THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Progress Report

PRESSURIZATION OF LIQUID OXYGEN CONTAINERS

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UMRI Project 2646

under contract with:

DEPARTMENT OF THE ARMY
DETROIT ORDNANCE DISTRICT
CONTRACT NO. DA-20-018-ORD-15316
DETROIT, MICHIGAN

December 1958

EXPERIMENTAL APPARATUS

The system has been completely reassembled with the piston level indicator installed, and has been satisfactorily tested.

Temperatures inside the tank are measured at five different levels inside the tank but at only three levels on the tank wall due to a lack of recording channels in the oscillograph.

A 50-microfarad capacitor has been added to the piston-level-indicator electrical circuit to permit a 500% increase in sensitivity over the original circuit. The increased sensitivity has been obtained via the capacitor by removing the high surface noise to applied voltage ratio that was found at the higher sensitivities.

EXPERIMENTAL DATA

Two types of runs have been made during this last period. They are:

1. The tank has been filled with liquid nitrogen and upon pressurization to 35 psig a decrease in volume of the liquid-vapor mixture has been recorded (see Fig. 1) using the piston level indicator as the volume measuring device.

2. The tank has been filled, pressurized to 35 psig and then discharged and the mass of residual gas obtained.

Figure 1 is a sample of the exact recording as was obtained from the Sanborn Recorder. Curve A shows the load cell measurement, or weight of liquid nitrogen in the tank. At point 1, pressurization is initiated and its effect can be seen on the curves B,C, and D. Curve B, which is the piston level indicator, shows a sharp drop - from 15.22 in. to 14.56 in. as the tank pressure—curve C—increases from essentially 0 psig to 35 psig. Curve D shows the flow of pressurizing gas into the tank. Also, while the tank is at 35 psig, the data from curve D allow the direct measurement of the rate of condensation of the nitrogen gas on the internal tank surfaces, while at a constant pressure. These values of condensation rate with the piston can be compared to the rates previously obtained without the piston and the effectiveness of the piston as an insulating medium can be evaluated.

At point 3 on curve C, of Fig. 1, the pressure has been increased to 44 psig with no effective change of the piston level indicator (point 3 on curve B). Then with a decrease of pressure back to 35 psig, which is point 4, essentially no additional change in piston level is observed. At point 4, the tank is vented to the atmosphere and it can be seen that, as the tank pressure approaches the initial tank pressure, the piston level indicator approaches its original level of 15.22 in.*

Figure 2 is a plot of ΔL, the change in the level of liquid nitrogen upon pressurization to 35 psig vs. L, which is the height of liquid nitrogen before pressurization. Figure 2 provides a vehicle to interpret the effect of the piston as well as the tank bottom on the change in volume. It would be expected that, as the liquid mass (L on Fig. 2) approaches zero, the change in liquid level also should approach zero (ΔL on Fig. 2). By plotting these data and extrapolating to zero liquid mass an intercept of $\Delta L = 0.69$ in. was obtained. This value was assumed to be a constant value for all tests. Curve B, Fig. 3 was plotted with these data and is discussed below.

Figure 3 is a plot of the percentage change in volume of the boiling liquid nitrogen system upon pressurization to 35 psig vs. M, the mass of liquid nitrogen before pressurization. The curve A, Fig. 3, was obtained experimentally and can be seen to approach 100% change in volume as the mass of liquid nitrogen approaches zero. Upon re-evaluation of the data and accounting for the effects of the piston plus the effect of the semi-insulated bottom on the vapor content in the tank, curve B of Fig. 3 has been obtained.

Residual mass runs have been performed with hot water and ambient as the heat-exchanger medium as a means to vary the inlet gas temperature and the preliminary results have been shown in Table I.

From runs without the insulating piston, ambient heat-exchanger medium results in 1.13-lbm residual gas, hot water as exchanger medium results in 1.07-lbm residual gas mass. For complete data on the residual gas mass without a piston, see Progress Report Number 9. From these data, it appears that the piston has about a 3 to 10% influence on reducing the mass of residual gas. More data will be required to confirm this.

*The consequence of such measurements as shown in Fig. 1 is of considerable importance. This shows a very simple but precise method whereby a direct measurement can be made of the vapor volume in a boiling system. It is the first such measurement known to this laboratory. Such a method should be fully exploited to determine experimentally the vapor volume for boiling in various configurations as a function of height or filling, sub-cooling and wall heat flux.

TABLE I

Run No.	Heat-Exchanger Medium	Residual Mass	
41-A	Hot Water	0.982	Avg. 1.024 lbm
41-C	Hot Water	1.049	
41-D	Hot Water	1.041	
41-G	Ambient	1.036	Avg. 1.034 lbm
41-E	Ambient	1.077	
41-F	Ambient	0.990	

WORK DURING THE NEXT PERIOD

During the next period, calculations will be completed on the change in volume of the liquid nitrogen system upon pressurization.

Runs will be completed on the pressurization and discharge series with the inlet gas temperatures the significant parameter. The residual mass of gas will be determined and any change in residual mass due to the insulating piston will be discussed.

H. MERTE'S THESIS

A STUDY OF POOL BOILING IN AN ACCELERATING SYSTEM

CURRENT STATUS OF WORK

After correcting the leak around the edge of the heater surface, further tests were made to determine the effect of the mercury slip ring assembly on the accuracy of the temperature measurements in the rotating vessel. In accordance with the procedure specified in Progress Report No. 12, these tests have indicated that an emf is generated by the iron wires moving through the mercury bath. However, these emf's are consistent, and correction factors have been obtained as a function of rpm for the temperatures whose measurements must be precise. Figure 4 shows these corrections for three pairs of mercury channels. It is believed that part of the error is due to the difference in the adiabatic stagnation temperature of the mercury moving past the wire, with the two channels of a pair acting as an iron-mercury thermocouple circuit. Calculations of the error due to this effect are in agreement as regards the sign, but are too low in magnitude. An additional surface effect may be present. In any event the net effect can be determined by calibration.

With the procurement of a portable conductivity cell and bridge, success was attained in eliminating the deposits on the heating surface attendant with vigorous boiling. Greater stability of the heating surface temperature resulted. The thermocouples used for measuring the temperature of the water were supported in ceramic tubing, and it was found that the solubility of the tubing contributed most of the foreign deposits. These thermocouples have been removed since the saturation temperature at the heating surface is used as one of the parameters. By carefully redistilling the double-distilled water as purchased, the specific resistivity was increased from 0.5×10^6 to 1.5×10^6 ohm-cm. Measurements of the specific resistivity of the water are made in the test vessel immediately prior to and after each test run.

One test was run with $q/A = 100{,}000$ Btu/hr-ft^2 for a number of accelerations. A plot of the heater surface temperature minus the saturation temperature at the surface is shown as a function of time in Fig. 5. The shaded areas surrounding the test points indicate the maximum limits of error. It is noted that over the test period of 10 hours, $t_{wall} - t_{sat}$ decreased 0.6°F. In order to show the effect that acceleration has upon this parameter, it is necessary that compensation be made for this change. This was done by a test procedure which establishes a stable reference for each acceleration by taking readings at gravitational acceleration before and after the reading at each additional acceleration.

Figure 6 is a plot of the change between $t_{wall} - t_{sat}$ under acceleration and $t_{wall} - t_{sat}$ under gravitational acceleration as a function of the total acceleration. Because of a trend observed in earlier tests, data were obtained for two accelerations below 5 g's. These measurements pointed out more clearly the decrease in $t_{wall} - t_{sat}$ up to a total acceleration of about 4 times gravity, with a subsequent increase in $t_{wall} - t_{sat}$ for further increases in acceleration. As this is the first run in which this phenomenon occurred, further tests will be necessary to determine its reproducibility and the effect of heat flux.

Subsequent to the above test the heater failed, apparently due to a slow deterioration of the heater ribbon. Owing to the complexity of the heater assembly, rewiring will result in a three- to five-week delay.

WORK DURING THE NEXT PERIOD

When the heater has been repaired, the test herein reported will be repeated, and similar tests will be run at fluxes of q/A - 15,000, 25,000, and 50,000 Btu/hr-ft^2.

S. FENSTER'S THESIS

The thesis entitled The Transient Thermal Response of a Step Pressurized Boiling Liquid Nitrogen System has been completed. Because of its length, it is still in the processing stage and will be mailed under separate cover.

SUMMARY OF WORK DONE

Liquid nitrogen, boiling at constant wall heat flux varying from approximately 1200 to 4200 Btu/hr - ft^2 in a cylindrical container was rapidly pressurized with ambient nitrogen gas. Two levels of pressurization were used, 20 and 35 psig.

Primary instrumentation provided means for measuring wall and liquid temperature transients at various locations, nitrogen boiloff, and electrical power input.

It was found that upon pressurizing the boiling liquid, boiling ceased and thereupon followed a single-phase convection process, two-phase convection, and finally re-established boiling at the new pressure.

Assuming that the step-pressurization causes a step decrease in the heat-transfer coefficient, a theoretical analysis was made to predict wall and average liquid temperature transients.

Actually, the heat-transfer coefficient does not decrease stepwise upon pressurization. Its transient behavior is instead characterized by a rapid rate of decrease, a minimum point or dwell period depending upon the longitudinal location and flux, and finally a rapid increase until steady boiling is reestablished.

It was deduced that no bubbles have formed up to the time of the minimum heat-transfer coefficient. The rise of the coefficient after the minimum is attributed to the very first vapor bubbles forming and the associated fluid agitation.

During the period following pressurization until the minimum in coefficient is reached, only two effects were presumed to influence the transients. These are:

1. The viscous deceleration of the high liquid turbulence due to boiling before pressurization. This tends to decrease the heat-transfer coefficient.

2. The increasing contribution of free convection, since during the time interval under discussion the difference in temperature between the wall and the liquid, at a given location from the cylinder, is increasing.

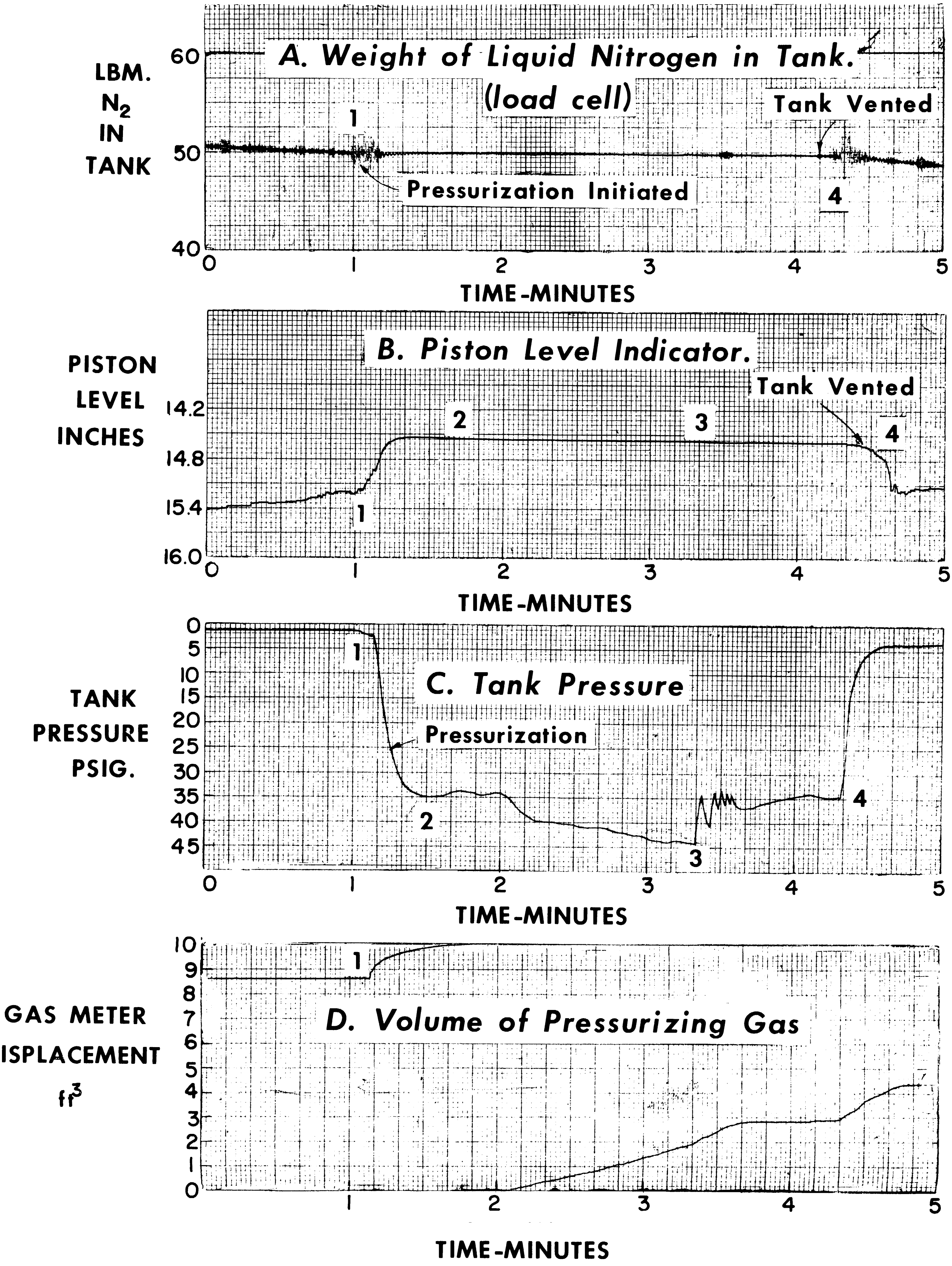
In order to prove the validity of the above description, it was necessary to subtract the free convection contribution from the experimentally determined heat-transfer coefficient for the period from pressurization to the minimum in coefficient. At the dwell period of the coefficient, it was reasoned that the effect of initial turbulence due to boiling had completely decayed, and the heat-transfer coefficient was due entirely to free convection. Since the usual correlation for free convection from vertical surfaces gave values of the heat-transfer coefficient which were too high, the constant in the correlation was adjusted on the basis of the experimentally determined minimum value. Using the new correlation, the free convection effect was subtracted from the transient coefficient and the remainder assumed to be due entirely to viscous deceleration of the liquid. This was supported by showing that the diminution of the contribution due to initial turbulence was a first-order effect, characteristic of viscous deceleration.

Assuming that the fluid motion inside the cylinder was characterized by an upward moving shell at the cylinder wall, and a downward moving core, and that no mass or heat diffusion occurs between the two, a method was devised to calculate approximately core velocities.

Such calculations showed that the liquid core velocity decreases as the cylinder bottom is approached, a result anticipated since the downward component of the core velocity must be zero at the bottom. This explains why the heat-transfer coefficients were found to decrease as the distance above the bottom decreases.

Calculation of the velocity history of the fluid at a particular longitudinal location showed a rapid deceleration of the liquid after pressurization. This serves to substantiate the diminution of the heat-transfer coefficient due to viscous effects.

SANBORN RECORD. FIG. NO. 1



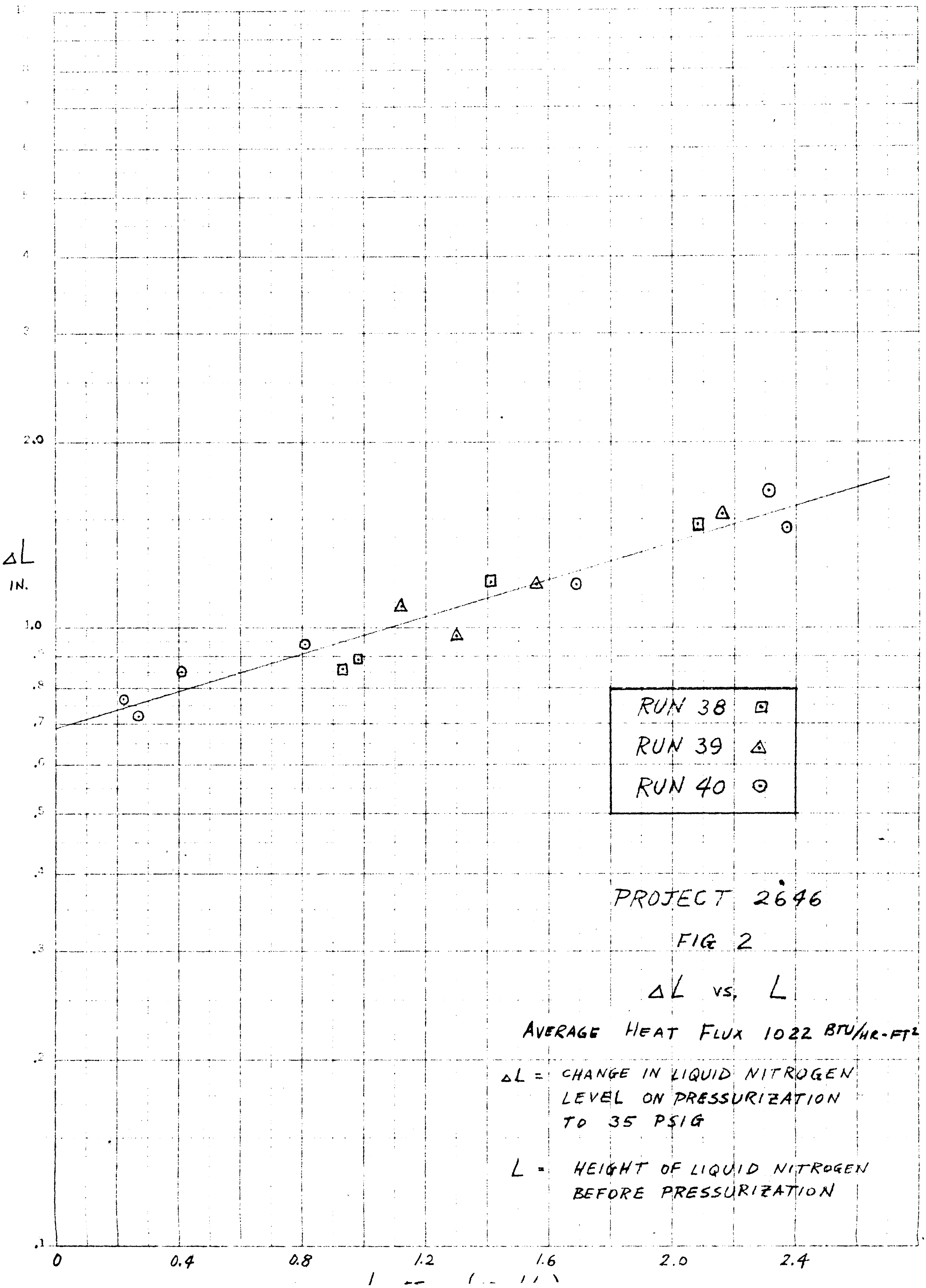


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FIG 2

ΔL vs. L

AVERAGE HEAT FLUX 1022 BTU/HR-FT²

ΔL = CHANGE IN LIQUID NITROGEN LEVEL ON PRESSURIZATION TO 35 PSIG

L = HEIGHT OF LIQUID NITROGEN BEFORE PRESSURIZATION

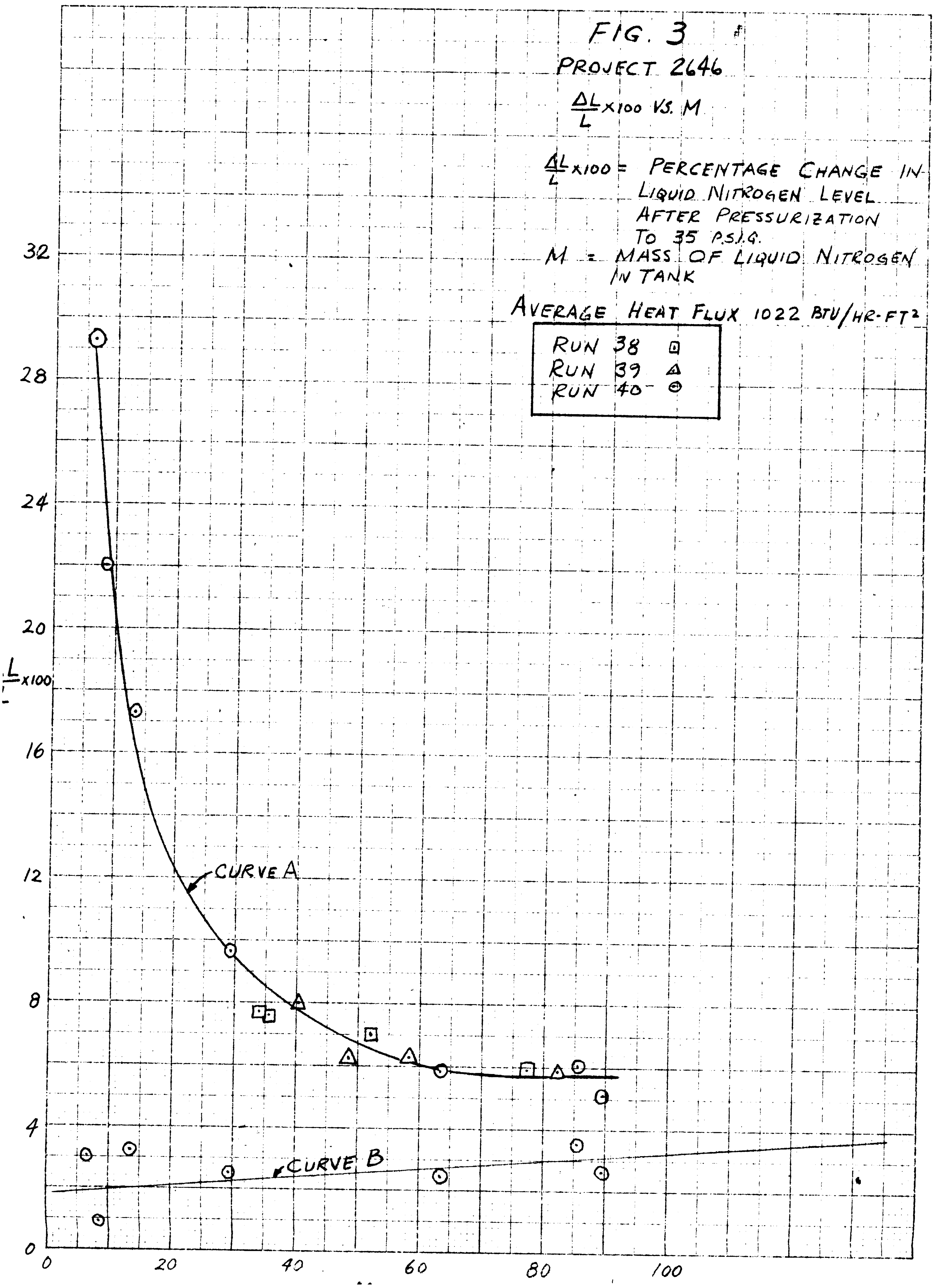
FIG. 3
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$\frac{\Delta L}{L} \times 100$ VS. M
$\frac{\Delta L}{L} \times 100$ = PERCENTAGE CHANGE IN LIQUID NITROGEN LEVEL AFTER PRESSURIZATION TO 35 P.S.I.G.
M = MASS OF LIQUID NITROGEN IN TANK
AVERAGE HEAT FLUX 1022 BTU/HR-FT²
RUN 38
RUN 39
RUN 40
CURVE A
CURVE B
32
28
24
20
16
12
8
4
0
20
40
60
80
100
L x100

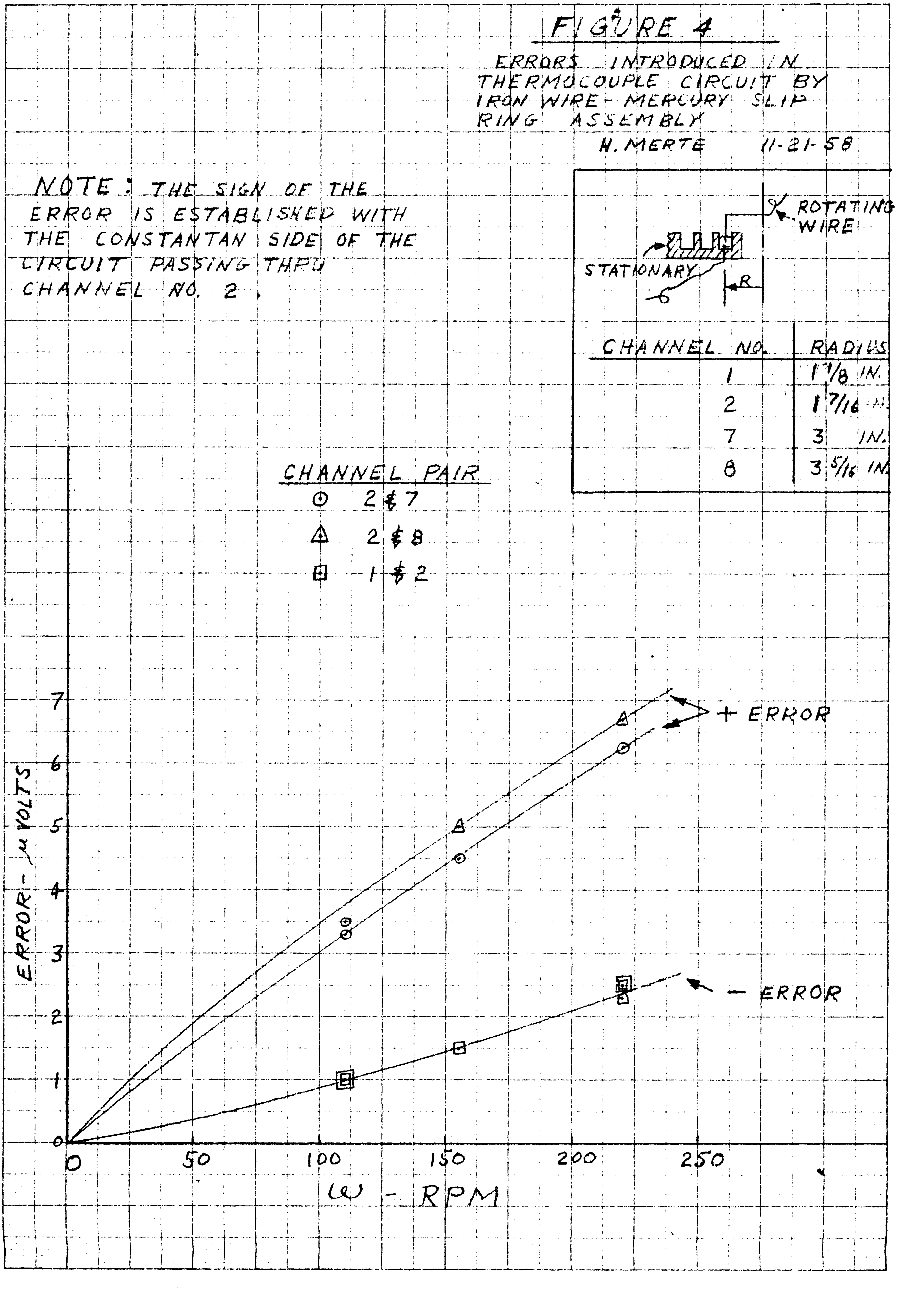
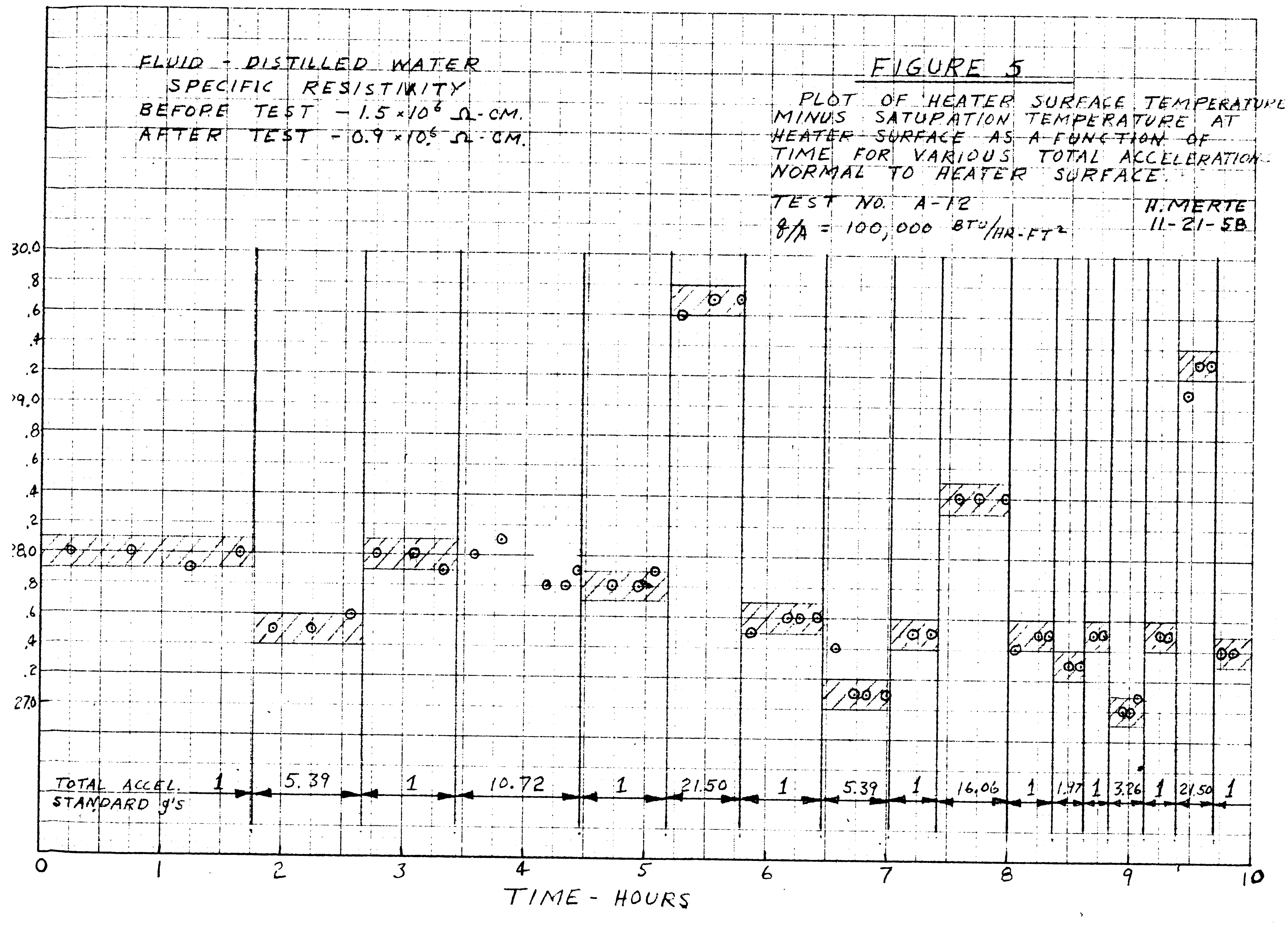


FIGURE 4
ERRORS INTRODUCED IN THERMOCOUPLE CIRCUIT BY IRON WIRE - MERCURY SLIP RING ASSEMBLY
H. MERTE 11-21-58
NOTE: THE SIGN OF THE ERROR IS ESTABLISHED WITH THE CONSTANTAN SIDE OF THE CIRCUIT PASSING THRU CHANNEL NO. 2.
ROTATING WIRE
STATIONARY
R
CHANNEL NO. | RADIUS
1 | 1 1/8 IN.
2 | 1 7/16 IN.
7 | 3 IN.
8 | 3 5/16 IN.
CHANNEL PAIR
⊙ 2 & 7
△ 2 & 8
⊡ 1 & 2
+ ERROR
− ERROR
ERROR - μ VOLTS
0
1
2
3
4
5
6
7
0
50
100
150
200
250
ω - RPM



FLUID - DISTILLED WATER
SPECIFIC RESISTIVITY
BEFORE TEST - 1.5 × 10^6 Ω-CM.
AFTER TEST - 0.9 × 10^6 Ω-CM.
FIGURE 5
PLOT OF HEATER SURFACE TEMPERATURE MINUS SATURATION TEMPERATURE AT HEATER SURFACE AS A FUNCTION OF TIME FOR VARIOUS TOTAL ACCELERATION NORMAL TO HEATER SURFACE.
TEST NO. A-12
Q/A = 100,000 BTU/HR-FT²
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11-21-58
30.0
.8
.6
.4
.2
29.0
.8
.6
.4
.2
28.0
.8
.6
.4
.2
27.0
TOTAL ACCEL. STANDARD g's
1
5.39
1
10.72
1
21.50
1
5.39
1
16.06
1
1.97
1
3.26
1
21.50
1
0
1
2
3
4
5
6
7
8
9
10
TIME - HOURS

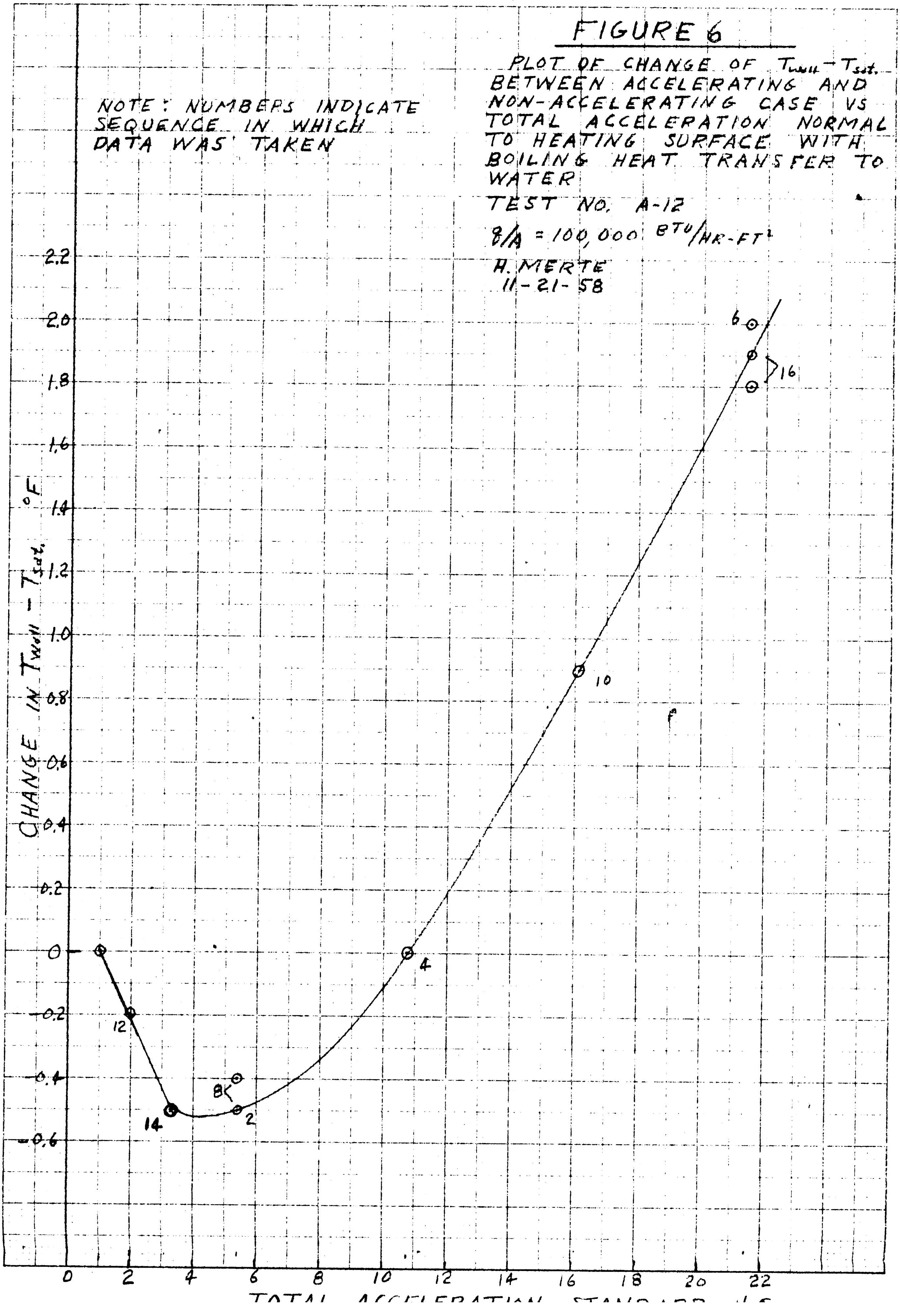

FIGURE 6

PLOT OF CHANGE OF $T_{wall} - T_{sat.}$ BETWEEN ACCELERATING AND NON-ACCELERATING CASE VS TOTAL ACCELERATION NORMAL TO HEATING SURFACE WITH BOILING HEAT TRANSFER TO WATER

TEST NO. A-12

$q/A = 100,000$ BTU/HR-FT²

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11-21-58