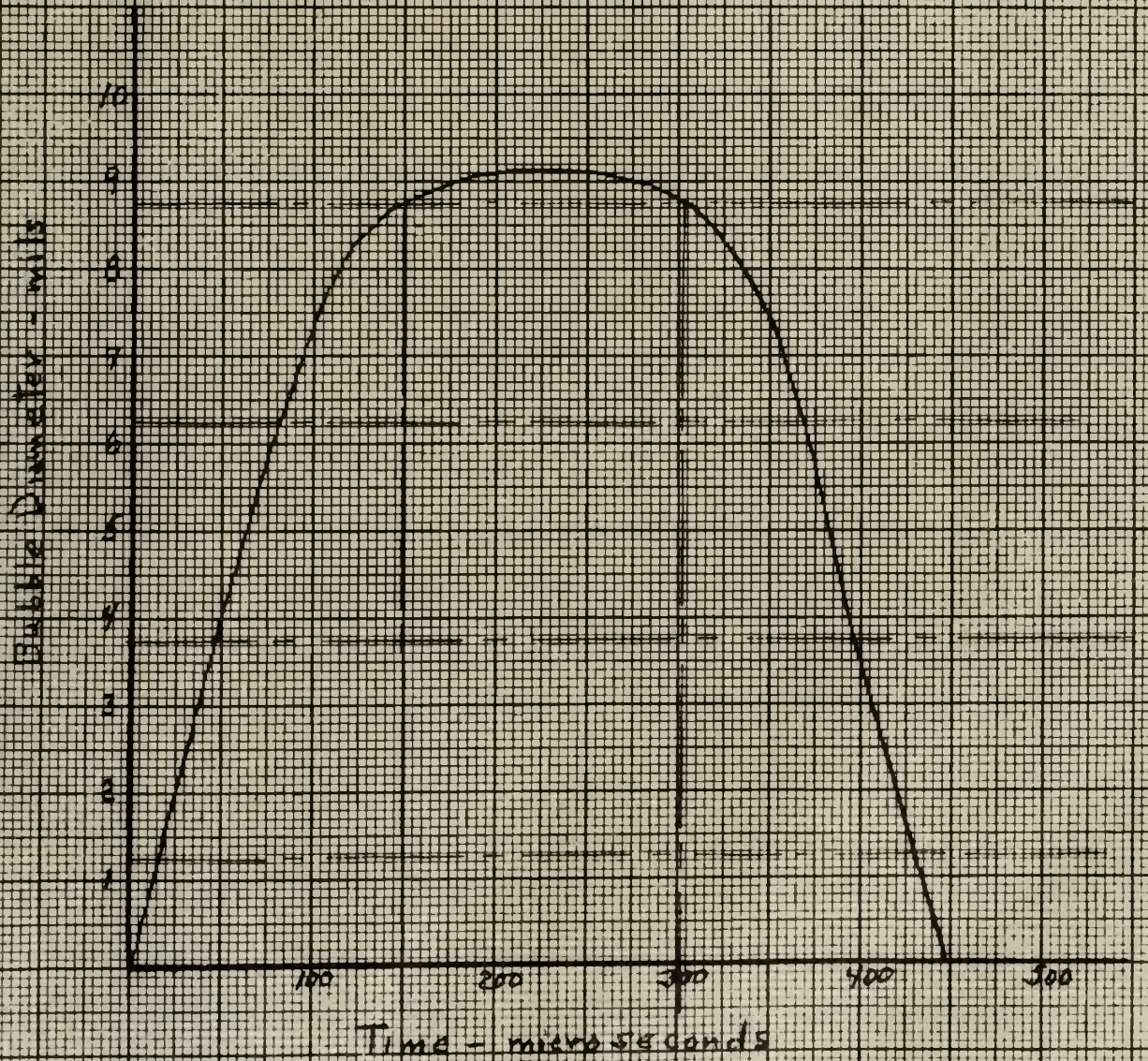


Figure (B)



bubble spends the majority of its life time at or nearly at its maximum size. Therefore, a picture which depicts a one or two microsecond instant will catch most of the bubbles at their maximum size. The data presented is really proportional to the maximum sizes attained by the bubbles. The peaks at a bubble size of 2.86 mils represent fully developed bubbles and are not the result of a picture catching a great number of bubbles in a growth or decay phase.

There are, for many cases, large numbers of relatively big bubbles counted. A close look at figure (4) suggests the cause. What were counted as large bubbles were most likely mergers of several smaller bubbles. The very fact that even in such a case at that in figure (4) - case H - the bubble count shows a peak in the vicinity of 2.86 mils tends to support this theory.

In summary, the data suggests that for highly turbulent flow of cold water over a heated surface, bubbles prefer to grow to one particular size. That this particular size is a constant for a wide range of conditions and rates of heat transfer is also strongly suggested.

### 3. Proposed Model for the Formation of a Bubble.

#### a. Temperature distribution.

In the experiments under consideration, highly turbulent water is flowing over a heated surface. Much work has been done in studying the velocity distribution for isothermal turbulent flow and is concisely summarized by McAdams, reference (M-3). Generally speaking, three regions of flow are usually described; a laminar sublayer next to the surface in which the flow is



streamline, a buffer layer in which the motion of the water oscillates between streamline and turbulent flow, and the bulk of the fluid in which the flow is completely turbulent. Experimental work, particularly by Nakuradse, has led to equations for the velocity distribution. From these, equations for the eddy diffusivity of momentum can be written. Now if we consider that the boundary is heated, we can say that heat is conducted through the laminar boundary layer by molecular conduction, through the buffer layer by both molecular conduction and mechanical mixing, and through the bulk by mechanical mixing. Equations similar to those for eddy diffusivity of momentum can be written for the eddy diffusivity of heat. The success of Martinelli and others in utilizing this analogy leads to the prediction that the temperature distribution through the water is very nearly the same as the velocity distribution. For the purpose of this paper we will consider two regions of flow and their temperature distribution. We will assume that in the laminar sublayer the temperature has a linear distribution varying from the boundary temperature to the bulk temperature, and that the bulk temperature is constant.

b. Adiabatic formation of a bubble.

We will assume that the bubble will originate at a point on a heated surface. Available to this point is the energy from the water around it. The model we propose is that a hemisphere of water on the heated surface with a temperature distribution as given in section 2. a. change adiabatically from this state to a state composed of part vapor and part liquid, all now at saturation temperature corresponding to the pressure



... a further layer is added to the water ...  
 ... between strata and turbulent flow, and the bulk ...  
 ... of the fluid in which the flow is completely turbulent ...  
 ... particularly by Reynolds, has led to equations ...  
 ... for the velocity distribution. These equations for the ...  
 ... eddy diffusivity of momentum can be written. Now if we consider ...  
 ... that the boundary is heated, we can say that heat is conducted ...  
 ... through the laminar boundary layer by molecular conduction, ...  
 ... through the latter layer by both molecular conduction and eddy ...  
 ... heat mixing, and through the bulk of turbulent mixing. ...  
 ... these mixing is done for eddy diffusivity of momentum and for ...  
 ... written for the eddy diffusivity of heat. The velocity of ...  
 ... itself and hence in utilizing this analogy leads to the ...  
 ... distribution that the temperature distribution through the water is ...  
 ... very nearly the same as the velocity distribution. For the ...  
 ... case of this paper we will consider two regions of flow and their ...  
 ... temperature distribution. We will assume that in the laminar ...  
 ... region the temperature has a linear distribution except near ...  
 ... the boundary temperature is the bulk temperature, and that the ...  
 ... bulk temperature is constant.

C. D. Laminar Flow in a Pipe.

... we will assume that the fluid will exit at a point ...  
 ... on a heated surface. ...  
 ... from the water around it. ...  
 ... heat of water at the heated surface with a temperature ...  
 ... below as given in section 2. ...  
 ... to a state constant at any point and any time, all ...  
 ... has an uniform temperature corresponding to the pressure

of the system. There are several reasons for choosing this model. First, bubbles are known to originate on surfaces. One of the reasons for this is that the presence of minute curvatures on the surface lessen the energy requirements for bubble formation. This is in accordance with Gibbs requirement for thermo dynamic equilibrium as regards pressures of a two phase system:

$$p'' - p' = (C_1 + C_2) \sigma$$

Secondly, as pointed out in section 3.1, there is almost certainly a slow moving layer of water next to the boundary through which a temperature gradient probably exists. Thirdly, a hemisphere is a reasonable shape as there appears no reason to suppose a preferential direction for the flow of energy to the selected point. In the fourth place, since the formation of a bubble is known to be a rapid process, it can well be assumed to be an adiabatic one. And lastly, it seems correct to state that at its maximum size, a bubble will have attained thermal equilibrium with its surrounding.

#### 4. Application of Gibbs Stability Criterion to Model.

In the proposed model, water existing in one state suddenly ceases to exist in this state and appears in an entirely different state. According to Gibbs, reference (G-2): "For the equilibrium of any isolated system it is necessary and sufficient that in all possible variations of the state of the system which do not alter its energy, the variation of its entropy shall either vanish or be negative." Conversely, if our model is a possible one, the entropy of the water in its second state must be greater than its entropy in the first state.

of the system. There are several reasons for choosing this model. First, models are known to originate in nature. One of the reasons for this is that the process of natural selection on the system favors the energy requirements for the system. This is in accordance with Gibbs requirements for the system equilibrium as regards pressure of a gas phase system.

$$p' - p = (C_1 + C_2) \sigma$$

Secondly, as pointed out in section 3.1, there is almost certainly a thin surface layer of water next to the boundary through which a transport process probably takes place. Thirdly, a hypothesis is a reasonable one in these systems no reason to suppose a potential difference between the flow of water to the surface. In the same place, since the formation of a nucleus is known to be a rapid process, it can well be assumed to be an equilibrium one. The latter is seen to occur in nature that is for natural water, a nucleus will have reached thermal equilibrium with the surrounding water.

4. Application of Gibbs stability criterion to model

In the proposed model, water existing in two states undergoes a transition to exist in this state and appears in an entirely different phase. According to Gibbs, criterion (4-5) for the stability of any isolated system is necessary and sufficient that in all possible variations of the state of the system which do not alter the energy, the volume or the entropy, the system either remains in its original state or it tends to a state of lower energy. The energy of the system in the two states is given by the entropy in the two states.

Gibbs criterion certainly implies that the most probable state of an isolated system is the one for which its entropy is greatest. There seems no reason why this cannot be interpreted to say that any non-equilibrium state of a system can spontaneously shift to a state of higher entropy. There seems no reason to require that the second or new state be an equilibrium state.

We have taken for our system the mass of a hemisphere of water on the heated surface. Such a system, as it occurs in the experimental setup, is one of an infinite number of such systems. It is not an isolated system. However, if we assume that during the transition from one state to another, the system suffers no gain or loss of energy, then we can still apply Gibbs criterion.

We can write this assumption as an equation

$$1.) \quad dE - dQ + dW = 0$$

Since we have postulated an adiabatic change of state:

$$2.) \quad dE + dW = 0$$

or

$$3.) \quad dA = 0$$

For the transition to take place,

$$4.) \quad M_g S_g + M_f S_f - \int_0^M c_p \ln T dm \geq 0$$

or the entropy of the vapor plus the entropy of the liquid in the second state must be greater than the entropy of the liquid in the first state. For simplicity, we have neglected the entropies of the interfaces.

These criteria certainly implies that the most probable state of an isolated system is the one for which the entropy is greatest. There seems no reason why this cannot be interpreted to say that any non-equilibrium state of a system can spontaneously shift to a state of higher entropy. There seems no reason to require that the second or new state be an equilibrium state.

We have taken for our system the case of a mechanical system as the bested system. Such a system, as it occurs in the experimental setup, is one of an infinite number of such systems. It is not an isolated system. However, if we assume that during the transition from one state to another, the system suffers no gain or loss of energy, then we can still apply Gibbs criterion.

We can write this assumption as an equation

$$1.) \quad dE - dQ + dW = 0$$

Since we have postulated an adiabatic change of state:

$$2.) \quad dE + dW = 0$$

$$3.) \quad dH = 0$$

For the transition to take place,

$$4.) \quad M_2^2 + M_1^2 - \int_0^M p \ln T dm \geq 0$$

or the entropy of the vapor plus the entropy of the liquid in the second state must be greater than the entropy of the liquid in the first state. If equality, we have neglected the entropy of the interface.

From our model we can write the temperature gradient:

$$5.) \quad T = T_w - \left( \frac{T_w - T_s}{r_s} \right) z$$

Since our system is a mass of water we can write:

$$6.) \quad M_g + M_f = \int_0^R dm = M = \frac{2}{3} \pi \rho_f R^3$$

The enthalpies of the two states being equal gives equation

$$7.) \quad M_g h_g + M_f h_f = \int_0^M h dm = \int_0^R \rho_f c_p \left( T_w - \left( \frac{T_w - T_s}{r_s} \right) z \right) (R^2 - z^2) dz$$

$$8.) \quad M_g h_g + M_f h_f = \pi \rho_f c_p \left( \frac{2}{3} T_w R^3 - \frac{1}{4} \left( \frac{T_w - T_s}{r_s} \right) R^4 \right)$$

Solving equations (6) and (8) for  $M_g$  gives:

$$9.) \quad M_g = \frac{\frac{2}{3} \pi h_f \rho_f R^3 - \pi \rho_f c_p \left( \frac{2}{3} T_w R^3 - \frac{1}{4} \left( \frac{T_w - T_s}{r_s} \right) R^4 \right)}{h_f - h_g}$$

We can rewrite equation (1) as

$$10.) \quad \Delta S = (M - M_f) s_g + M_f s_f - \int_0^M s dm$$

$$11.) \quad \Delta S = -M_f s_{fg} + \int_0^M (s_g - s) dm$$

$$12.) \quad \Delta S = M_g s_{fg} - \int_0^M (s - s_f) dm$$

$$13.) \quad \Delta S = M_g s_{fg} - \int_0^M c_p \ln T/T_s dm$$

$$14.) \quad \int_0^M c_p \ln T/T_s dm = \int_0^R \rho_f c_p \pi \left[ \ln \left( \frac{T_w}{T_s} - \left( \frac{T_w - T_s}{T_s r_s} \right) z \right) \right] (R^2 - z^2) dz$$

Rearranging and using the series expansion for  $\ln(x)$  we can write the integral as:

$$T = T_w - T \left( \frac{T_w - T}{T} \right)^2 \quad (2)$$

$$M = M_1 + M_2 = \int_0^R \rho \pi r^2 dr = \frac{2}{3} \pi R^3 \rho \quad (3)$$

$$M_1 = M_2 + M_1 = \int_0^R \rho \pi r^2 dr = \frac{2}{3} \pi R^3 \rho \quad (4)$$

$$M_1 = M_2 + M_1 = \int_0^R \rho \pi r^2 dr = \frac{2}{3} \pi R^3 \rho \quad (5)$$

$$M_0 = \frac{\frac{2}{3} \pi R^3 \rho \left( \frac{T_w - T}{T} \right)^2 - \frac{1}{4} \left( \frac{T_w - T}{T} \right)^2}{N_1 - N_2} \quad (6)$$

$$\Delta z = (M - M_1) z_1 + M_1 z_2 - \int_0^M z^2 dm \quad (7)$$

$$\Delta z = -M_1 z_1 + M_1 z_2 + \int_0^M (z_2 - z) dm \quad (8)$$

$$\Delta z = M_1 z_2 - \int_0^M (z - z_2) dm \quad (9)$$

$$\Delta z = M_1 z_2 - \int_0^M \rho \pi r^2 dr \quad (10)$$

$$\int_0^M \rho \pi r^2 dr = \int_0^R \rho \pi r^2 dr \left( \frac{T_w - T}{T} \right)^2 - \frac{T_w - T}{T} \left( \frac{T_w - T}{T} \right)^2 \quad (11)$$

Deriving the mass of the system using the integral we

can write the integral as:

$$15.) \int_0^R \rho_f c_p \pi (R^2 - z^2) \left[ \left( \frac{T_w - T_s}{T_s} \right) \left( 1 - \frac{z}{x_s} \right) - \frac{1}{2} \left( \frac{T_w - T_s}{T_s} \right)^2 \left( 1 - \frac{z}{x_s} \right)^2 + \dots \right] dz$$

If this integral is worked out it is found that the first term in the brackets produces the true value of the integral almost exactly for all values of  $R$  less than  $3x_s$ , which turns out to include all systems having positive  $\Delta S$ . Therefore we will write the integral

$$16.) \int_0^R \rho_f c_p \pi (R^2 - z^2) \left[ \left( \frac{T_w - T_s}{T_s} \right) \left( 1 - \frac{z}{x_s} \right) \right] dz$$

This is readily integrated to:

$$17.) \rho_f c_p \pi \left( \frac{T_w - T_s}{T_s} \right) \left( \frac{2}{3} R^3 - \frac{R^4}{4x_s} \right)$$

Combining this result with equation (6) we can write:

$$18.) \Delta S = \left( 1 - \frac{1}{T_s} \right) (T_w - T_s) \left[ \frac{s_{fg} \pi \rho_f c_p}{h_{fg}} \right] [R^3] \left[ \frac{2}{3} - \frac{1}{4} \frac{R}{x_s} \right]$$

The first term is almost exactly one and we can write:

$$19.) \Delta S = (T_w - T_s) \left[ \frac{s_{fg} \pi \rho_f c_p}{h_{fg}} \right] [R^3] \left[ \frac{2}{3} - \frac{1}{4} \frac{R}{x_s} \right]$$

$$20.) \Delta S = \left( \frac{T_w - T_s}{T_s} \right) (\pi \rho_f c_p) (R^3) \left( \frac{2}{3} - \frac{1}{4} \frac{R}{x_s} \right)$$

Throughout this derivation we have assumed that the density,  $\rho_f$ , and the heat capacity,  $c_p$ , of the liquid in its initial state were constants. This is reasonable if the wall temperature,  $T_w$ , is not very much greater than the saturation temperature,  $T_s$ .

If we consider a situation in which the wall temperature is constant, above saturation temperature, but below that which causes vigorous boiling, the term  $x_s$  which appears in equation



$$22.1) \int_{\frac{1}{2} \pi}^{\pi} \cos \pi (R - \frac{1}{2} \tau) \left[ \left( \frac{\tau - \tau_2}{2} \right) - \left( \frac{\tau - \tau_1}{2} \right) + \dots \right] d\tau$$

If the interval is chosen such that the value of the integrand is the same throughout the interval, the value of the integral is almost exactly the same for all values of  $\tau$  less than  $\frac{1}{2} \pi$ . Therefore we can to include all values having positive  $\Delta z$ . Therefore we will write the integral

$$22.2) \int_{\frac{1}{2} \pi}^{\pi} \cos \pi (R - \frac{1}{2} \tau) \left[ \left( \frac{\tau - \tau_2}{2} \right) - \left( \frac{\tau - \tau_1}{2} \right) \right] d\tau$$

This is readily integrated to

$$22.3) \int_{\frac{1}{2} \pi}^{\pi} \cos \pi (R - \frac{1}{2} \tau) \left( \frac{\tau - \tau_2}{2} - \frac{\tau - \tau_1}{2} \right) d\tau$$

Comparing this result with equation (2) we can write

$$22.4) \Delta z = \left( \frac{\tau_1 - \tau_2}{2} \right) \cos \pi (R - \frac{1}{2} \tau) \left[ \left( \frac{\tau_1}{2} - \frac{1}{4} \right) - \left( \frac{\tau_2}{2} - \frac{1}{4} \right) \right]$$

The first term in equation (22.4) is the same as the result

$$22.5) \Delta z = \left( \frac{\tau_1 - \tau_2}{2} \right) \cos \pi (R - \frac{1}{2} \tau) \left[ \left( \frac{\tau_1}{2} - \frac{1}{4} \right) - \left( \frac{\tau_2}{2} - \frac{1}{4} \right) \right]$$

$$22.6) \Delta z = \left( \frac{\tau_1 - \tau_2}{2} \right) \cos \pi (R - \frac{1}{2} \tau) \left( \frac{\tau_1}{2} - \frac{1}{4} - \frac{\tau_2}{2} + \frac{1}{4} \right)$$

Throughout this derivation we have assumed that the density  $\rho$  and the wave velocity  $v$  of the liquid in the capillary tube were constant. This is reasonable if the wall thickness is not very much greater than the wavelength. It is of course a simplification to assume that the density is constant, since variations in density, but since this effect cannot be taken into account, we will assume in equation

(20) must be a constant. For a given system, a given mass of water, a given hemisphere of water, a given  $R$  in equation (20) we will get one value of  $\Delta S$ . If this value is positive we can say that the given system is in an unstable state. If we plot this function against system size,  $R$ , we get the curve of figure (9). This has been normalized.

##### 5. Interpretation of Result of Applying Gibbs Criterion.

Bearing in mind the model chosen, we can say that whenever there is a temperature gradient through the water, there exists a most unstable system. This is the peak point in figure (9). The model and equation (20) cannot be interpreted to give the conditions necessary for a change of state. They should be interpreted as follows. If the gradient through the water is changing, one particular system has a much greater probability of changing state before any other system does. That is, one particular system (the one for which  $R = 2x_g$ ) will reach critical instability first. Critical instability is probably determined by such factors as roughness of surface, purity of water, wettability of surface, work to produce interfaces, both solid-vapor and liquid-vapor, and many others.

Equation (9) can be differentiated with respect to  $R$  and, it is found that  $R = 2x_g$  is the necessary condition, for the differentiated equation to equal zero. Thus the system that first reaches critical instability is also the system that produces the greatest mass of vapor when it changes state. If we substitute  $R = 2x_g$  in equation (9) and let

(20) shall be a constant, for a given system, a given case of  
 view, a fixed relationship of bodies, a given  $\theta$  is specified (20)  
 we will not be able to find it.  $\Delta$  is a constant of the  
 system and given system in its own right. It is the  
 only function of the system,  $\Delta$ , to the right of the  
 line (20). This is the case.

2. Investigation of limits of applying the system

Seeing in mind the model chosen, we can say that man-  
 ever there is a background problem through the view, there  
 exists a real analysis system. This is the best kind of limit  
 (21). The model and equation (20) cannot be interpreted to give  
 the conditions necessary for a system of state. They should be  
 interpreted as follows. If the system is given for view it  
 changes, one condition is given for a real analysis system  
 of change, state state and other system data, this is, the  
 particular system (the one for which  $\Delta = 2\Delta$ ) will never be  
 local analysis time. Global analysis is possible in  
 terms of two factors as mentioned of various, partly of view,  
 possibility of various, and in general, analysis, with solid  
 view and light-rays, and any detail.  
 Equation (21) can be interpreted with respect to  $\Delta$  and  
 is in fact that  $\Delta = 2\Delta$  is the necessary condition for the  
 differential equation to hold true. Thus the system that  
 first becomes critical analysis is also the system that  
 does the greatest work of view that it should be able. It is  
 possible  $\Delta = 2\Delta$  in equation (21) and (22)



Figure (9)



$$21.) \quad M_g = \frac{4}{3} \pi \rho_g r^3$$

we arrive readily at the result

$$22.) \quad x_s^3 = \frac{h_{fg} \rho_g}{c_p \rho_f} \frac{r^3}{(T_w - T_s)}$$

From the discussion in section 2, it appears that we could substitute for  $r$  the value 1.43 mils, and the equation becomes

$$23.) \quad x_s^3 = \frac{h_{fg} \rho_g}{c_p \rho_f} \frac{3.92}{(T_w - T_s)}$$

The interpretation to be placed on this equation is as follows. If we can satisfy the equation, boiling will occur. The quantity  $x_s$  is dependent on the thickness of the laminar sublayer, which is determined by fluid flow mechanics if there is no boiling. Thus some solutions of equation (23) will just not be possible. That is, if  $(T_w - T_s)$  is a very small number, the  $x_s$  required by the equation may be so great as to place it outside the laminar sublayer.

In the experiments being considered it has been found, reference (H-1), that  $(T_w - T_s)$  is approximately  $15^\circ$  F at a pressure of 1000 psia. The value of  $h_{fg} \rho_g / c_p \rho_f$  at this pressure is 2.9 in the appropriate units. Therefore, the value of  $x_s$  required is approximately 0.91 mils. This must have been a realizable value. It seems a reasonable one.

The action of boiling is generally considered to increase the turbulence near the heated surface. This added turbulence or mixing is felt to account for the greatly increased heat transfer. As the vapor moves away from the surface, liquid from the

$$M_2 = \frac{1}{2} \pi r^2 \gamma \quad (21)$$

we arrive readily at the result

$$\gamma = \frac{N_2 r^2}{C_p r^2} \frac{1}{(T_2 - T_1)} \quad (22)$$

From the discussion in section 2, it appears that we could substitute for  $r$  the value  $1.4 \mu$ , and the equation becomes

$$\gamma = \frac{N_2 r^2}{C_p r^2} \frac{1}{(T_2 - T_1)} \quad (23)$$

The investigation to be placed on this equation is as follows. It was really the equation, boiling will occur. The value  $\gamma$  is dependent on the thickness of the liquid surface, which is determined by liquid film thickness if there is no boiling. Thus some addition of equation (23) will just not be possible. That is, if  $(T_2 - T_1)$  is a very small number, the  $\gamma$  required by the equation may be so great as to place it outside the liquid surface.

In the experiments being considered it has been found, between  $(T_2 - T_1)$ , that  $(T_2 - T_1)$  is approximately  $10^3$  of a pressure of 1000 bars. The value of  $N_2 r^2 / C_p r^2$  at this pressure is 2.9 in the approximate value. Therefore, the value of  $\gamma$  required is approximately 0.41 atm. This may have been a real value. It seems a reasonable one.

The action of boiling is generally considered to involve the condensation of the vapor surface. This added temperature of mixing is this to account for the greatly increased heat transfer. It is the vapor space away from the surface, liquid from the

surrounding, replaces it. Thus although there can be no real laminar sub-layer once boiling has commenced, we can feel confident that there is always water next to some portions of the surface and this water is being heated by molecular conduction. We can think and speak of an effective laminar sub-layer. As the heat supplied to a surface is increased, the amount of bubbles increases but the surface temperature remains constant. There is a limit, however. When the bubbling becomes sufficiently vigorous, the heat transfer rate suddenly goes down and the temperature of the surface goes up until the surface melts. Another way of looking at equation (23) would be to say that only a very, very thin laminar sub-layer is necessary to keep the surface temperature no more than  $15^{\circ}$  above the saturation temperature at 1000 psia. Now we suppose the bubbling to get so vigorous as to reduce the thickness of even this layer. The value of the critical unstability is still the same, however, and to produce it a steeper gradient will be required. The wall temperature will rise and the bubbling continue and become more vigorous. Once started this process almost instantly proceeds to burn out.

Perhaps the most profitable use that this equation (22) can be put to is in connection with transients. If the temperature of a surface rises very rapidly with time, the temperature distribution in the water is obtained from the solution of the diffusion equation, considering that no natural convection can take place in such a short time and considering the water as a semi infinite medium. If the temperature rises lin-



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early with time, the solution is of the form:

$$24.) \quad T = k t \left[ \left( 1 + \frac{z^2}{2Kt} \right) \operatorname{erfc} \frac{z}{2\sqrt{Kt}} - \frac{z}{\sqrt{\pi Kt}} e^{-\frac{z^2}{4Kt}} \right]$$

At some time both this equation and equation (22) will give the same value of  $x_g$ .  $x_g$  is the value of  $z$  in equation (24) for which  $T = T_g$ . At this time the water next to the heated surface will begin to boil.

### 6. Summary.

Although this paper has some numbers in it, only the qualitative ideas are of any importance. For a particular combination of surface and liquid some critical amount of instability is required in the liquid before vapor can form. If as in the case of highly turbulent flow of liquid over a heated surface, a temperature gradient exists through the liquid, one particular mass of the liquid will be the most likely to attain this degree of instability. Therefore, the bubbles of vapor formed will be of one size. If the amount of instability required is known, or if the particular bubble size is known, a condition for boiling can be written. This simply requires that the distance that the saturation temperature reach into the liquid be inversely proportional to the cube root of the degrees of superheat of the surface. However, neither the distance,  $x_g$ , nor the superheat are independent variables. The equation can only be thought of as a limiting condition. If the laminar sub-layer is thick, the degrees of superheat required for boiling is determined by the critical instability and the degrees



of superheat will remain constant until the layer is so reduced in thickness by bubble action that the surface temperature must rise to produce the critical unstability. If a surface is rising in temperature rapidly, then the equation can be interpreted to mean that boiling will not occur until the actual gradient in the water is the one required for the temperature of the surface at the moment.



## 7. Nomenclature

E - internal energy-BTU

Q - Heat quantity-BTU

W - Work quantity-BTU

H - Enthalpy-BTU

$M_g$  - Mass of saturated vapor-LBm

$s_g$  - Entropy per unit mass saturated vapor BTU/LBm- $^{\circ}$ R

$M_f$  - Mass of saturated liquid-LBm

$s_f$  - entropy per unit mass saturated liquid BTU/LBm- $^{\circ}$ R

M - Total Mass of system LBm

$c_p$  - Constant pressure specific heat BTU/LBm- $^{\circ}$ F

T - Temperature- $^{\circ}$ R

$T_w$  - Temperature of boundary (heated surface)  $^{\circ}$ R

$T_s$  - Temperature of saturated liquid  $^{\circ}$ R

$\chi_s$  - perpendicular distance from heated surface at which  
water temperature is  $T_s$  -mils

$z$  - perpendicular distance from heated surface -mils

R - Radius of hemisphere of liquid-mils

$s_{fg}$  -  $s_g - s_f$

$h_{fg}$  -  $h_g - h_f$

$h_f$  - enthalpy per unit mass of saturated liquid BTU/LBm

$h_g$  - enthalpy per unit mass of saturated vapor BTU/LBm

r - Radius of bubble - mils

$\rho_s$  - density of saturated liquid LBm/ft<sup>3</sup>

$\rho_g$  - density of saturated vapor LBm/ft<sup>3</sup>

$p''$  - Pressure of vapor

Table of Contents

E - External energy - 100

Q - Heat quantity - 100

W - Work quantity - 100

H - Enthalpy - 100

M<sub>g</sub> - Mass of saturated vapor - 100

z<sub>g</sub> - Enthalpy per unit mass saturated vapor - 100

M<sub>f</sub> - Mass of saturated liquid - 100

z<sub>f</sub> - Enthalpy per unit mass saturated liquid - 100

M - Total mass of system - 100

c<sub>p</sub> - Constant pressure specific heat - 100

T - Temperature - 100

T<sub>w</sub> - Temperature of boundary heated surface - 100

T<sub>s</sub> - Temperature of saturated liquid - 100

X<sub>2</sub> - Molar fraction of component 2 in mixture - 100

z<sub>g</sub> - Enthalpy per unit mass saturated vapor - 100

z<sub>f</sub> - Enthalpy per unit mass saturated liquid - 100

h<sub>g</sub> - Enthalpy per unit mass of saturated vapor - 100

h<sub>f</sub> - Enthalpy per unit mass of saturated liquid - 100

h<sub>g</sub> - Enthalpy per unit mass of saturated vapor - 100

h<sub>f</sub> - Enthalpy per unit mass of saturated liquid - 100

h<sub>g</sub> - Enthalpy per unit mass of saturated vapor - 100

h<sub>f</sub> - Enthalpy per unit mass of saturated liquid - 100

$p'$  - pressure of liquid

$C_1, C_2$  - radii of curvature

$\sigma$  - surface tension



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APPENDIX

Case	Pressure psia	Velocity ft/sec	Q/A BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
A	500	20	$2.34 \times 10^6$	117	32.25	1.225
					493.25	2.453
					360.25	4.907
					170.5	7.351
					69.5	9.802
					23.25	12.251
					8.25	14.457
					1.5	17.152
					1.75	19.630
					.2	22.053
.75	24.699					
B	500	20	$2.02 \times 10^6$	117	22.25	1.223
					445	2.447
					185.25	4.894
					150	7.345
					55.25	9.787
					10.5	12.233
					6.75	14.677
					3.5	17.127
					2.75	22.021
.5	29.361					



Case	Pressure psia	Velocity ft/sec	Q/A BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
C	500	20	$2.04 \times 10^6$	92	87.0	1.23
					743.2	2.46
					591.5	4.91
					121.0	7.36
					20.2	9.80
					3.2	12.30
					.7	14.70
D	500	20	$1.705 \times 10^6$	92	16.7	1.23
					512.2	2.46
					209.0	4.91
					40.5	7.36
					11.5	9.80
					5.0	12.30
					6.0	14.70
					.5	17.20
					.7	19.60
.2	24.60					
E	500	20	$1.43 \times 10^6$	92	12.5	1.23
					427.0	2.46
					186.5	4.91
					30.2	7.36
					10.2	9.80
					2.5	12.30
					3.5	14.70

Case	Pressure P <sub>1</sub>	Velocity V <sub>1</sub>	$\frac{V_1^2}{2g}$	Hydraulic Depth h <sub>1</sub>	Water Depth h <sub>1</sub>	Distance from Weir
C	2.00	1.00	0.05	1.00	1.05	1.00
	3.00	1.50	0.11	1.00	1.11	1.00
	4.00	2.00	0.20	1.00	1.20	1.00
	5.00	2.50	0.31	1.00	1.31	1.00
	6.00	3.00	0.45	1.00	1.45	1.00
	7.00	3.50	0.61	1.00	1.61	1.00
	8.00	4.00	0.80	1.00	1.80	1.00
D	2.00	1.00	0.05	1.00	1.05	1.00
	3.00	1.50	0.11	1.00	1.11	1.00
	4.00	2.00	0.20	1.00	1.20	1.00
	5.00	2.50	0.31	1.00	1.31	1.00
	6.00	3.00	0.45	1.00	1.45	1.00
	7.00	3.50	0.61	1.00	1.61	1.00
	8.00	4.00	0.80	1.00	1.80	1.00
E	2.00	1.00	0.05	1.00	1.05	1.00
	3.00	1.50	0.11	1.00	1.11	1.00
	4.00	2.00	0.20	1.00	1.20	1.00
	5.00	2.50	0.31	1.00	1.31	1.00
	6.00	3.00	0.45	1.00	1.45	1.00
	7.00	3.50	0.61	1.00	1.61	1.00
	8.00	4.00	0.80	1.00	1.80	1.00

Case	Pressure psia	Velocity ft/sec	Q/A BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter in.
					0.7	17.20
					1.2	19.60
					.5	24.60
F	1000	20	$2.155 \times 10^6$	95	44.7	1.26
					1277.3	2.50
					974.2	5.00
					232.3	7.600
					24.0	10.10
					5.0	12.60
					.75	15.10
					.75	17.60
					.75	20.20
G	1000	20	$1.820 \times 10^6$	95	336.8	1.26
					466.2	2.50
					179.1	5.00
					103.2	7.60
					98.8	10.10
					23.9	12.60
					9.0	15.10
H	1000	20	$1.516 \times 10^6$	95	94.1	1.25
					467.8	2.50
					136.7	5.00
					37.1	7.50





Case	Pressure psia	Velocity ft/sec	$\frac{1}{A} \frac{d^2}{dt^2}$ FTO/FT <sup>2</sup> -hr	Subcooling °F	Number	Diameter in.
					17.6	10.00
					0.56	12.50
					0.78	15.00
I	1000	20	$1.253 \times 10^6$	70	344.9	1.26
					385.5	2.51
					348.1	5.02
					132.5	7.54
					98.3	10.05
					21.7	12.56
					6.5	15.07
J	1000	20	$1.258 \times 10^6$	70	108.4	1.26
					546.6	2.50
					326.2	5.00
					159.5	7.60
					51.5	10.10
					7.5	12.60
					2.0	15.10
					.9	17.60
K	1000	20	$1.28 \times 10^6$	45	43.7	1.22
					488.5	2.45
					339.7	4.89
					198.2	7.34
					100.5	9.79

Case Number	Subsidiary	Number	Class
10001	1000	1000	1000
10002	1000	1000	1000
10003	1000	1000	1000
10004	1000	1000	1000
10005	1000	1000	1000
10006	1000	1000	1000
10007	1000	1000	1000
10008	1000	1000	1000
10009	1000	1000	1000
10010	1000	1000	1000
10011	1000	1000	1000
10012	1000	1000	1000
10013	1000	1000	1000
10014	1000	1000	1000
10015	1000	1000	1000
10016	1000	1000	1000
10017	1000	1000	1000
10018	1000	1000	1000
10019	1000	1000	1000
10020	1000	1000	1000
10021	1000	1000	1000
10022	1000	1000	1000
10023	1000	1000	1000
10024	1000	1000	1000
10025	1000	1000	1000
10026	1000	1000	1000
10027	1000	1000	1000
10028	1000	1000	1000
10029	1000	1000	1000
10030	1000	1000	1000
10031	1000	1000	1000
10032	1000	1000	1000
10033	1000	1000	1000
10034	1000	1000	1000
10035	1000	1000	1000
10036	1000	1000	1000
10037	1000	1000	1000
10038	1000	1000	1000
10039	1000	1000	1000
10040	1000	1000	1000
10041	1000	1000	1000
10042	1000	1000	1000
10043	1000	1000	1000
10044	1000	1000	1000
10045	1000	1000	1000
10046	1000	1000	1000
10047	1000	1000	1000
10048	1000	1000	1000
10049	1000	1000	1000
10050	1000	1000	1000

Case	Pressure psia	Velocity ft/sec	$Q/A$ BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
					46.7	12.24
					25.0	14.60
					15.75	17.13
					11.5	19.58
					9.0	22.02
					5.25	24.47
					1.25	26.92
L	1000	20	$1.022 \times 10^6$	45	38.7	1.22
					496.5	2.45
					666.7	4.89
					72.5	7.34
					3.5	9.79
M	1000	20	$.799 \times 10^6$	45	344.5	1.22
					530.7	2.45
					344.7	4.89
					96.0	7.34
					29.0	9.78
					1.5	12.23
N	1000	20	$1.035 \times 10^6$	20	63	1.22
					364.0	2.43
					358.75	4.87
					292.5	7.30
					197.5	9.73

Case Number	Year	Volume	Page	Notes
1000	1900	100	100	
1001	1901	100	100	
1002	1902	100	100	
1003	1903	100	100	
1004	1904	100	100	
1005	1905	100	100	
1006	1906	100	100	
1007	1907	100	100	
1008	1908	100	100	
1009	1909	100	100	
1010	1910	100	100	
1011	1911	100	100	
1012	1912	100	100	
1013	1913	100	100	
1014	1914	100	100	
1015	1915	100	100	
1016	1916	100	100	
1017	1917	100	100	
1018	1918	100	100	
1019	1919	100	100	
1020	1920	100	100	
1021	1921	100	100	
1022	1922	100	100	
1023	1923	100	100	
1024	1924	100	100	
1025	1925	100	100	
1026	1926	100	100	
1027	1927	100	100	
1028	1928	100	100	
1029	1929	100	100	
1030	1930	100	100	
1031	1931	100	100	
1032	1932	100	100	
1033	1933	100	100	
1034	1934	100	100	
1035	1935	100	100	
1036	1936	100	100	
1037	1937	100	100	
1038	1938	100	100	
1039	1939	100	100	
1040	1940	100	100	
1041	1941	100	100	
1042	1942	100	100	
1043	1943	100	100	
1044	1944	100	100	
1045	1945	100	100	
1046	1946	100	100	
1047	1947	100	100	
1048	1948	100	100	
1049	1949	100	100	
1050	1950	100	100	

Case	Pressure psia	Velocity ft/sec	$\frac{L}{A}$ BPS/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
					91.25	12.14
					37.50	14.60
					20.75	17.03
					13.25	19.46
					6.00	21.92
					4.00	24.33
					3.00	26.76
					1.5	29.20
					.25	31.63
					0	34.06
					15	36.50
0	1000	20	$.810 \times 10^6$	20	16.7	1.22
					308.7	2.45
					162.7	4.90
					169.0	7.35
					117.7	9.80
					93.5	12.25
					37.5	14.70
					23.0	17.15
					16.0	19.60
					11.0	22.05
					6.2	24.50
F	1000	20	$.610 \times 10^6$	20	220.5	1.22
					372.75	2.44
					220.75	4.89



Case	Pressure psia	Velocity ft/sec	$\frac{G}{A}$ lb/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
					136.75	7.33
					80.75	9.77
					36.50	12.22
					16.75	14.66
					6.50	17.10
					2.75	19.54
					.25	21.99
Q	1000	30	$2.57 \times 10^6$	95	132.5	1.26
					751.0	2.51
					164.5	5.03
					47.5	7.55
					19.0	10.06
					3.0	12.58
					1.7	15.09
R	1000	30	$2.22 \times 10^6$	97	433	1.25
					399.5	2.52
					180.2	5.04
					26.7	7.56
					7.7	10.03
					0.0	12.50
					0.2	15.12
S	1000	30	$1.9 \times 10^6$	97	55.75	1.26
					203.5	2.52



Class	Time	Location	Room	Section	Days	Time
101	8:00	101	101	101		
102	8:30	102	102	102		
103	9:00	103	103	103		
104	9:30	104	104	104		
105	10:00	105	105	105		
106	10:30	106	106	106		
107	11:00	107	107	107		
108	11:30	108	108	108		
109	12:00	109	109	109		
110	12:30	110	110	110		
111	1:00	111	111	111		
112	1:30	112	112	112		
113	2:00	113	113	113		
114	2:30	114	114	114		
115	3:00	115	115	115		
116	3:30	116	116	116		
117	4:00	117	117	117		
118	4:30	118	118	118		
119	5:00	119	119	119		
120	5:30	120	120	120		
121	6:00	121	121	121		
122	6:30	122	122	122		
123	7:00	123	123	123		
124	7:30	124	124	124		
125	8:00	125	125	125		
126	8:30	126	126	126		
127	9:00	127	127	127		
128	9:30	128	128	128		
129	10:00	129	129	129		
130	10:30	130	130	130		
131	11:00	131	131	131		
132	11:30	132	132	132		
133	12:00	133	133	133		
134	12:30	134	134	134		
135	1:00	135	135	135		
136	1:30	136	136	136		
137	2:00	137	137	137		
138	2:30	138	138	138		
139	3:00	139	139	139		
140	3:30	140	140	140		
141	4:00	141	141	141		
142	4:30	142	142	142		
143	5:00	143	143	143		
144	5:30	144	144	144		
145	6:00	145	145	145		
146	6:30	146	146	146		
147	7:00	147	147	147		
148	7:30	148	148	148		
149	8:00	149	149	149		
150	8:30	150	150	150		

Case	Pressure psia	Velocity ft/sec	$Q/A$ $10^6/\text{ft}^2\text{-hr}$	Subcooling $^{\circ}\text{F}$	Number	Diameter mils
					90.75	1.04
					31.0	7.56
					10.75	10.08
					1.25	12.60
					2.00	15.12
T	1000	30	$1.882 \times 10^6$	70	196.5	1.25
					985.3	2.52
					215.8	5.04
					31.7	7.56
					8.2	10.01
					1.5	12.60
					0.25	15.10
U	1000	30	$1.545 \times 10^6$	70	14	1.29
					445	2.58
					111.5	5.17
					25.7	7.75
					1.7	10.30
V	1000	30	$.846 \times 10^6$	20	119.2	1.22
					639.0	2.45
					232.0	4.90
					79.7	7.34
					16.2	9.79
					3.0	12.24

Case	Quantity	Velocity	W/V	Resolving	Number	Time
						00.12
						01.0
						02.12
						03.0
						04.12
						05.0
						06.12
						07.0
						08.12
						09.0
						10.12
						11.0
						12.12
						13.0
						14.12
						15.0
						16.12
						17.0
						18.12
						19.0
						20.12
						21.0
						22.12
						23.0
						24.12
						25.0
						26.12
						27.0
						28.12
						29.0
						30.12
						31.0
						32.12
						33.0
						34.12
						35.0
						36.12
						37.0
						38.12
						39.0
						40.12
						41.0
						42.12
						43.0
						44.12
						45.0
						46.12
						47.0
						48.12
						49.0
						50.12
						51.0
						52.12
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						54.12
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						56.12
						57.0
						58.12
						59.0
						60.12
						61.0
						62.12
						63.0
						64.12
						65.0
						66.12
						67.0
						68.12
						69.0
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						81.0
						82.12
						83.0
						84.12
						85.0
						86.12
						87.0
						88.12
						89.0
						90.12
						91.0
						92.12
						93.0
						94.12
						95.0
						96.12
						97.0
						98.12
						99.0
						100.12

Case	Pressure psia	Velocity ft/sec	$\frac{G}{A}$ MW/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter in.
					1.5	14.34
					.5	17.14
W	1000	30	$.614 \times 10^6$	20	182	1.22
					355.7	2.45
					207.0	4.90
					69.5	7.35
					24.7	9.80
X	1000	30	$.452 \times 10^6$	10	330.25	1.26
					362.00	2.52
					334.24	5.04
					182.75	7.57
					116.50	10.09
					22.50	12.62
					6.25	15.14
Y	1000	30	$.321 \times 10^6$	7	291.5	1.25
					314.5	2.50
					260.0	5.00
					163.75	7.49
					104.0	9.99
					29.0	12.49
					5.75	14.99

Station	Time	Temperature	Direction	Force	Remarks	Notes
10.0	10.00	10.0				
10.1	10.05	10.1				
10.2	10.10	10.2				
10.3	10.15	10.3				
10.4	10.20	10.4				
10.5	10.25	10.5				
10.6	10.30	10.6				
10.7	10.35	10.7				
10.8	10.40	10.8				
10.9	10.45	10.9				
11.0	10.50	11.0				
11.1	10.55	11.1				
11.2	11.00	11.2				
11.3	11.05	11.3				
11.4	11.10	11.4				
11.5	11.15	11.5				
11.6	11.20	11.6				
11.7	11.25	11.7				
11.8	11.30	11.8				
11.9	11.35	11.9				
12.0	11.40	12.0				
12.1	11.45	12.1				
12.2	11.50	12.2				
12.3	11.55	12.3				
12.4	12.00	12.4				
12.5	12.05	12.5				
12.6	12.10	12.6				
12.7	12.15	12.7				
12.8	12.20	12.8				
12.9	12.25	12.9				
13.0	12.30	13.0				
13.1	12.35	13.1				
13.2	12.40	13.2				
13.3	12.45	13.3				
13.4	12.50	13.4				
13.5	12.55	13.5				
13.6	13.00	13.6				
13.7	13.05	13.7				
13.8	13.10	13.8				
13.9	13.15	13.9				
14.0	13.20	14.0				
14.1	13.25	14.1				
14.2	13.30	14.2				
14.3	13.35	14.3				
14.4	13.40	14.4				
14.5	13.45	14.5				
14.6	13.50	14.6				
14.7	13.55	14.7				
14.8	14.00	14.8				
14.9	14.05	14.9				
15.0	14.10	15.0				
15.1	14.15	15.1				
15.2	14.20	15.2				
15.3	14.25	15.3				
15.4	14.30	15.4				
15.5	14.35	15.5				
15.6	14.40	15.6				
15.7	14.45	15.7				
15.8	14.50	15.8				
15.9	14.55	15.9				
16.0	15.00	16.0				

Case	Pressure psia	Velocity ft/sec	Q/A BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter mils
Z	1500	20	$2.7 \times 10^6$	144	412	1.27
					641.2	2.54
					371.0	5.07
					72.7	7.60
					29.0	10.14
					0.7	12.68
					0.5	15.21
AA	1500	20	$2.27 \times 10^6$	146	294.5	1.26
					617.75	2.53
					137.5	5.64
					25	7.58
					6.25	10.11
					.25	12.63
BB	1500	20	$1.97 \times 10^6$	94	10.7	1.26
					755.7	2.51
					222.5	5.02
					30.0	7.53
					5.2	10.04
CC	1000	20	$1.669 \times 10^6$	69	21.7	1.26
					684.2	2.51
					249.2	5.02
					58.5	7.54

Gas Pressure	Gas Pressure	Velocity	Flow	Impedance	Number	Diameter
2	1200	80	$5.7 \times 10^2$	111	112	1.27
					113	1.27
					114	1.27
					115	1.27
					116	1.27
					117	1.27
					118	1.27
					119	1.27
4A	1200	80	$5.27 \times 10^2$	116	120	1.27
					121	1.27
					122	1.27
					123	1.27
					124	1.27
					125	1.27
					126	1.27
					127	1.27
					128	1.27
					129	1.27
					130	1.27
					131	1.27
					132	1.27
					133	1.27
					134	1.27
					135	1.27
					136	1.27
					137	1.27
					138	1.27
					139	1.27
					140	1.27
					141	1.27
					142	1.27
					143	1.27
					144	1.27
					145	1.27
					146	1.27
					147	1.27
					148	1.27
					149	1.27
					150	1.27
					151	1.27
					152	1.27
					153	1.27
					154	1.27
					155	1.27
					156	1.27
					157	1.27
					158	1.27
					159	1.27
					160	1.27
					161	1.27
					162	1.27
					163	1.27
					164	1.27
					165	1.27
					166	1.27
					167	1.27
					168	1.27
					169	1.27
					170	1.27
					171	1.27
					172	1.27
					173	1.27
					174	1.27
					175	1.27
					176	1.27
					177	1.27
					178	1.27
					179	1.27
					180	1.27
					181	1.27
					182	1.27
					183	1.27
					184	1.27
					185	1.27
					186	1.27
					187	1.27
					188	1.27
					189	1.27
					190	1.27
					191	1.27
					192	1.27
					193	1.27
					194	1.27
					195	1.27
					196	1.27
					197	1.27
					198	1.27
					199	1.27
					200	1.27

Case	Pressure psia	Velocity ft/sec	Q/A BTU/ft <sup>2</sup> -hr	Subcooling °F	Number	Diameter in
					21.2	10.00
					2.7	12.57
					1.5	15.10



Case Number	Transverse Gain	Velocity (ft/sec)	$\frac{S}{L} \frac{A}{V} \frac{D}{C}$	Revolving Speed (rpm)	Number of Revs	Radius (in)
10.00	5.25					
12.51	1.9					
15.12	2.4					
17.73	3.0					
20.34	3.6					
22.95	4.2					
25.56	4.8					
28.17	5.4					
30.78	6.0					
33.39	6.6					
36.00	7.2					
38.61	7.8					
41.22	8.4					
43.83	9.0					
46.44	9.6					
49.05	10.2					
51.66	10.8					
54.27	11.4					
56.88	12.0					
59.49	12.6					
62.10	13.2					
64.71	13.8					
67.32	14.4					
69.93	15.0					
72.54	15.6					
75.15	16.2					
77.76	16.8					
80.37	17.4					
82.98	18.0					
85.59	18.6					
88.20	19.2					
90.81	19.8					
93.42	20.4					
96.03	21.0					
98.64	21.6					
101.25	22.2					
103.86	22.8					
106.47	23.4					
109.08	24.0					
111.69	24.6					
114.30	25.2					
116.91	25.8					
119.52	26.4					
122.13	27.0					
124.74	27.6					
127.35	28.2					
130.00	28.8					
132.61	29.4					
135.22	30.0					
137.83	30.6					
140.44	31.2					
143.05	31.8					
145.66	32.4					
148.27	33.0					
150.88	33.6					
153.49	34.2					
156.10	34.8					
158.71	35.4					
161.32	36.0					
163.93	36.6					
166.54	37.2					
169.15	37.8					
171.76	38.4					
174.37	39.0					
176.98	39.6					
179.59	40.2					
182.20	40.8					
184.81	41.4					
187.42	42.0					
190.03	42.6					
192.64	43.2					
195.25	43.8					
197.86	44.4					
200.47	45.0					
203.08	45.6					
205.69	46.2					
208.30	46.8					
210.91	47.4					
213.52	48.0					
216.13	48.6					
218.74	49.2					
221.35	49.8					
223.96	50.4					
226.57	51.0					
229.18	51.6					
231.79	52.2					
234.40	52.8					
237.01	53.4					
239.62	54.0					
242.23	54.6					
244.84	55.2					
247.45	55.8					
250.06	56.4					
252.67	57.0					
255.28	57.6					
257.89	58.2					
260.50	58.8					
263.11	59.4					
265.72	60.0					
268.33	60.6					
270.94	61.2					
273.55	61.8					
276.16	62.4					
278.77	63.0					
281.38	63.6					
283.99	64.2					
286.60	64.8					
289.21	65.4					
291.82	66.0					
294.43	66.6					
297.04	67.2					
299.65	67.8					
302.26	68.4					
304.87	69.0					
307.48	69.6					
310.09	70.2					
312.70	70.8					
315.31	71.4					
317.92	72.0					
320.53	72.6					
323.14	73.2					
325.75	73.8					
328.36	74.4					
330.97	75.0					
333.58	75.6					
336.19	76.2					
338.80	76.8					
341.41	77.4					
344.02	78.0					
346.63	78.6					
349.24	79.2					
351.85	79.8					
354.46	80.4					
357.07	81.0					
359.68	81.6					
362.29	82.2					
364.90	82.8					
367.51	83.4					
370.12	84.0					
372.73	84.6					
375.34	85.2					
377.95	85.8					
380.56	86.4					
383.17	87.0					
385.78	87.6					
388.39	88.2					
391.00	88.8					
393.61	89.4					
396.22	90.0					
398.83	90.6					
401.44	91.2					
404.05	91.8					
406.66	92.4					
409.27	93.0					
411.88	93.6					
414.49	94.2					
417.10	94.8					
419.71	95.4					
422.32	96.0					
424.93	96.6					
427.54	97.2					
430.15	97.8					
432.76	98.4					
435.37	99.0					
437.98	99.6					
440.59	100.2					
443.20	100.8					
445.81	101.4					
448.42	102.0					
451.03	102.6					
453.64	103.2					
456.25	103.8					
458.86	104.4					
461.47	105.0					
464.08	105.6					
466.69	106.2					
469.30	106.8					
471.91	107.4					
474.52	108.0					
477.13	108.6					
479.74	109.2					
482.35	109.8					
484.96	110.4					
487.57	111.0					
490.18	111.6					
492.79	112.2					
495.40	112.8					
498.01	113.4					
500.62	114.0					
503.23	114.6					
505.84	115.2					
508.45	115.8					
511.06	116.4					
513.67	117.0					
516.28	117.6					
518.89	118.2					
521.50	118.8					
524.11	119.4					
526.72	120.0					
529.33	120.6					
531.94	121.2					
534.55	121.8					
537.16	122.4					
539.77	123.0					
542.38	123.6					
544.99	124.2					
547.60	124.8					
550.21	125.4					
552.82	126.0					
555.43	126.6					
558.04	127.2					
560.65	127.8					
563.26	128.4					
565.87	129.0					
568.48	129.6					
571.09	130.2					
573.70	130.8					
576.31	131.4					
578.92	132.0					
581.53	132.6					
584.14	133.2					
586.75	133.8					
589.36	134.4					
591.97	135.0					
594.58	135.6					
597.19	136.2					
599.80	136.8					
602.41	137.4					
605.02	138.0					
607.63	138.6					
610.24	139.2					
612.85	139.8					
615.46	140.4					
618.07	141.0					
620.68	141.6					
623.29	142.2					
625.90	142.8					
628.51	143.4					
631.12	144.0					
633.73	144.6					
636.34	145.2					
638.95	145.8					
641.56	146.4					
644.17	147.0					
646.78	147.6					
649.39	148.2					
652.00	148.8					
654.61	149.4					
657.22	150.0					
659.83	150.6					
662.44	151.2					
665.05	151.8					
667.66	152.4					
670.27	153.0					
672.88	153.6					
675.49	154.2					
678.10	154.8					
680.71	155.4					
683.32	156.0					
685.93	156.6					
688.54	157.2					
691.15	157.8					
693.76	158.4					
696.37	159.0					
698.98	159.6					
701.59	160.2					
704.20	160.8					
706.81	161.4					
709.42	162.0					
712.03	162.6					









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