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BURNOUT IN FORCED CONVECTION NUCLEATE BOILING OF WATER

by

John M. Reynolds III

S.B. (1951) Yale University

Submitted in Partial Fulfillment of the

Requirements for the Degree of

MASTER OF SCIENCE

at the

Massachusetts Institute of Technology

(1956)

Signature of Author _____
Department of Mechanical Engineering, January 14, 1957

Signature of Professor
in charge of Research _____

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Dr. Leicester F. Hamilton
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

In partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering, I hereby submit this thesis entitled, "Burnout in Forced Convection Nucleate Boiling of Water".

Sincerely yours,

John M. Reynolds

Burnout in Forced Convection Nucleate Boiling of Water

by

John M. Reynolds

Submitted to the Department of Mechanical Engineering on January 14, 1957 in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

ABSTRACT

Burnout heat flux data are presented for water at 500, 1000, and 1500 psia with specific flow rates of 1×10^6 and 2×10^6

A dimensional correlation of the data is developed and extended to include the data of other observers. The correlation yields the relation

$$\left(\frac{\delta}{R}\right)^{1.94} = 1.48 \times 10^{-15}\left[\left(\frac{\nu_0}{\nu_f}\right)\left(\frac{q''D}{\sigma v_0}\right) G^{1.5} D^{.5}\right]$$

for the prediction of burnout heat flux.

Thesis Supervisor

John A. Clark
Assistant Professor
Mechanical Engineering

ACKNOWLEDGEMENTS

This thesis was carried out under the supervision of Professor J. A. Clark whose advice and intuition provided me with new approaches to the problem during some dark moments.

I wish to thank Professor Peter Griffith for the many hours spent with me in discussing and clarifying the concepts involved in the correlation.

Thanks for a wealth of practical advice is due to M. W. Raymond who designed and developed the burnout detector used on this project as well as most of the other equipment used on it, and to J. Lehtinen whose data are included in this report.

Fred Johnson kept the project running through some trying times and assisted in taking the data. Thanks also to Messrs. T. M. Alexander and R. E. Smith who assisted with the calculations.

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1. INTRODUCTION

In recent years the process of nucleate boiling has been energetically investigated as a means of increasing the effectiveness of heat exchanger coolants. This work has been in progress at MIT since 1949 under the sponsorship of the Atomic Energy Commission and the Office of Naval Research.

The boiling heat transfer process is characterized by the curve shown in fig 1. Upon the initiation of boiling, the heat flux increases rapidly with increasing temperature difference between the wall and fluid. This is the region of nucleate boiling. Eventually, a maximum value of heat flux is attained which can only be exceeded by increasing the temperature difference by a factor of the order of one hundred. This rise in wall temperature will, in most practical apparatus, result in rupture of the wall. The maximum nucleate boiling heat flux is called the "burnout" heat flux and is readily seen as an important design consideration in equipment utilizing a boiling liquid as the coolant.

The present experimental program was undertaken to provide data on burnout heat flux in water systems at moderately high pressures, 500 to 2000 psia, and flow rates, 0.5 to 5.0 x 10^6 lb/Hr-ft^2.

2. PROCEDURE

2.1 Description of the test apparatus

All data were taken on a 9-inch length of "A" nickel tubing, .1805 inches in internal diameter and .2105 inches in external diameter. The tube was vertical and the flow upward. Electrical heating of the test section was accomplished by passing current through the tube wall. The source of electrical power was two 3000 amp 12 volt DC generators in series which could be effectively controlled from zero to maximum output.

The system consists of a closed loop with a stainless steel De Laval Hayward-Tyler circulating pump. Pressure was maintained on the system by a pressure vessel in which water was evaporated by electrical heating. The pressure vessel was also used to degass the water before each run.

Distilled water was used in all runs and an ion exchanger was used to maintain the impurity level below 1 ppm NaCl.

The test section instrumentation consisted of inlet and outlet pressure and voltage taps and nine chromel-constantan thermocouples on the tube outer wall. The wall thermocouples were insulated from the tube by .0015 in. mica. The temperature difference across the tube wall was computed by the method of ref 1. Inlet and outlet fluid temperatures were measured 6 inches upstream and 4 inches downstream of the test section.

The tube was surrounded by four thermal shields, each

enclosing the tube for one-quarter of its length. The shields are individually variable resistance heaters wound on aluminum tubing. The shield temperatures were adjusted to the average temperature of the portion of the test section under each.

A detailed discussion of orifice and thermocouple calibration, tube size and resistance determination, and current and voltage measurement is given in ref 1. The details of the test stand apparatus and operation are described in ref 2.

2.2 Burnout Detector

The requirements of the burnout program necessitated the use of a device to prevent failure of the test section at the burnout point. This device, or burnout detector, must be one which can identify the burnout condition and take action to prevent failure, but must also allow the experimentor to attain burnout heat flux. In principle, the detector relies on the fact that burnout occurs at the test section exit and is characterized by a large temperature rise. Actually, the detector compares the electrical resistance of the final 1/16" of the test section with that of the remaining tube length. These two resistances constitute two legs of a Wheatstone bridge circuit, the other two legs consisting of fixed and variable resistors of suitable ratings as shown in

fig 2. At burnout the resistance of the final length of the test section increases rapidly in relation to the remainder causing an unbalance in the bridge circuit.

The net voltage at the reference point in the bridge circuit provides the input signal to a four stage DC amplifier. The output of the amplifier fires a triggering circuit which discharges a 7000 volt .10 µf capacitor through a spark gap in the circuit breaker.

The time which the burnout detector needs to "decide" when a bona fide burnout has occurred is relatively long, about 6 millisec. At peak pressure and heat flux, this interval is virtually all the time available before the test section allowable overtemperature of 50° F is reached. Consequently, an instantaneous circuit breaker is required to act on the signal from the amplifier.

Rapid circuit interruption is accomplished by using the .10 µf capacitor discharge to fire an explosive cap* which is contained in a hollow copper tube. The test section current flows continually through the copper tube which is one-half inch in outer diameter, 3 1/32" long, and has an .047" wall. The copper tube is restrained by a mechanical device which insures clean circuit interruption when the cap is fired and low voltage drop through the tube and its connections during the run. Details of

* duPont X-98-N blasting cap, #6 strength, RDX loaded for 300° F temperature stability.

the breaker construction as well as a complete discussion of the detector circuitry will be subject of a project report to be issued shortly.

The burnout detector has been an effective instrument for the study of the burnout phenomenon and is well suited to the characteristics of the process. During a run, the test section current is increased quite slowly. Therefore, any bridge unbalance due to varying tube temperature can be readily compensated by manual adjustment of the variable resistors. At high heat flux, in the neighborhood of burnout, the test section is normally almost isothermal, so that, while the magnitude of the test section resistance may vary, the resistances of the two portions of the tube remain in a relatively constant ratio to each other. This makes manual adjustment unneccessary near the burnout point.

Almost without exception, the burnout detector has successfully prevented failure of the test section and permitted the actual attainment of the burnout condition in the test section. The fact that the detector is not firing prematurely is periodically checked by allowing a test section to fail while observing the time between the discharge of the capacitor and the appearance of steam or other evidence of failure. These two events occur simultaneously.

3. RESULTS

Burnout data were taken at 500, 1000, and 1500 psia system pressure and at water inlet velocities of 5 and 10 feet per second. These data are presented in Appendix A. Another, previously unpublished, set of data taken in the same range by John Lehtinen on the MIT apparatus is presented in Appendix B.

3.1 Discussion of results

The apparatus used in this work has been employed for a number of years in studying the nucleate boiling process, and its limitations are well known. The following errors are inherent in the measurement and computation schemes:

a. Heat flux is known to ± 3%

b. Flow rate is known to ± 2%

c. Inlet and outlet fluid temperatures and test section wall temperatures are known to ± 2° F.

These quantities are interrelated in three measures of heat rejection. Heat rejection may be determined from:

a. The difference in inlet and outlet fluid enthalpy as measured by the corresponding temperatures and obtained from a table of properties (ref 3) multiplied by the flow rate.

b. The product of current and test section voltage drop.

e. Summation of heat release in the nine wall sections based on the tube resistivity at each of the nine wall temperatures and current.

These three computations yield the ± 3% heat flux variation.

Burnout data is recorded on a 16-point Brown recorder since it is impossible to achieve steady state at the burnout point. A complete set of data extends 16 seconds before burnout occurs so that some temperature readings must be estimated to ± 3° F. Current and voltage readings are printed three times during each recorder cycle, so that errors in those readings are not increased.

4. CORRELATION OF THE DATA

4.1 The Correlation

In an effort to evaluate the data taken at MIT in relation to other data, a burnout correlation was undertaken. The objectives of this correlation were to unify the MIT data, to bring outside data into agreement, and to do it in terms of easily measurable quantities. In short, to determine a reliable engineering correlation.

The logical starting point for such a correlation is the fluid properties at the point at which burnout occurs, i.e. test section exit. It was found that the simple properties such as bulk quality, density, flow rate, pressure, and heat flux do not exhibit any obvious relationship among themselves over a sufficiently wide range of conditions to be of significant use. Qualitatively, however, one can deduce from these properties the fact that increasing exit bulk quality reflects a decreasing maximum heat flux at constant pressure.

In studying the effects of the various properties on heat flux, it was found that the average exit flow velocity (assuming no slip between phases) correlated the MIT data extremely well over the range of pressures studied. Unfortunately, the relatively simple expressions involved in that correlation proved inadequate when applied to data taken elsewhere, and it was discarded. Another parameter

was sought in the area of bubble mechanics. The theoretical limiting size of bubbles has been extensively discussed in the literature and recently by P. Griffith in reference 4. Conceptually, the results of these analyses reveal that there is a predictable maximum bubble size attainable in nucleate boiling when the average temperature of the stream is below the saturation temperature at the pressure under consideration. From a mechanistic viewpoint, the bubble is considered to reach its maximum size when there is zero net heat transfer across its surface. That is, the bubble will not grow beyond the point at which the heat rejection to the subcooled stream equals the heat absorbtion from the superheated region near the wall. This reasoning cannot be used to determine bubble size in a stream at saturated temperature, but the concept of a limiting bubble diameter can be applied if a suitable criterion is devised..

Consider a two-phase flow having a vapor core and a liquid film on the tube walls. According to data taken by Marchatierre (ref 5), it can be reliably assumed that no slip exists between the phases for sufficiently high flow velocities. Assuming that one can predict it, the thickness of the liquid film on the wall constitutes an acceptable measure of maximum bubble diameter. In this report, the film thickness is described as that consisting of all

the liquid in the exit plane evaluated at average outlet quality. While it is quite certain that this is not actually the flow condition, this approximation is sufficiently accurate for correlation purposes if a constant fraction of the film height so computed is realized in each case. This film thickness (referred to as δ hereafter) showed significant promise as a correlating factor.

At this point a dimensional analysis was performed with the assumed relation of the form

$$q'' = f(D, \delta, V_o, \gamma_f, \gamma_o, \sigma) \qquad (1)$$

where q'' = heat flux

D = cross sectional dimension

δ = liquid film thickness

V_o = exit velocity

γ_o = exit specific weight

γ_f = saturated liquid specific weight

σ = surface tension

The choice of outlet density in this functional relationship was somewhat arbitrary since it is already implicit in the exit velocity term. It was felt, however, that some means of distinguishing between exit velocities for high flow rate-low quality systems as opposed to low flow rate-high quality systems was necessary. Since γ_f was needed if the film concept was significant, γ_o was used as the distinguishing factor rather than G, the

specific flow rate.

Surface tension was felt to be important since it is reflected in two important areas.

a. Surface tension plays a significant role in the determination of wall superheat, $T_w - T_{SAT}$ (ref 5)

b. Surface tension can be expected to be an important factor in determining the character of their liquid films on the wall. It would have an effect on the stability of such a film and, for vertical tubes, would determine the amount of liquid thrown off the wall to the stream by the action of the bubbles.

The result of the dimensional analysis is:

$$\frac{\delta}{R} = f\left[\frac{q''D}{\sigma V_0}, \frac{V_0}{V_f}\right] \qquad (1)$$

Qualitatively, the first term on the right-hand side appears reasonable. Increasing surface tension indicates increased wall superheat and increased integrity of the liquid film, both of which would be expected to increase heat flux. Increasing velocity would be expected to have a similar effect. The appearance of all terms to the first power is, however, somewhat startling. It will be be seen below that this dimensionless group is in fact erroneous in the relationship of the quantities within it to each other. It does, however, appear to include all the significant quantities.

When this functional relationship was tried with

the two right-hand groups involved as a product to the first power, a satisfactory correlation was obtained for data at constant specific flow rate, G, over a range of pressures. It was necessary to multiply the right hand term by $G^{1.5}$ to make the correlation applicable to all flow rates.

This added flow parameter apparently reflects the changing heat capacity per unit time of the system rather than a flow phenomenon, since Reynolds Number was tried unsuccessfully as a correlating factor.

Finally, an additional factor of $D^{.5}$ was required to bring the Argonne National Laboratory data on a .306" diameter test section and the Westinghouse data on a 2" x 0,097" rectangular test section in line with the well established curve for .18" diameter tubes.

The final correlation is

$$\left(\delta/R\right)^{1.94} = 1.48 \times 10^{-15} \left[\left(\frac{\nu_0}{\nu_f}\right)\left(\frac{q''D}{\sigma V_0}\right) G^{1.5} D^{.5}\right] \qquad (3)$$

for the units as shown in the nomenclature, i.e. all quantities in consistent feet-hour-pound units except q" (Btu/hr-ft^2).

The quantity δ/R is computed as follows. For both circular and rectangular test sections,

$$\delta/R = 1 - \sqrt{y} \qquad (4)$$

where y is volume quality $\equiv \frac{x v_{fg}}{v_f + x v_{fg}}$ in the notation of ref 3.

For rectangular test sections the characteristic channel dimension is taken as the hydraulic diameter.

* incorrectly written: $y \equiv \frac{x v_g}{v_f + x v_{fg}}$

Equation (3) may be written

$$\left[1-\sqrt{z\frac{r_o}{r_g}}\right]^{1.94} = 1.48\times10^{-15}\left(\frac{q''\,r_o^2\,D^{1.5}\,G^{.5}}{\sigma\,r_f}\right) \qquad (5)$$

for circular test sections.

4.2 Discussion of the correlation

The correlation is shown in fig 3 for the MIT data which is summarized in appendices A and B. Figure 4 presents the correlation of the Argonne National Laboratory data (ref 6) which was taken at 2000 psia on a .18" diameter tube and some Westinghouse data (ref 7) taken at 2000 psia on a 2" x .097" rectangular test section. Finally, the correlation is extended (fig 5) considerably to include some low flow rate data taken at MIT by J. A. Clark (ref 8).

The dimensional correlation presented above appears to have considerable merit because of the range of variables covered with a reasonable scatter of the points. The range of variables covered by the data in the main body of the correlation, above $\delta/R = .01$ is as follows:

a. heat flux, .2 x 10^6 to 2.8 x 10^6 Btu/Hr-ft^2

b. specific flow rate, .17 x 10^6 to 3.6 x 10^6 lb/Hr-ft^2

c. pressure, 500 to 2000 psia

d. exit bulk quality, .006 to .60 (successfully)

e. tube L/D 50 to 76

The low δ/R data extend these ranges as follows

f. heat flux to .05 x 10^6 Btu/Hr-ft^2

g. flow rate to .01 x 10^6 lb/Hr-ft^2

h. pressure to 100 psia

i. exit bulk quality to .93

The fact that the correlation extends through these ranges indicates continuity of one mechanism of burnout. There are possible exceptions, however. As shown in figure 7, the large tube ANL data scatters considerably and completely breaks away at qualities above 65%. Study of the 2000 psia Westinghouse data qualitatively confirms this trend at high quality. The reason for this departure is not known. While it appears to be a physical phenomenon, it may prove to be amplified scatter as discussed below.

One question which arises immediately on examination of the correlation is with regard to the prediction of decreasing burnout heat flux with increasing flow rate at constant quality. For example, the correlation predicts a value of 700 x 10^{-12} for the quantity $\frac{q'' r_o^2 D''^5 G^{.5}}{\sigma r_f}$ at $\delta/R = 1.0$, requiring that heat flux at constant pressure be inversely proportional to the square root of flow rate. While this relationship is accurate over the range covered by the M.I.T. data, there exist significant quantities of data showing an increase of heat flux with flow rate. The dividing line between these opposite relationships has not been determined. Comparative data is difficult to obtain in this area since heat flux is so sensitive to exit quality that it must be more than approximately

constant for a valid comparison to be made. From the success of the correlation, it is obvious that the inverse relationship generally applies when large change of the parameters occurs. In general, large increases in flow rate yield lower heat flux if quality remains constant, but there are "runs" of data over small ranges of flow rate (less than 20% variation) showing the opposite effect. While the answer to this problem is not now known, it is hoped that the correlation will satisfactorily tie the major effects together permitting a study of this "secondary" effect which is necessary before a complete understanding of the burnout process can be obtained.

4.3 Validity of the liquid film concept

This paper shows that a correlation based on a somewhat nebulous model involving a liquid film in two-phase flow with at least a constant slip ratio between phases is applicable over a S/R range from .8 to .0001. The probability of such a simple concept being valid over such a wide range appears slight and consideration has been given to dropping the S/R nomenclature in favor of a "function of quality." The fact that a logical application of the film parameter to rectangular channels is not successful lends some weight to this consideration. However, the lack of conclusive evidence coupled with a dearth of other physical explanations for the particular combination of

properties resulted in retention of the film thickness notation. In short, there appears to be no other explanation for its success except that it is reasonably accurate over a wide range.

4.4 Limitations of the correlation

The correlation presented in this paper is restricted to the prediction of burnout in water systems with vertical test sections and upflow. In addition, it is specifically directed toward burnouts with net exit bulk qualities. The correlation is also untried for tube diameters outside the range .18" to .306" and the L/D range from 50 to 76.

In addition to the above limitations of the data correlated, the form of the parameters involved have the characteristic of greatly amplifying experimental errors in certain ranges. These parameters are, to a certain extent, self compensating. For example, a positive error in exit quality produces a negative error in the ordinate and a negative error in the abcissa tending to bring the point back on the curve. The degree of self compensation is a function of pressure and quality as shown in fig 8. In general, increasing quality and decreasing pressure increase plotting errors. The relations shown in fig 8 represent the equations

$$\frac{d(\delta/R)}{\delta/R} = -\frac{\sqrt{y}}{2(1-\sqrt{y})}\left(1 - \frac{x\,\rho_{fg}}{\rho_o}\right)\frac{dx}{x} \qquad (6)$$

$$\frac{dr_o}{r_o} = -\frac{x\,\rho_{fg}}{\rho_o}\,\frac{dx}{x} \qquad (7)$$

Errors in the direct measurements of heat flux and flow rate are reflected in quality as a function of inlet subcooling.

$$\frac{dx}{x} = \frac{\Delta H}{\Delta H - \Delta h_i}\left(\frac{dq''}{q''} - \frac{dG}{G}\right) \qquad (8)$$

Where $\Delta H = 4\frac{L}{D}\frac{q''}{G}$ is the enthalpy rise in the test section and Δh_i is the difference between saturated liquid enthalpy (h_f of ref 3) and fluid inlet enthalpy. This relation indicates that very large errors in quality are possible at low qualities although the direct measurements are quite accurate. This explains the deviation of the highest points of the MIT 1500 psia data which have qualities of .06 and .006.

4.5 Prediction of peak heat flux from the correlation

Unfortunately, heat flux is implicit in S/R and Y_o so that it cannot be read directly from the curve for a given pressure and flow rate. A trial and error computation is required. However, experience with the correlation indicates that three trials are sufficient for an accurate graphical solution. Convergence on the predicted value is rapid and the value itself clearly defined. A graphical interpolation for one solution is given in figure 9.

5. NOMENCLATURE

q'' - heat flux,

G - specific flow rate,

D - cross sectional dimension,

R - one-half channel height,

γ - specific weight,

v - specific volume,

σ - surface tension,

δ - thickness of liquid film,

x - mass quality

y - volume quality

L - test section heated length

subscripts

o - test section outlet

i - test section inlet

f - saturated liquid

g - saturated vapor

fg - difference of property values at saturated liquid and saturated vapor.

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7. Personal Communication to J. A. Clark.

APPENDIX A

M.I.T. Burnout Data

Inlet Temp. (F)	Pressure (psia)	Flow Rate (lb/hr-ft^2)10^{-6}	Heat Flux (Btu/hr-ft^2)10^{-6}	Exit Quality
372	516	1.97	2.22	.15
356	565	1.10	2.05	.33
395	555	2.00	2.11	.17
380	585	1.94	2.07	.14
394	545	0.99	1.98	.42
308	545	2.04	2.54	.09
314	545	1.02	2.09	.31
249	545	2.12	2.80	.03
249	535	1.04	2.30	.30
476	1005	1.77	1.53	.145
474	1005	1.64	1.53	.155
409	1065	1.88	1.97	.072
485	1035	1.84	1.47	.138
463	1025	1.85	1.60	.123
509	1015	0.89	1.32	.41
461	1015	0.95	1.49	.35
357	1015	1.00	1.84	.24
336	1040	1.01	1.79	.19
318	1065	1.02	1.83	.16
259	999	1.04	2.08	.14
323	1015	1.02	1.81	.18
253	1010	2.13	2.56	subc.
232	1035	2.13	2.81	"
279	981	2.08	2.55	"
312	1020	2.05	2.39	"
347	1025	2.02	2.18	"
530	1520	1.76	1.25	.11
509	1515	1.76	1.37	.08
495	1525	1.79	1.45	.06
464	1525	1.88	1.60	.006
320	1510	1.02	1.67	.019
352	1485	0.986	1.52	.06
375	1505	0.99	1.61	.12
382	1505	0.964	1.47	.10
400	1510	0.954	1.45	.13
454	1515	0.938	1.33	.20
516	1511	0.86	1.14	.29
495	1510	0.881	1.21	.27
506	1550	0.882	1.15	.24
434	1533	1.89	1.59	subc.
438	1525	1.89	1.69	"
388	1525	1.94	1.97	"

APPENDIX B

M.I.T. Burnout Data
taken by J. Lehtinen

Inlet Temp. (^{o}F)	Pressure (psia)	Flow Rate $(lb/hr\text{-}ft^2)10^{-6}$	Heat Flux $(Btu/hr\text{-}ft^2)10^{-6}$	Exit Quality
420	1055	0.964	1.51	.25
436	993	0.961	1.62	.29
424	1015	1.91	1.80	.062
421	1053	1.91	1.82	.053
343	985	0.965	1.82	.20
538	985	0.866	1.41	.46
544	1045	0.859	1.29	.44
396	975	0.895	1.39	.38
502	1009	0.886	1.43	.41
497	1010	0.889	1.51	.43
499	1010	0.886	1.52	.44
542	1020	1.70	1.67	.20
509	1020	1.80	1.62	.21
465	539	0.936	1.82	.47
462	527	0.939	1.82	.47
461	585	1.86	1.87	.23
462	555	1.86	1.90	.24
427	575	0.971	2.00	.45
410	535	0.936	1.92	.42
387	550	0.962	2.01	.39
400	565	2.01	2.14	.16
414	575	1.93	2.12	.18
291	535	1.02	2.19	.30
298	555	2.13	2.59	.057
264	540	2.10	2.86	.036

APPENDIX C

ARGONNE NATIONAL LABORATORY

Burnout Data, .18" Tube

2000 psia, L/D = 64.5

Inlet Temp. (°F)	Flow Rate $(lb/hr\text{-}ft^2)10^{-6}$	Heat Flux $(Btu/hr\text{-}ft^2)10^{-6}$	Exit Quality
625	3.50	0.850	.094
613	3.56	1.00	.074
636	3.44	0.745	.121
595	2.57	0.995	.077
633	2.94	0.682	.118
586	3.30	1.15	.028
590	3.68	1.15	.016
627	3.50	0.820	.095
614	3.58	1.00	.077
635	3.47	0.742	.115
608	2.54	0.863	.091
613	2.51	0.845	.104
619	3.00	0.820	.090
587	3.58	1.14	.014
622	2.71	0.746	.101
623	2.68	0.735	.104
616	2.74	0.805	.091
613	2.88	0.799	.071
611	2.94	0.850	.072
612	2.90	0.840	.075
597	3.05	0.990	.048
598	3.10	1.00	.049
581	3.22	1.15	.019
624	2.04	0.658	.136
610	1.89	0.660	.102
624	1.86	0.660	.151

APPENDIX C (cont'd)

ARGONNE NATIONAL LABORATORY

Burnout Data .306" Tube

2000 psia, L/D = 76

Inlet Temp. (°F)	Flow Rate (lb/hr-ft^2)10^{-6}	Heat Flux (Btu/hr-ft^2)10^{-6}	Exit Quality
600	0.186	0.176	.495
631	0.188	0.169	.574
609	0.167	0.198	.676
628	0.210	0.196	.580
633	0.241	0.215	.580
615	0.183	0.258	.840
632	0.304	0.245	.515
584	0.314	0.276	.407
629	0.353	0.278	.491
525	0.230	0.328	.619
525	0.246	0.327	.560
633	0.435	0.318	.470
510	0.268	0.360	.525
504	0.226	0.357	.570
513	0.241	0.356	.620
599	0.398	0.356	.459
618	0.505	0.346	.384
507	0.618	0.587	.262
523	0.725	0.600	.218
580	0.441	0.393	.403
465	0.534	0.595	.380
627	0.550	0.346	

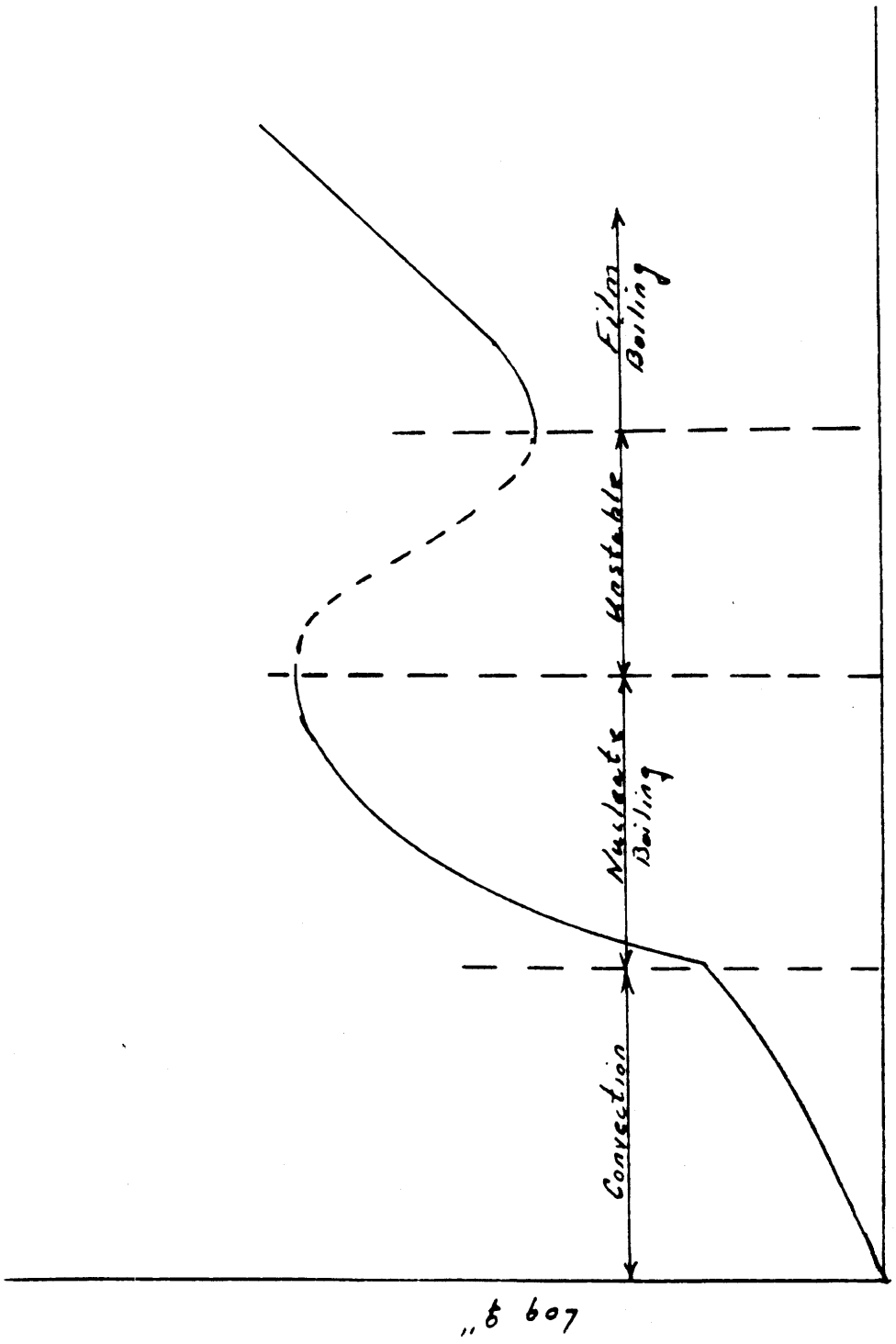
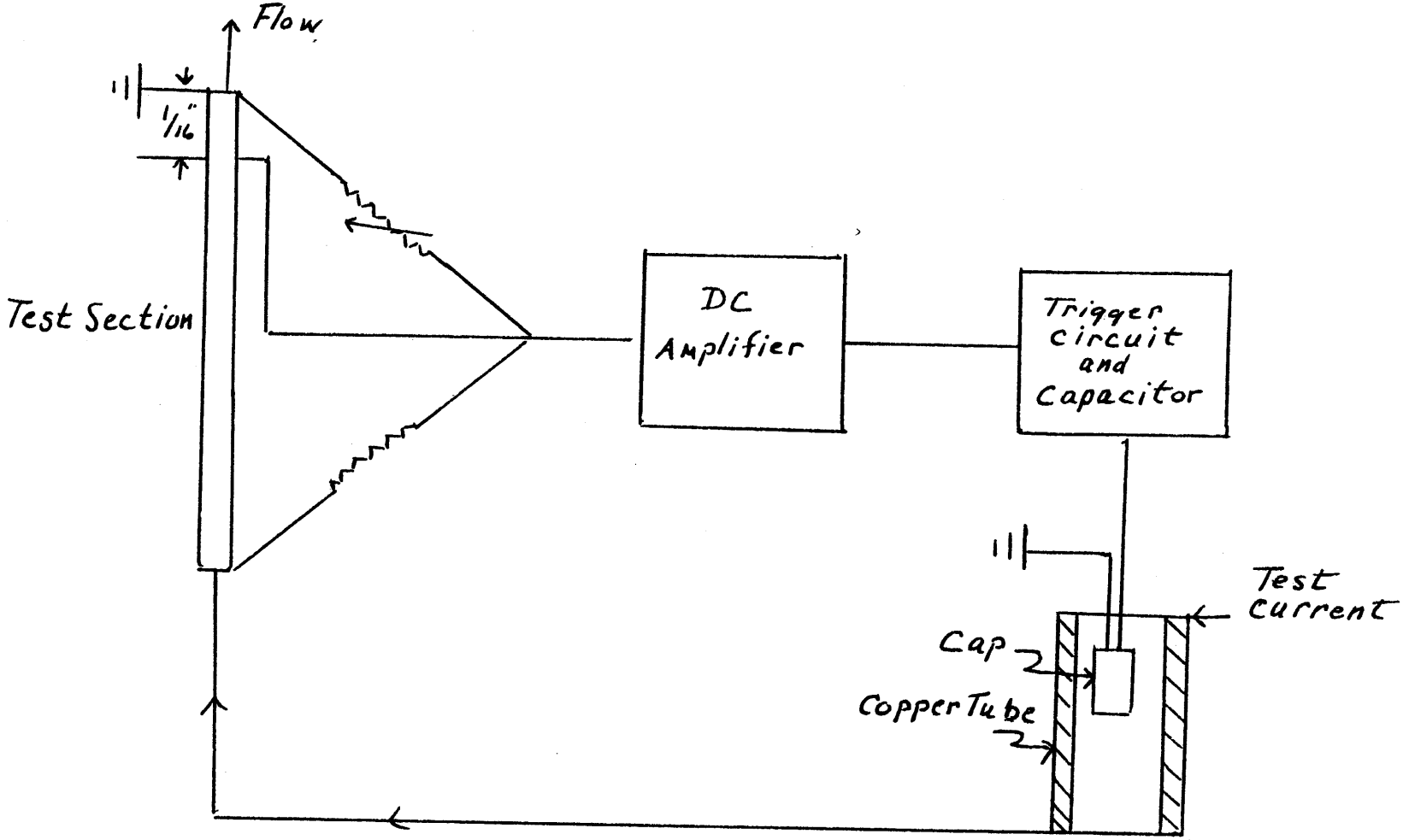


Fig 1 Characteristic Boiling Curve

FIG. 2. BURNOUT DETECTOR SCHEMATIC



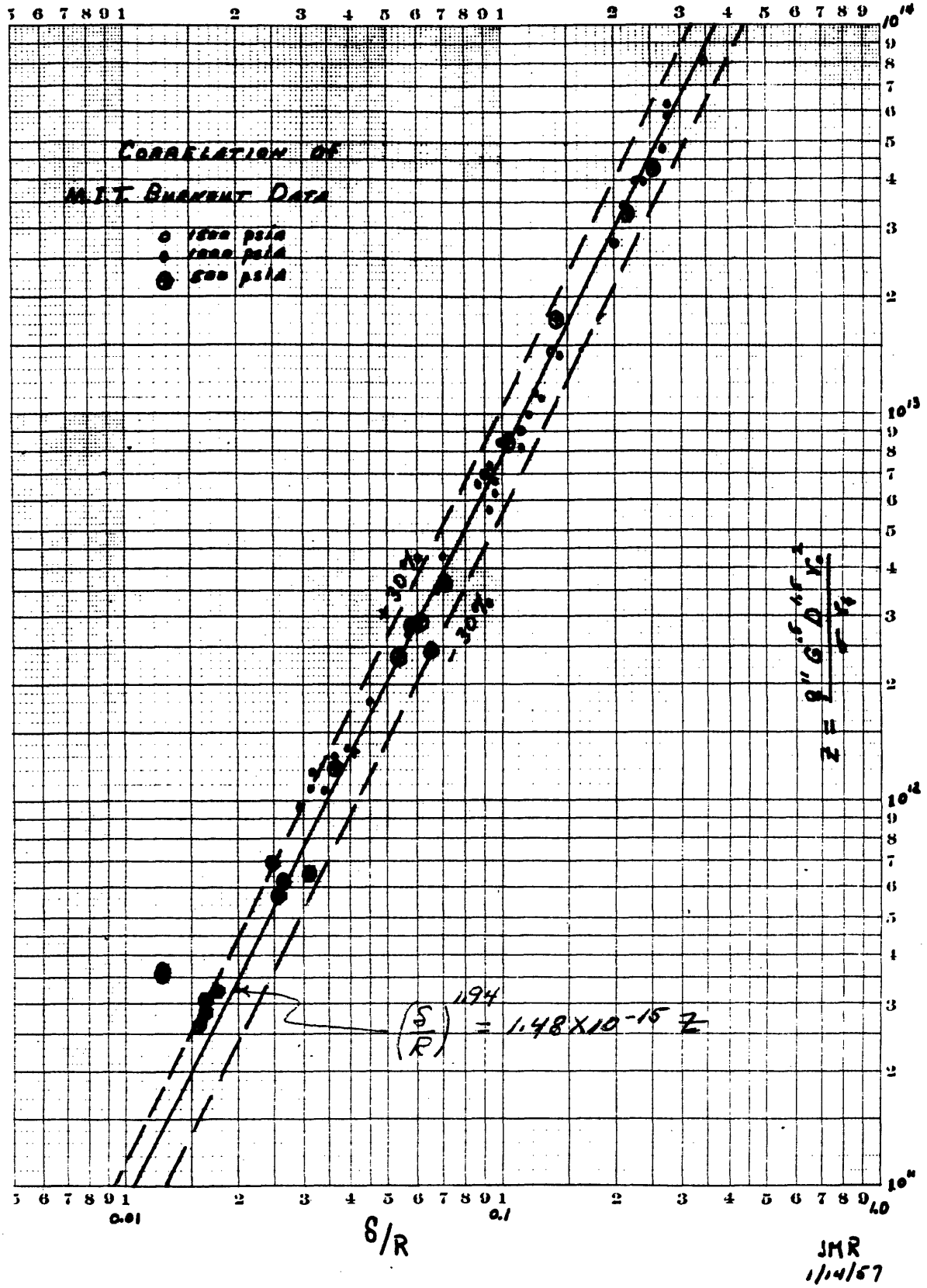
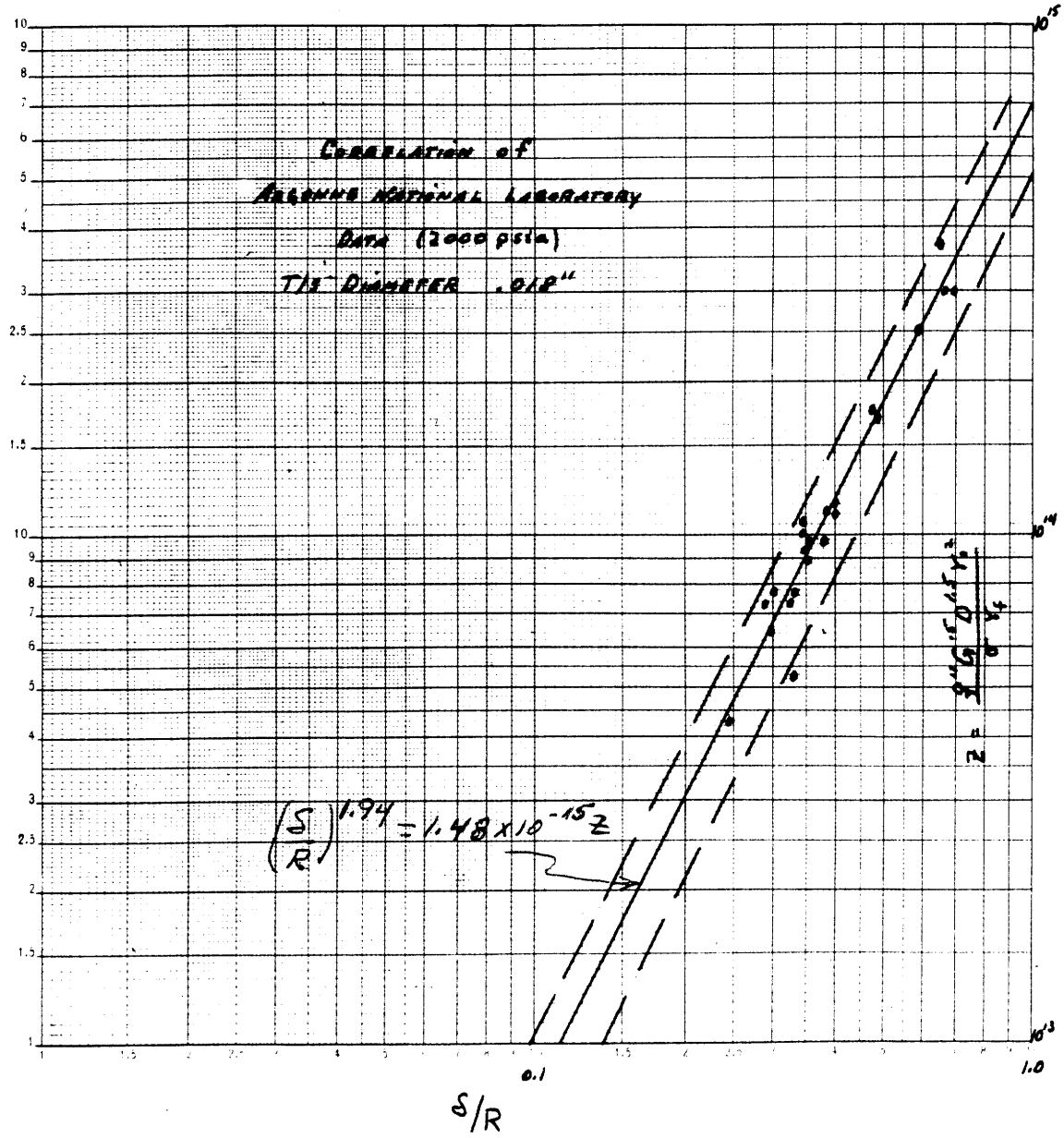
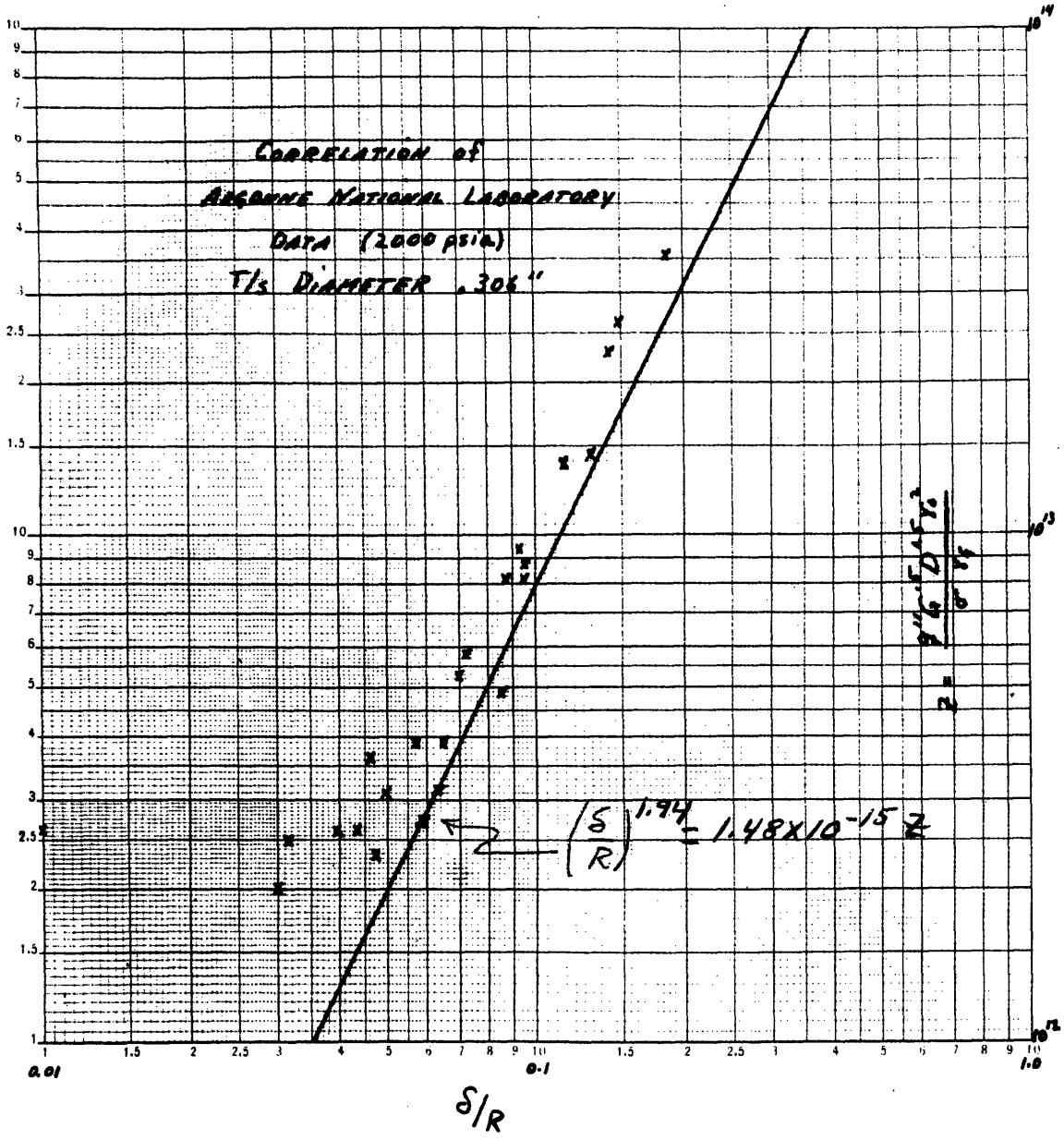

CORRELATION OF
M.I.T. BUREAU DATA
1500 psia
1000 psia
500 psia
$\left(\frac{S}{R}\right)^{1.94} = 1.48 \times 10^{-15} Z$
S/R
JMR
1/14/57

Fig. 3.



JMR
1/14/69

FIG. 4.



JMR
1/14/57

Fig. 5.

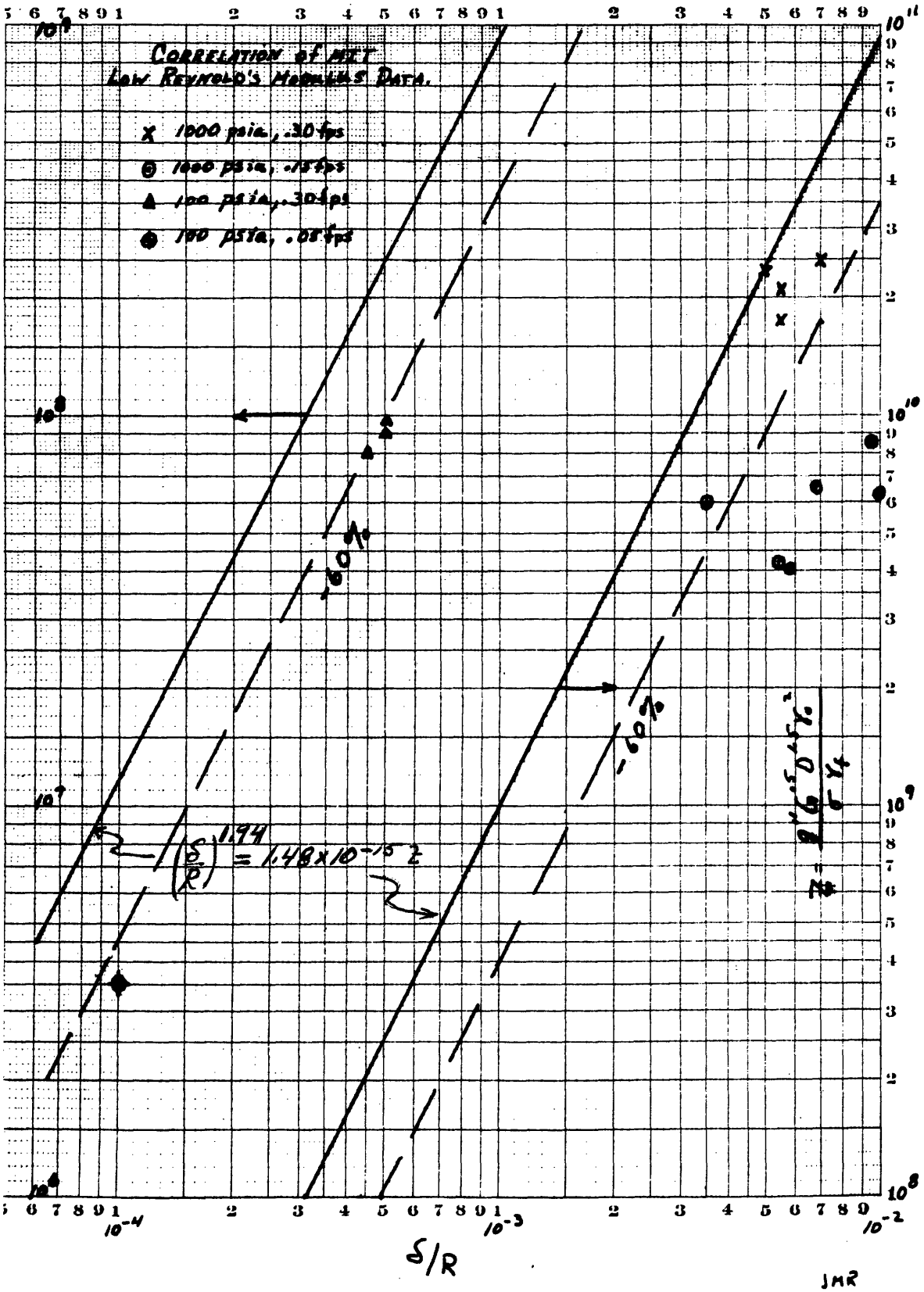
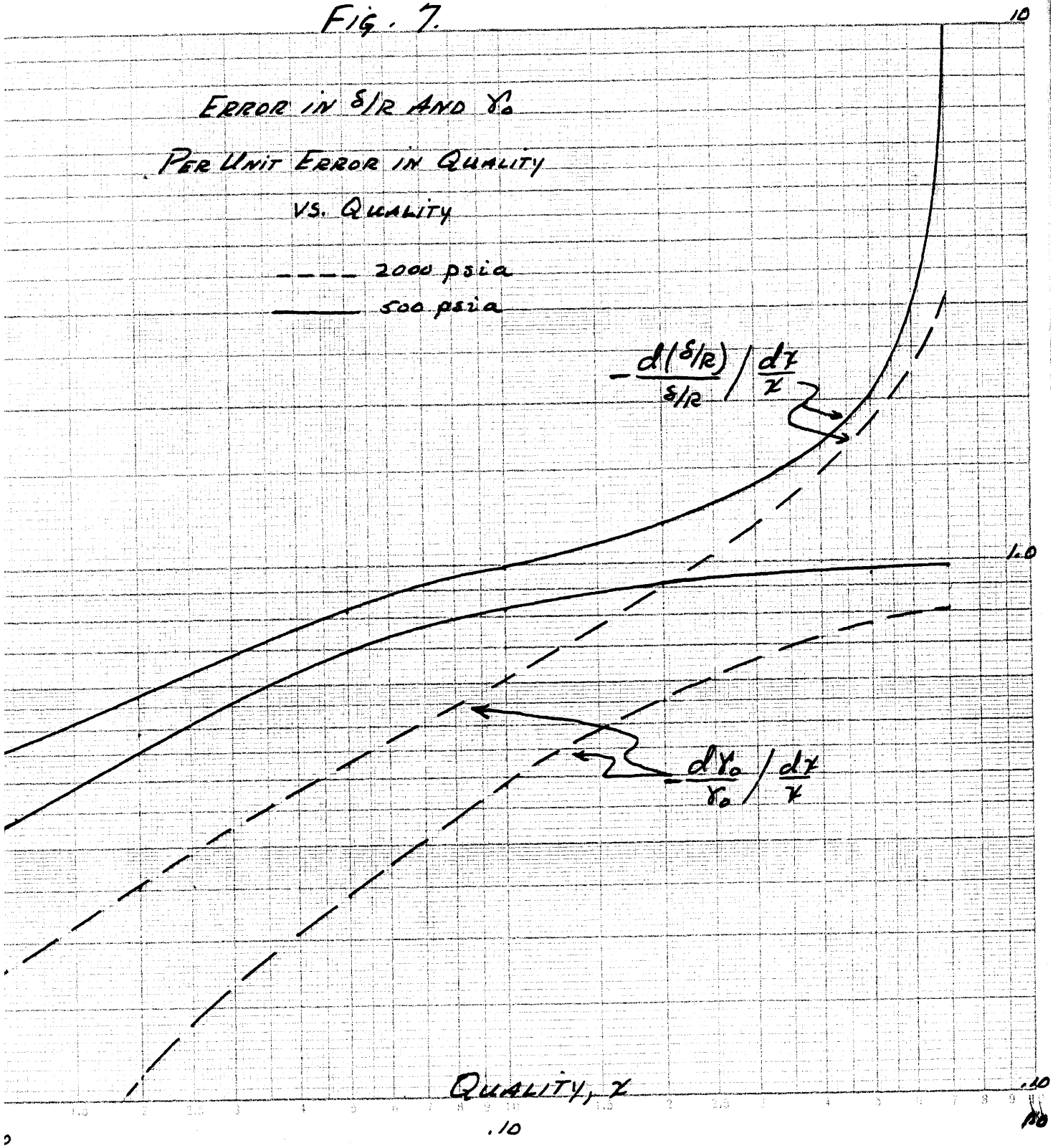

CORRELATION OF MIT
LOW REYNOLD'S NUMBER'S DATA.
X 1000 psia, .30 fps
1000 psia, .15 fps
100 psia, .30 fps
100 psia, .08 fps
$\left(\frac{\delta}{R}\right)^{1.94} = 1.48 \times 10^{-5} Z$
$\frac{\delta}{R} = 1.60 Z^{.515}$
$\frac{\delta}{R} = 1.60 Z^{.5}$
$\frac{\delta}{R} = 1.6 Z^{.5}$
10^{-4}
10^{-3}
10^{-2}
δ/R
JMR
11/14/57

FIG. 6

FIG. 7.


ERROR IN δ/R AND γo
PER UNIT ERROR IN QUALITY
VS. QUALITY
2000 psia
500 psia
$-\frac{d(\delta/R)}{\delta/R} \Big/ \frac{dx}{x}$
$-\frac{d\gamma_o}{\gamma_o} \Big/ \frac{dx}{x}$
10
1.0
.10
QUALITY, x
.10
1.0

Sample Calculation

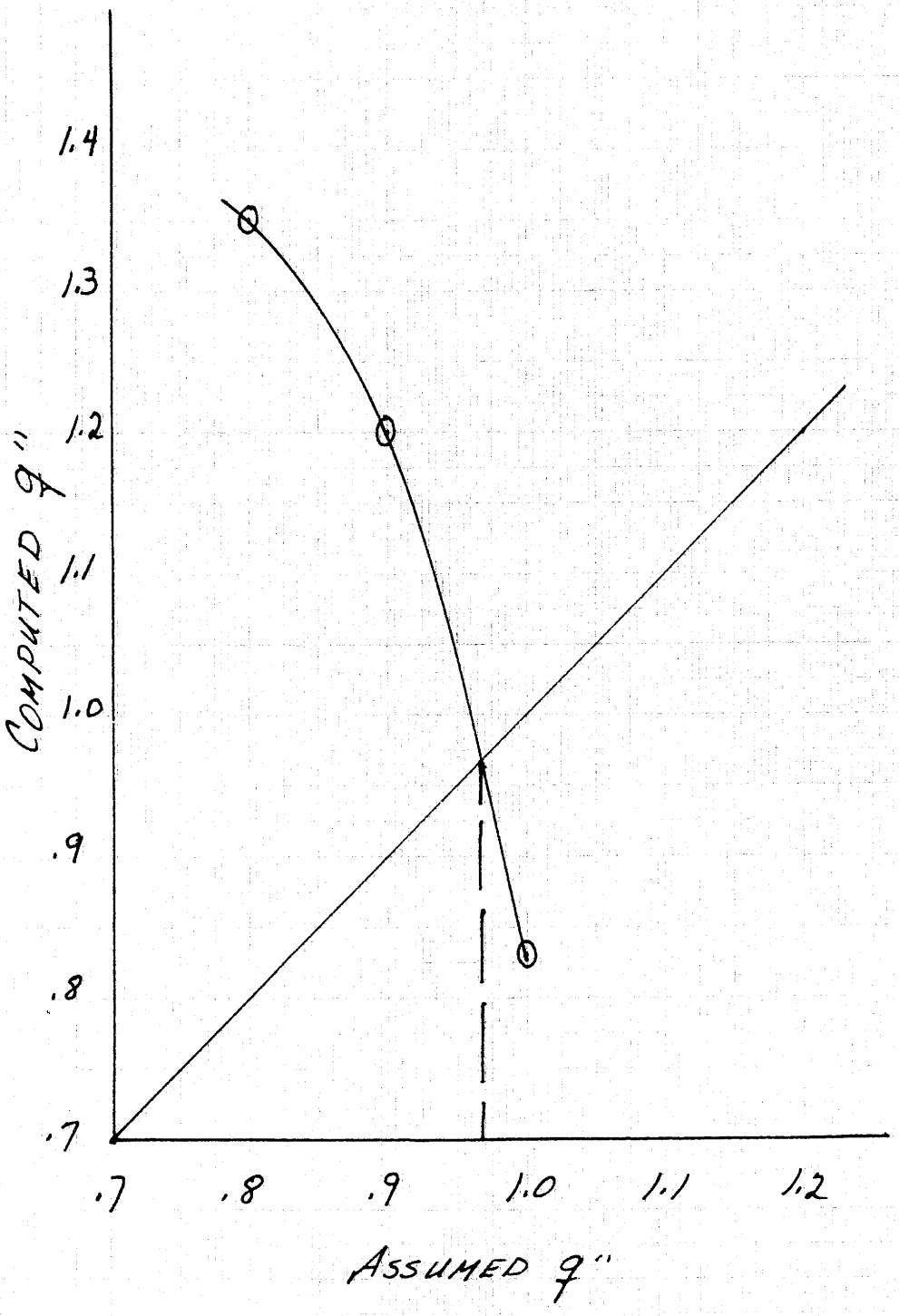


Fig 8