THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
Department of Mechanical Engineering

Progress Report

PRESSURIZATION OF LIQUID OXYGEN CONTAINERS

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EXPERIMENTAL APPARATUS

Efforts over the last period have been primarily directed toward the construction of the new system. The tank has been completely assembled, with thermocouples and piston level indicator in place and working satisfactorily.

Preliminary tests with water have been successful but were discontinued when the bushings that were designed for operation in liquid nitrogen seized during a run. This problem has been corrected by the use of glass tubes for bushings.

Difficulties were encountered when the piston was accidentally exposed to acetone vapors (see Fig. 7). To calculate an accurate inside heat-transfer coefficient, the inside of the tank had been cleaned with acetone. Acetone vapors remaining in the tank attacked the styrafoam piston when the piston was reinserted into the tank, causing it to buckle and break the thermocouple wires. A subsequent reconstruction of the internal system was required. This reconstruction is now 90% complete.

WORK DURING THE NEXT PERIOD

Efforts during the next period will be directed toward reassemblying the system and performing experimental measurements on residual gas mass using an insulated piston.

The reassembly will involve piping and sealing off the system to operate under pressure. Thermocouples and galvanometers will be calibrated, and the final calibration required for the piston level indicator will be made.

Experiments will be run, as in previously reported tests, using pressurizing gas temperature levels corresponding to liquid nitrogen, ambient, and boiling water.

H. MERTE'S THESIS

CURRENT STATUS OF WORK

A number of test runs have been made using the flat-plate heater both in the natural convection and boiling heat-transfer regions with accelerations varying from gravitational to approximately 20 times gravitational.

In addition, several attempts have been made to determine what effect, if any, the mercury slip-ring assembly (used for taking the thermocouple emf's from the rotating assembly) has on the accuracy of the temperature measurements. Results show that the measurements are accurate within at least 0.5°F, but greater precision is necessary for the boiling heat-transfer tests. The procedure was to allow the water to remain in the test vessel for a period of several days with no heat input so that equilibrium conditions are reached. With no heat input, temperature measurements were made alternating between no rotation and rotation, successively moving up to higher speeds. Figure 1 is a plot of the raw data for two of the thermocouples. Thermocouple TC-1 is located in the center of the copper heater 1/16 in. below the heating surface, and thermocouple TC-5 is located in the water 1/2 in. above the heating surface. The temperature of the water rises continuously during the test, but the temperature in the copper heater changes quite abruptly. An initial difference in temperature is noted, in spite of the fact that the assembly had been sitting for several days. The space between the inner test chamber and the outer container is filled with Fiberglas, but this evidently is not sufficient to insure complete equilibrium. Due to the thermal inertia of the system, a change in the ambient temperature does not cause an immediate change in temperature of the system. Upon rotation, however, it is possible that increased heat transfer takes place between the ambient and the test assembly, and internal convection currents may cause the difference between TC-1 and TC-5 to change as is noted. Attempts will be made to insulate the rotating assembly to a greater degree so that changes in ambient temperature will have less effect.

Two tests were run with water in the convective region. The results are plotted in Fig. 2 using the standard parameters of Nusselt No. and the Prandtl-Grashof product as coordinates. In calculating the Grashof number

$$\frac{L^3 \rho_f^2 \, g \, \beta_f \, \Delta t}{\mu_f^2}$$

the vector sum of the gravitational acceleration and the centrifugal acceleration at the heating surface was substituted for the gravitational term g. The acceleration was always normal to the heating surface. The diameter of the heating surface was used for the geometrical factor L. The correlation recommended in M. M. McAdams' Heat Transmission* for a horizontal heated surface is included for comparison as the solid curve.

The heating surface is the end of a 3-in.-diam cylinder, to which is attached a thin stainless-steel skirt so that a continuous surface is presented to the fluid. Since only the center section is heated, strong bulk convection currents are set up in the vessel. To determine the effect of having the entire bottom surface as the heater and thus eliminate these currents, a stainless-steel cylindrical shell having a small bottom clearance and the same diameter as the heater was placed over the heater. The results of this test are plotted in Fig. 3, and again the correlation from McAdams is included for reference. The increased Nusselt number is to be noted for this configuration. This effect is to be given further study.

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*McGraw-Hill, New York, third edition.

Figure 4 is a plot of the bulk water temperature and ΔT for the above three tests versus the acceleration normal to the surface. It should be noted that, for any given test, the bulk water temperature decreased with increasing acceleration. This is believed to be due to the increased convection occurring in the vapor space above the liquid, carrying the heat more rapidly away from the liquid to the cooling coils built into the upper section of the test vessel.

With the boiling heat-transfer tests, it has not been possible as yet to achieve a stable surface condition. Figure 5 is a plot of T_w-T_{sat} versus acceleration for one of the tests and shows the effect of the unstable condition. The numbers along the data points indicate the sequence in which they were taken. Each point is the average of two readings which never varied by more than 0.2°F, and for the majority were within 0.1°F. The test covered a total period of about 12 hours. T_w is the temperature of the heating surface, obtained by extrapolating the measured temperature 1/16 in. below the surface, and T_{sat} is the saturation temperature at the heating surface. In this way compensation is made for the increase in the hydrostatic head due to the acceleration.

After several tests with heat fluxes such that vigorous boiling occurred, it was observed that the chrome-plated heating surface was dotted with rust-colored spots. It was believed that contamination of the water resulted either from the corrosion of the stainless-steel inner tank at the welded seam, or from the copper condensing coils in the upper section. Upon chrome-plating these parts, the rust-color was largely eliminated, but white spots still remained. Figure 6 shows these discolorations. The bare spot on the left side of the heating surface was made by rubbing the surface with a finger, indicating that the discolorations were deposited and did not come through the chrome plating. The water in all the tests was double-distilled water purchased locally. It was delivered in 5-gallon carboys, and most likely the white discolorations are due to silicates which dissolved from the soft glass containers. Pyrex apparatus for distilling water has been obtained and the water will be redistilled immediately prior to use. Further, samples of the water will be taken before and after tests for conductivity measurements to determine the presence of contaminants.

The heating surface recently developed a leak between the copper cylinder and the stainless-steel skirt. Attempts are being made to eliminate this. Also, the neoprene gasket between the lower surface and the sidewalls has been replaced with teflon.

WORK DURING THE NEXT PERIOD

When the leaks in the heater surface have been corrected, further attempts will be made to determine the effect of the mercury slip-ring assembly on the accuracy of the temperature measurements. Additional runs will then be made with boiling heat transfer with various heat fluxes and accelerations.

S. FENSTER'S THESIS

CURRENT STATUS

During the report period just passed, experimental runs were made at heat fluxes from approximately 1000 Btu/hr-ft^2 to 4500 Btu/hr-ft^2. The flux through the walls of the primary liquid nitrogen container was maintained constant throughout the entire run, that is, throughout initial boiling at atmospheric pressure, pressurization, single-phase free convection, two-phase free convection, and re-established steady boiling at the new pressure. The magnitude of the pressurization was 20 and 35 pounds per square inch gage. The maximum heat flux used was approximately eight to ten times the flux available from ambient heating alone.

The following phenomena were observed:

1. Immediately upon pressurization of the boiling liquid, boiling ceases, the liquid now being subcooled with respect to the new pressure.

2. Immediately upon pressurization, the wall temperatures undergo a transient increase, the rate of which depends upon the magnitude of the heat flux.

3. After pressurization, the temperatures in the liquid begin to increase, their rates of increase depending upon the magnitude of the heat flux.

4. The higher in the liquid from the bottom of the cylinder, the greater the rate of increase of the liquid temperature with respect to time.

5. The wall temperatures observed during the steady-state boiling, before and after pressurization, are highest at the bottom-most thermocouple, which is located 2 in. from the cylinder bottom.

6. The same phenomenon as described in (5) was observed generally during the period immediately after pressurization and up to the time when boiling is resumed at the new pressure.

7. First boiloff occurs at greater time after pressurization as the magnitude of the heat flux is decreased and as the magnitude of the pressurization is increased.

8. There are no significant axial or radial temperature gradients measurable by thermocouples located at the centerline and the wall, and at five axial locations, during the steady-state boiling periods.

9. There are no significant radial gradients, as measured in the experiment during the period in which the liquid is heating up after the pressurization.

10. At a given value of the heat flux, the difference in temperature between the wall and the liquid decreases as the pressure increases, during steady boiling.

11. The higher the thermocouple is located from the bottom of the cylinder, the sooner a noticeable hump in the wall-temperature-vs.-time curve appears.

12. After the wall temperature passes the hump, there are no changes.

13. Less pressurizing gas is required the higher the heat flux and the lower the magnitude of the pressurization.

Figures 8 and 10 show the wall and liquid temperature transients for two different heat fluxes with pressurization to 20 psig. They are representative of higher flux runs, and also runs at 35 psig, in that the liquid transients show the same characteristic axial temperature gradient. The wall temperatures at different conditions of flux and pressure show the same "e-power" response and humps as shown.

The curves of heat-transfer coefficient versus time are shown in Figs. 9 and 11. The coefficient for the lowest axial location in the tank is the lowest in magnitude.

The curves for heat flux versus time for first boiloff are given in Figs. 12 and 13.

WORK DURING THE NEXT PERIOD

In the next report period, a theoretical analysis will be completed to approximate the wall and liquid temperature transients. In addition, an attempt will be made to interpret the slope of the curve of heat flux versus time for first net boiloff on log-log coordinates. All the data should be completely processed in the next report period. An attempt will be made to interpret these data in a generalized form.

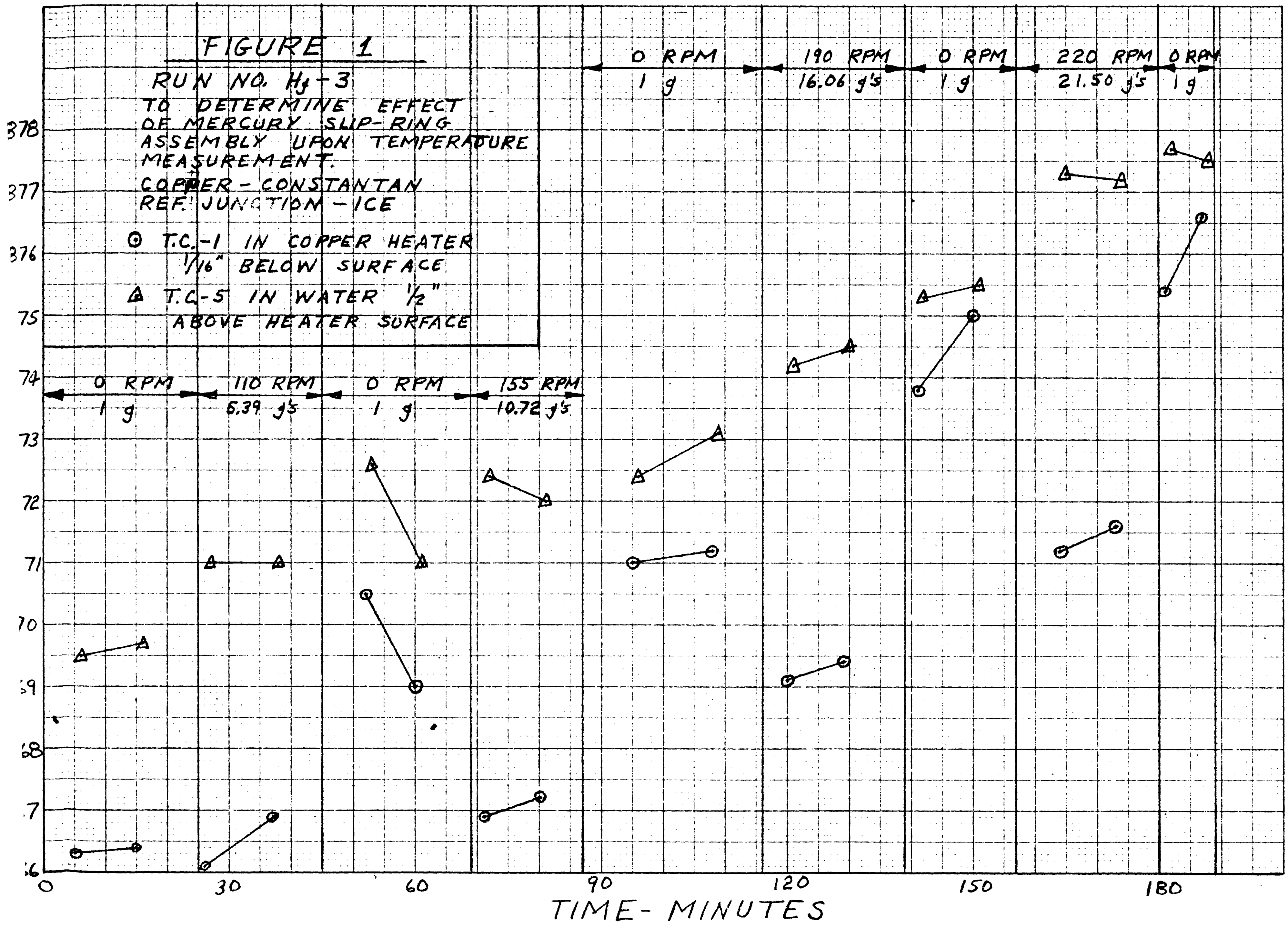


FIGURE 1
RUN NO. Hg-3
TO DETERMINE EFFECT OF MERCURY SLIP-RING ASSEMBLY UPON TEMPERATURE MEASUREMENT.
COPPER - CONSTANTAN
REF. JUNCTION - ICE
T.C.-1 IN COPPER HEATER 1/16" BELOW SURFACE
T.C-5 IN WATER 1/2" ABOVE HEATER SURFACE
0 RPM
1 g
110 RPM
5.39 g's
0 RPM
1 g
155 RPM
10.72 g's
0 RPM
1 g
190 RPM
16.06 g's
0 RPM
1 g
220 RPM
21.50 g's
0 RPM
1 g
378
377
376
75
74
73
72
71
70
69
68
67
66
0
30
60
90
120
150
180
TIME- MINUTES

PLOT OF N_{Nu} VS $N_{Gr} \cdot N_{Pr}$ FOR NATURAL CONVECTION FROM FLAT CIRCULAR HEATING SURFACE TO WATER WITH SYSTEM SUBJECTED TO ACCELERATIONS OVER RANGE FROM GRAVITY (32.17 ft/sec²) TO 691 ft/sec²

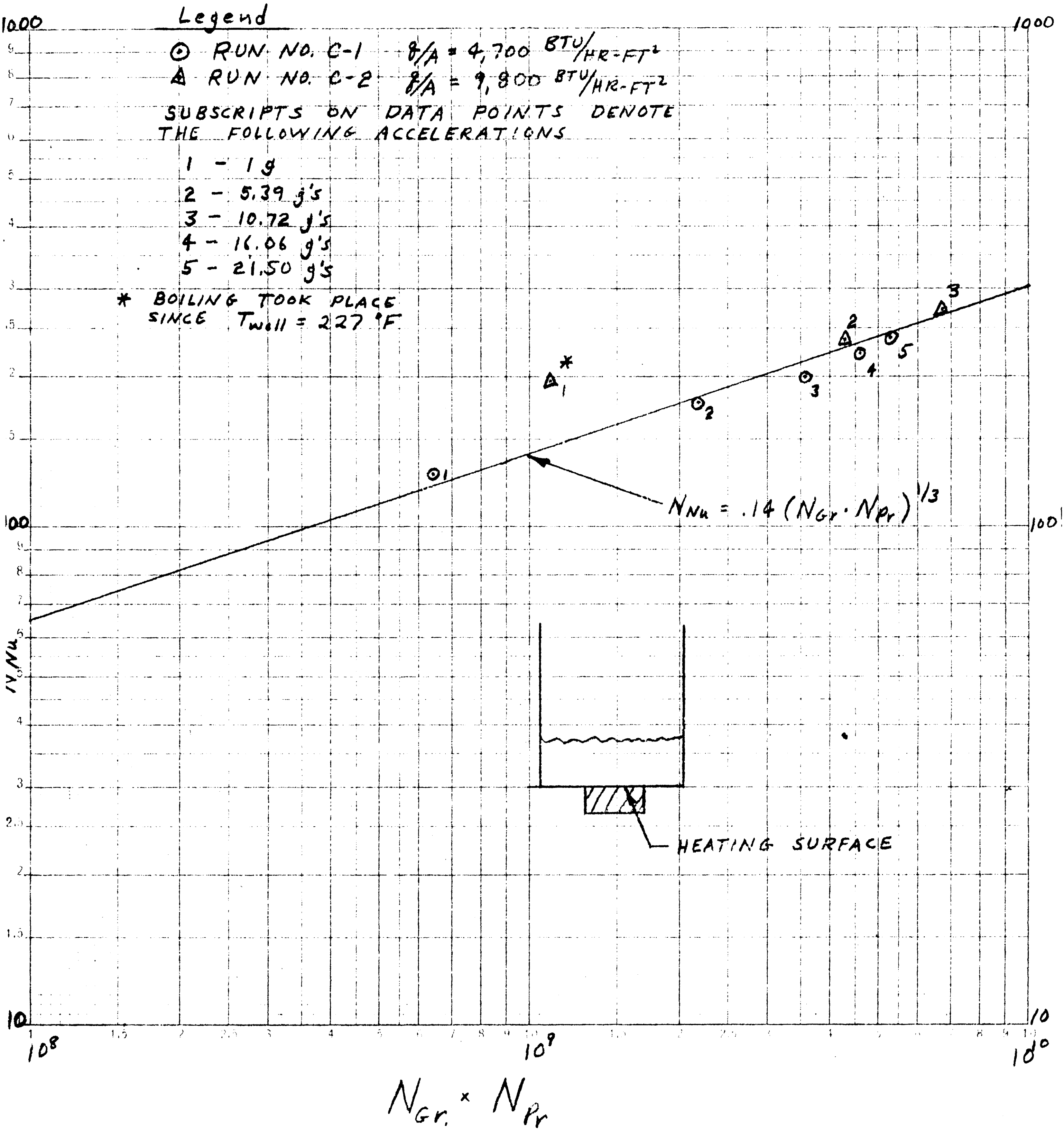


FIGURE 2

PLOT OF N_{Nu} vs. $N_{Gr} \cdot N_{Pr}$ FOR NATURAL CONVECTION FROM FLAT CIRCULAR HEATING SURFACE WITH A FLOW GUIDE TO WATER WITH SYSTEM SUBJECTED TO ACCELERATIONS OVER RANGE FROM GRAVITY (32.17 ft/sec²) TO 691 ft/sec²

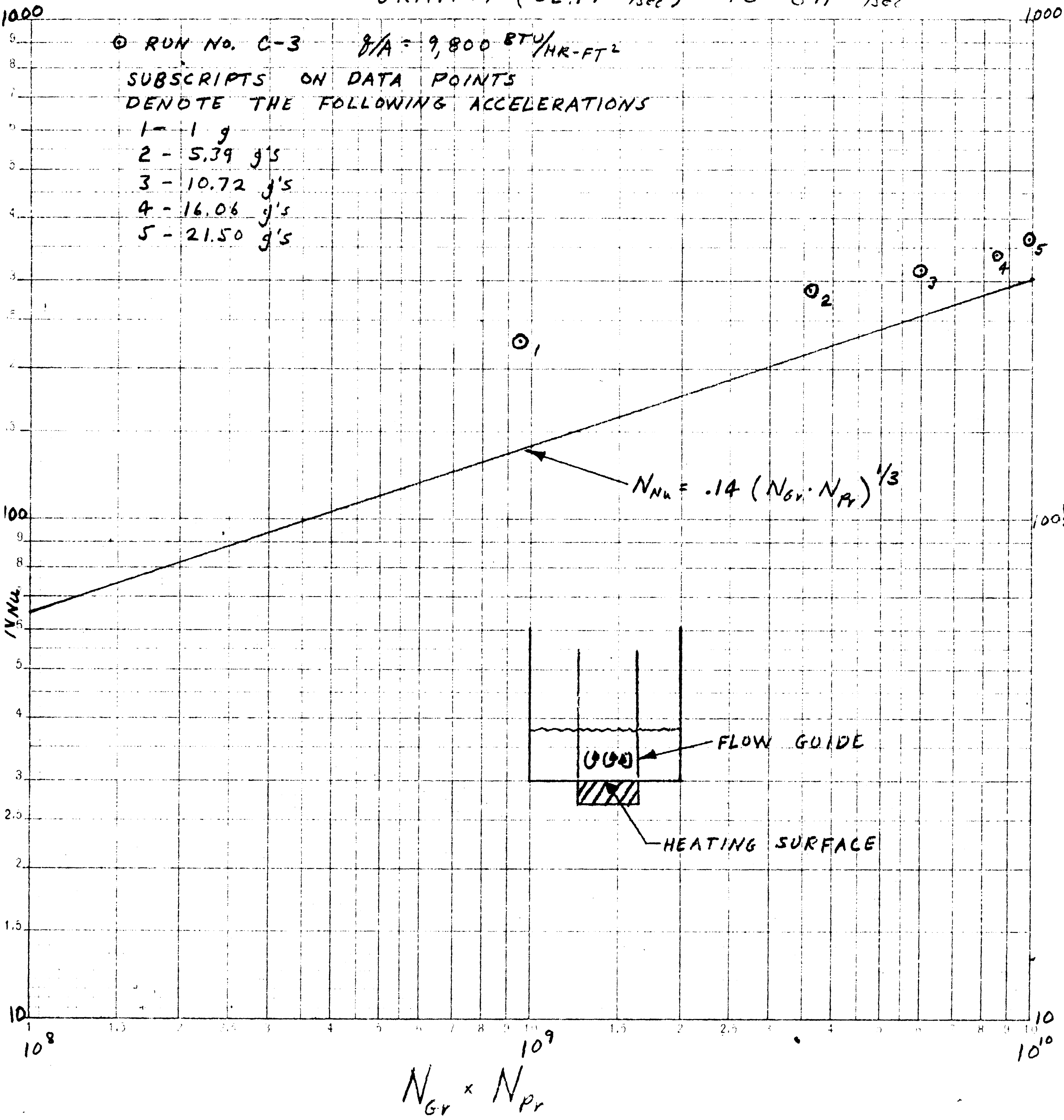


FIGURE 3

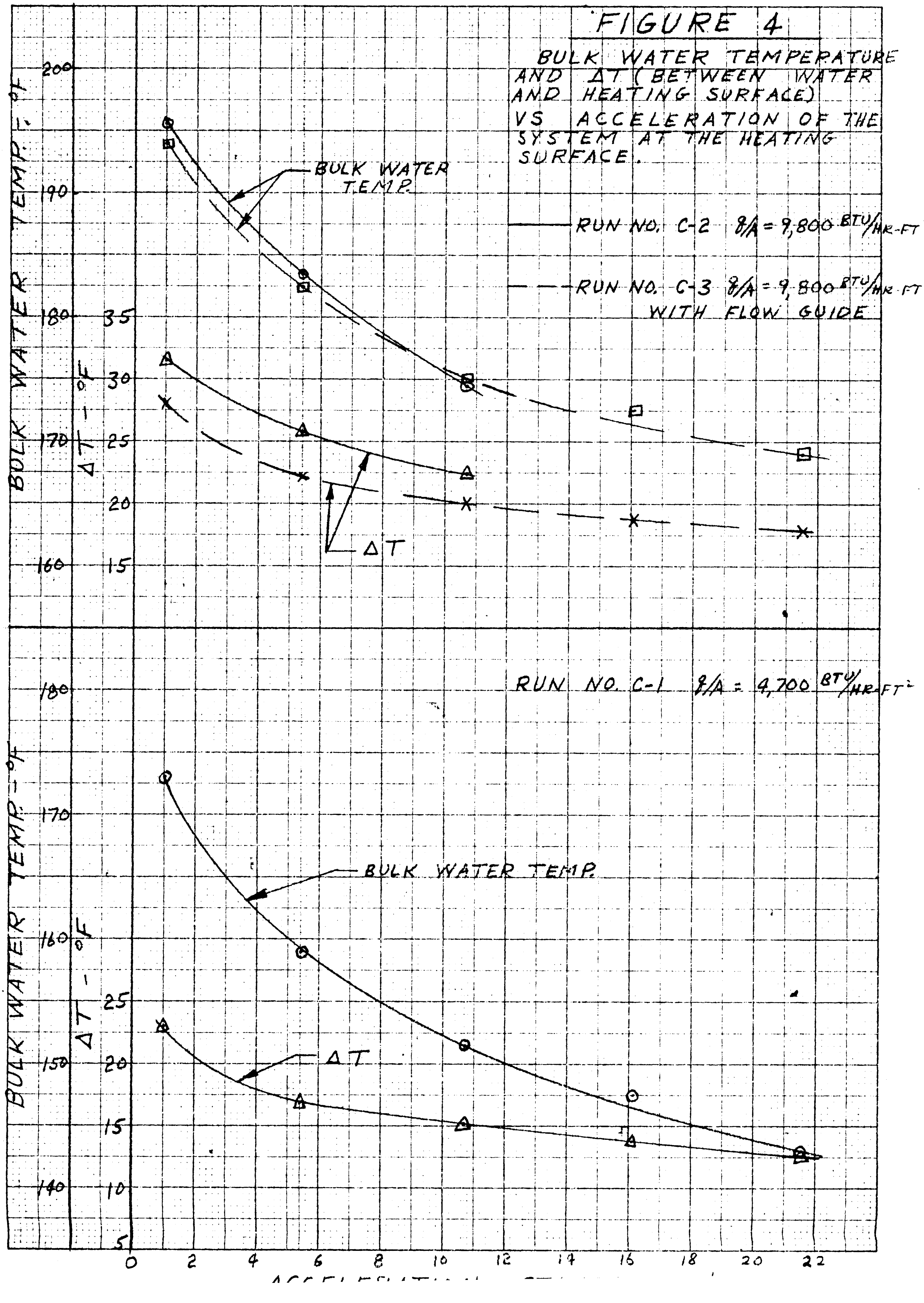


FIGURE 4
BULK WATER TEMPERATURE AND ΔT (BETWEEN WATER AND HEATING SURFACE) VS ACCELERATION OF THE SYSTEM AT THE HEATING SURFACE.
BULK WATER TEMP.
RUN NO. C-2 q/A = 9,800 BTU/HR-FT²
RUN NO. C-3 q/A = 9,800 BTU/HR-FT² WITH FLOW GUIDE
ΔT
BULK WATER TEMP. - °F
ΔT - °F
RUN NO. C-1 q/A = 4,700 BTU/HR-FT²
BULK WATER TEMP.
ΔT
BULK WATER TEMP. - °F
ΔT - °F
200
190
180
170
160
35
30
25
20
15
180
170
160
150
140
25
20
15
10
5
0
2
4
6
8
10
12
14
16
18
20
22

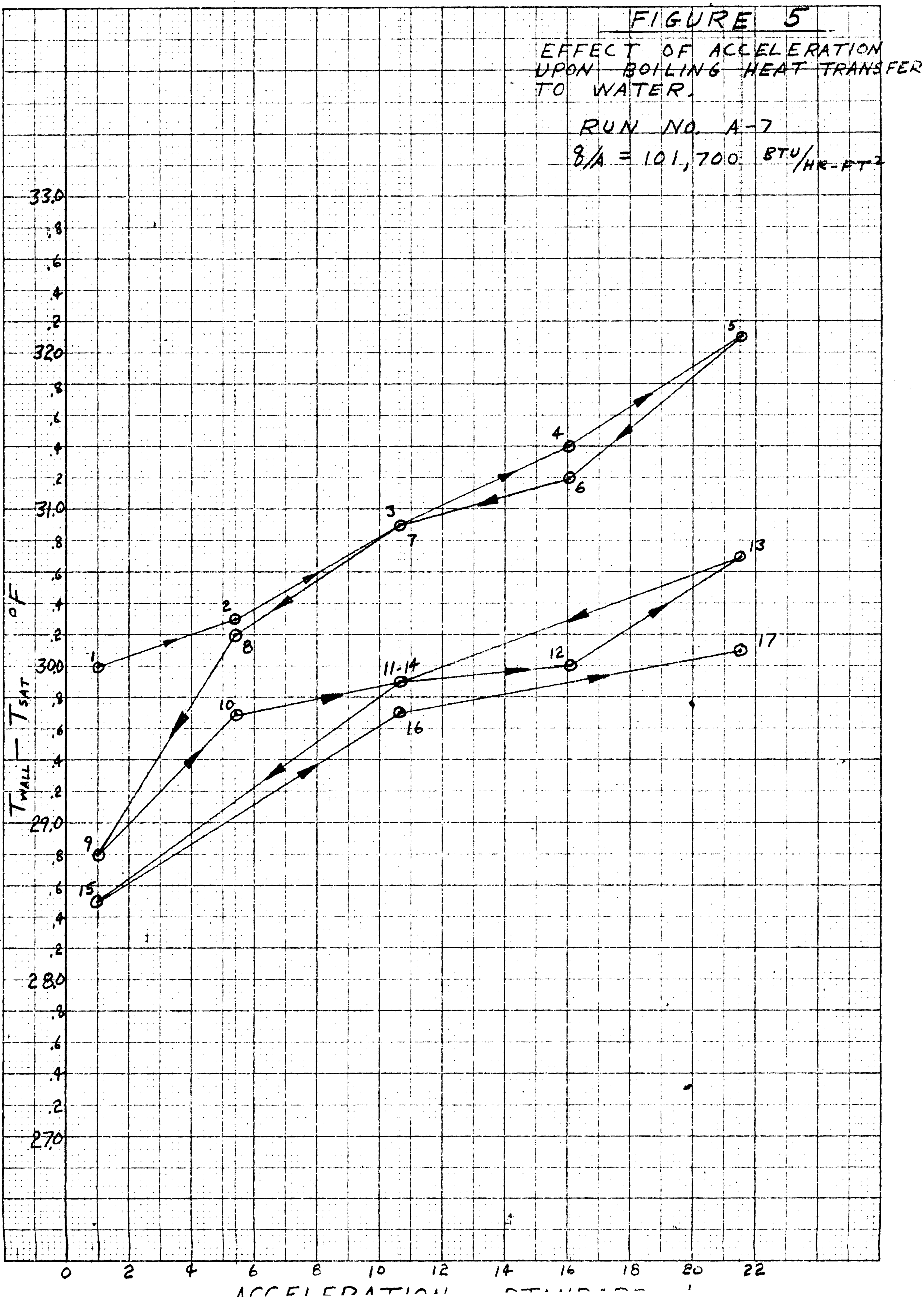


FIGURE 5

EFFECT OF ACCELERATION UPON BOILING HEAT TRANSFER TO WATER.

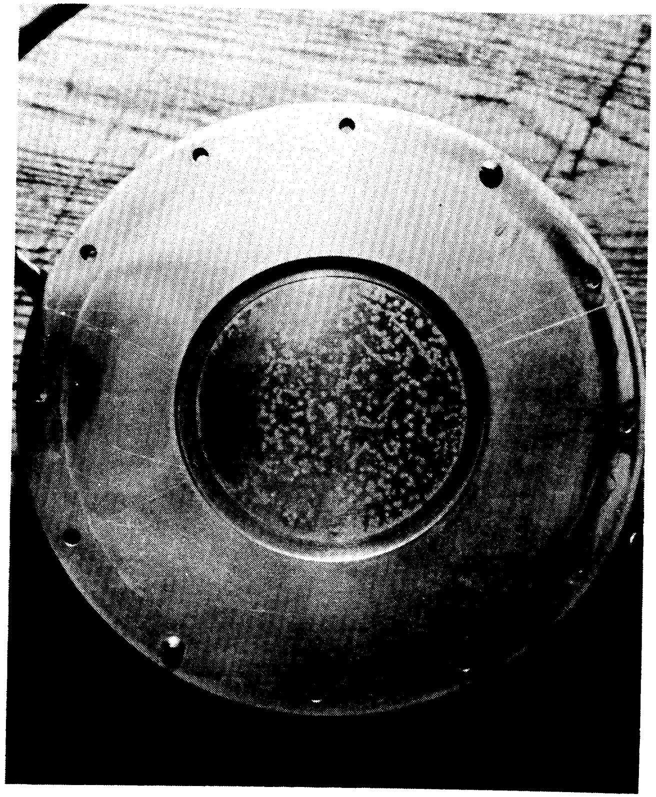


Fig. 6

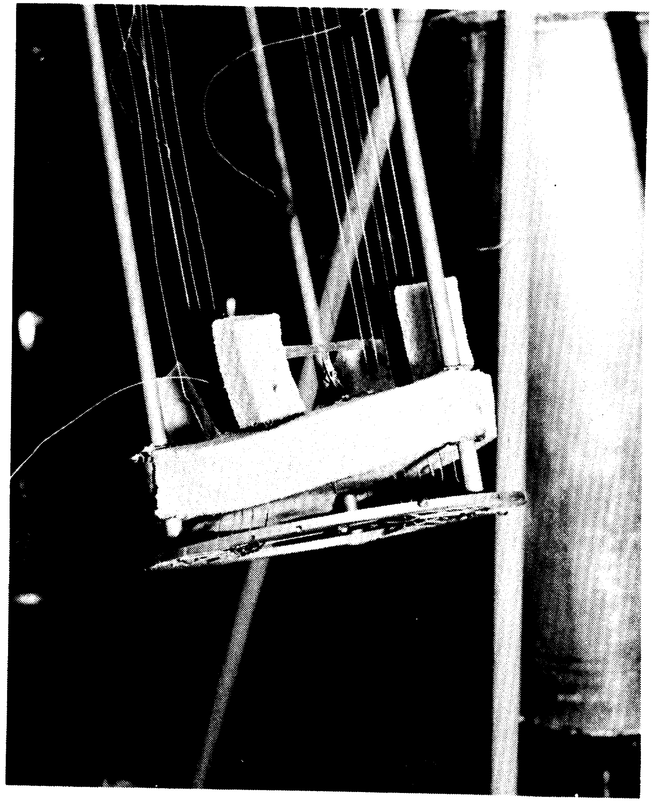
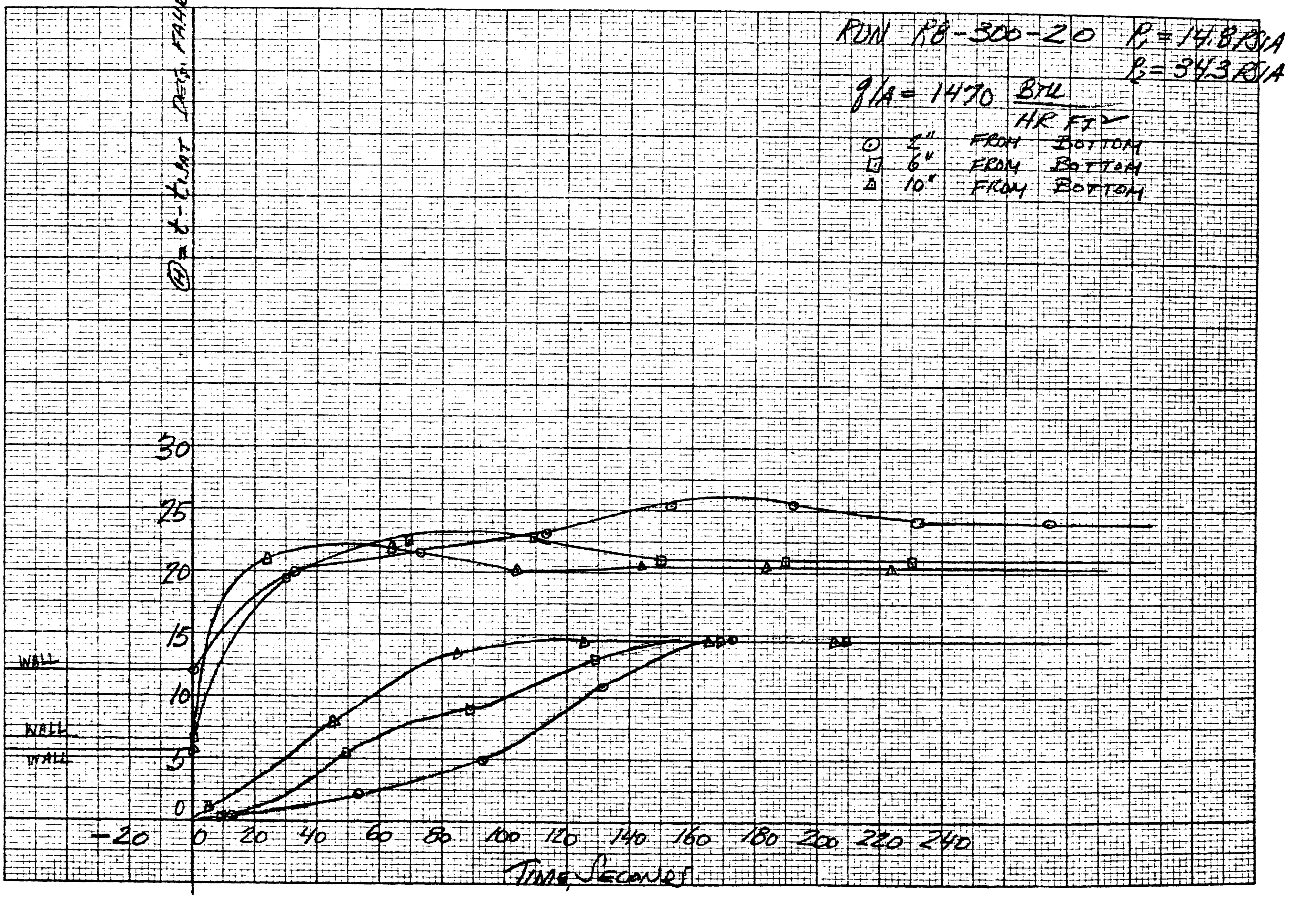


Fig. 7



RUN RB-300-20 P_1=14.8 PSIA
P_2=34.3 PSIA
q/A = 1470 BTU / HR FT²
⊙ 2" FROM BOTTOM
□ 6" FROM BOTTOM
△ 10" FROM BOTTOM
Ⓗ = t - t_SAT DEG. FAHR.
30
25
20
15
10
5
0
WALL
WALL
WALL
-20
0
20
40
60
80
100
120
140
160
180
200
220
240
TIME, SECONDS

FIG. 8

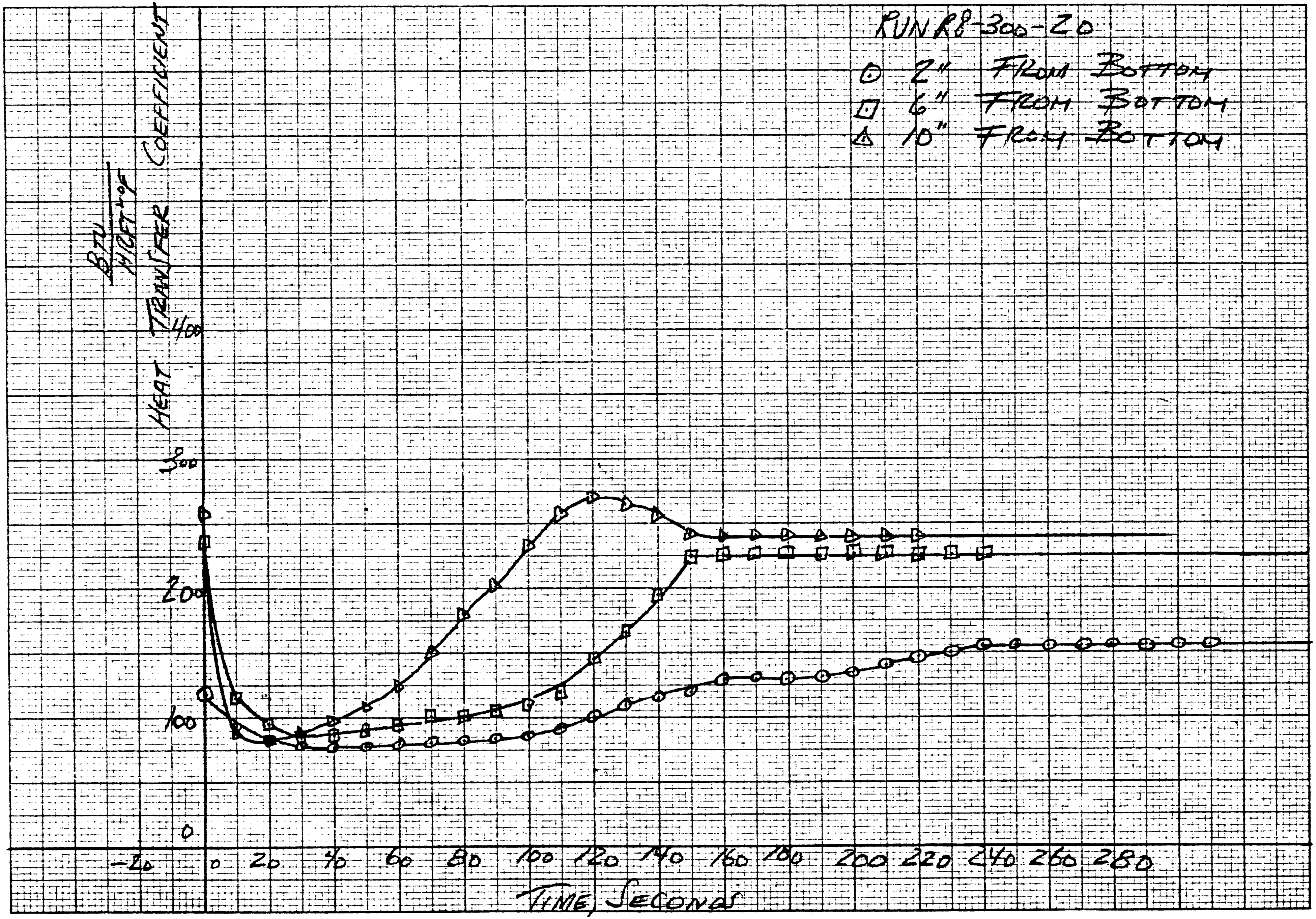
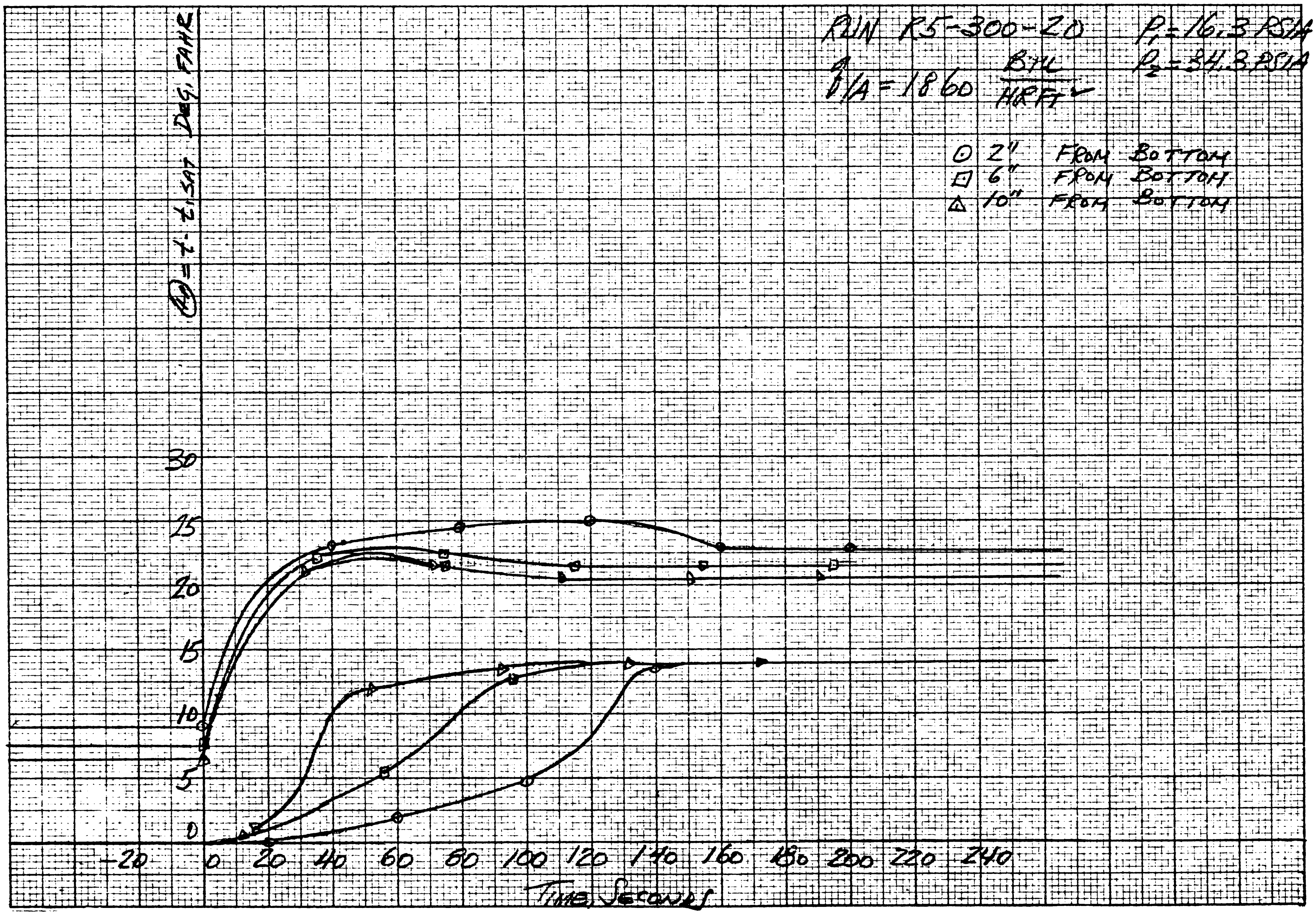

RUN R8-300-20
2" FROM BOTTOM
6" FROM BOTTOM
10" FROM BOTTOM
BTU/HR FT² °F
HEAT TRANSFER COEFFICIENT
400
300
200
100
0
-20
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
TIME, SECONDS

FIG. 9



RUN R5-300-20
P_1 = 16.3 PSIA
P_2 = 34.3 PSIA
q/A = 1860 BTU/HR FT²
⊙ 2" FROM BOTTOM
□ 6" FROM BOTTOM
△ 10" FROM BOTTOM
Δt = t - t SAT DEG. FAHR
30
25
20
15
10
5
0
-20
0
20
40
60
80
100
120
140
160
180
200
220
240
TIME, SECONDS

FIG. 10

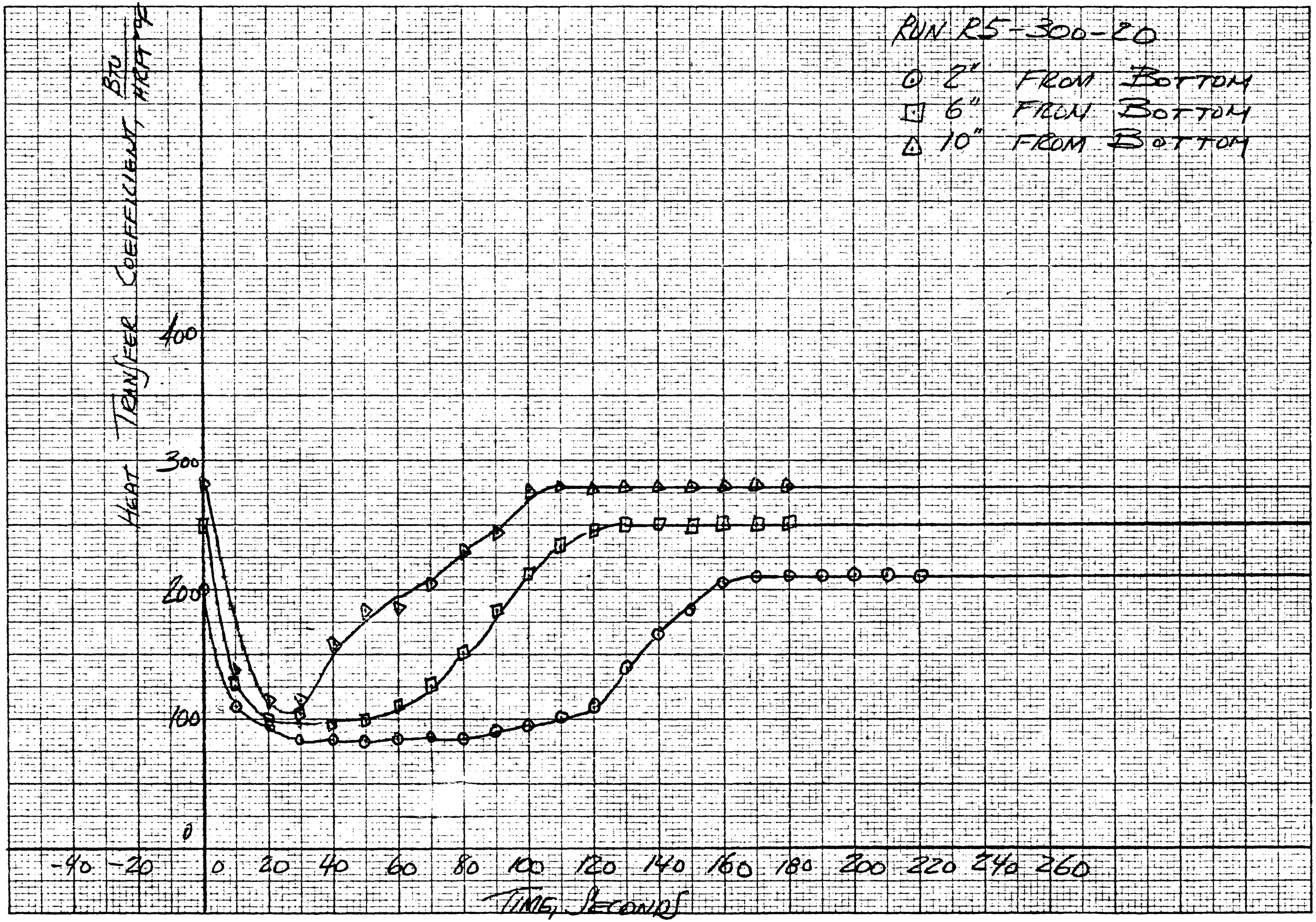

RUN R5-300-20
⊙ 2" FROM BOTTOM
⊡ 6" FROM BOTTOM
△ 10" FROM BOTTOM
HEAT TRANSFER COEFFICIENT, BTU/HR FT² °F
400
300
200
100
0
-40 -20 0 20 40 60 80 100 120 140 160 180 200 220 240 260
TIME, SECONDS

FIG. 11

PRESSURIZATION TO 35 PSIG

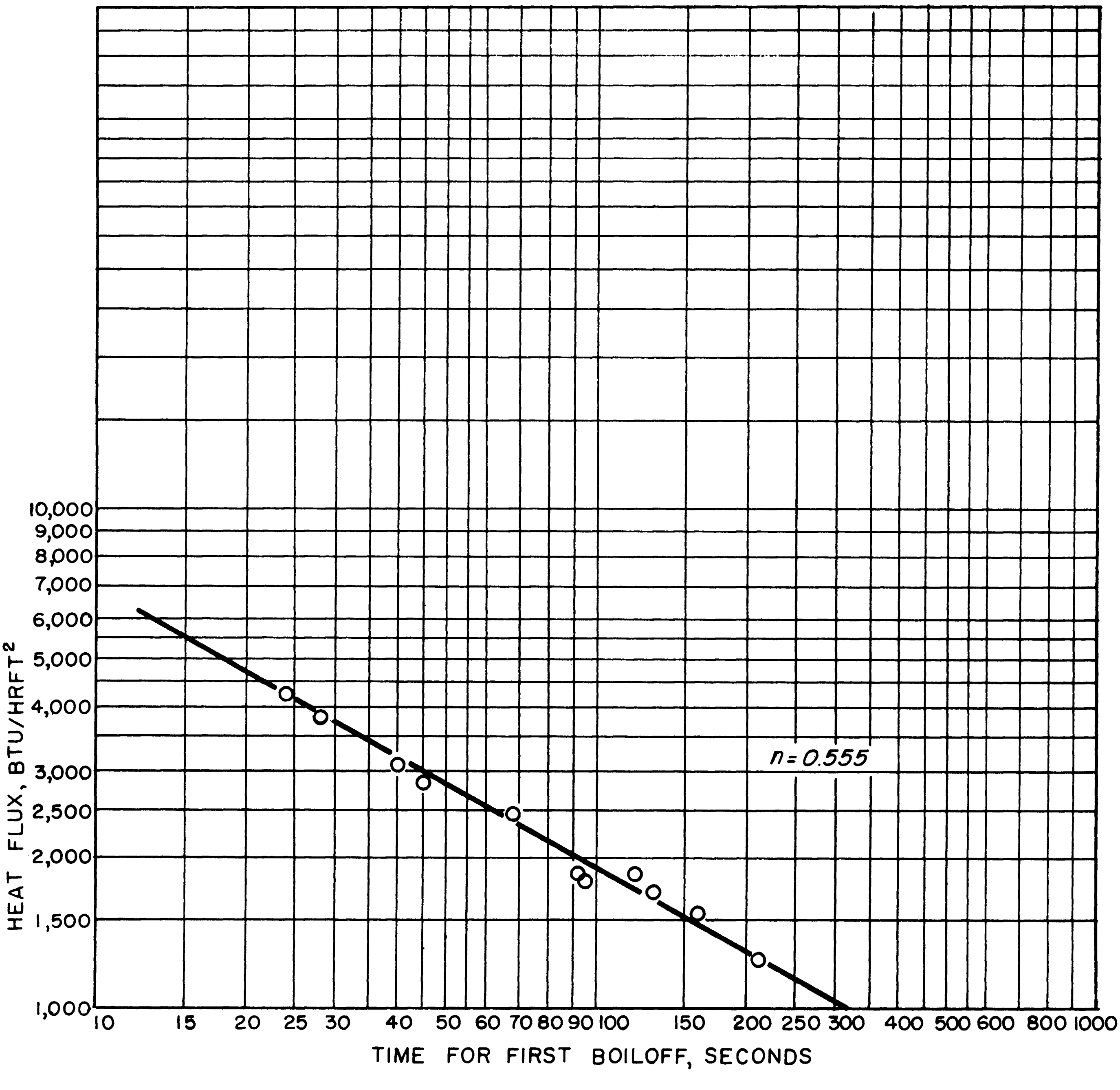


FIG. 12

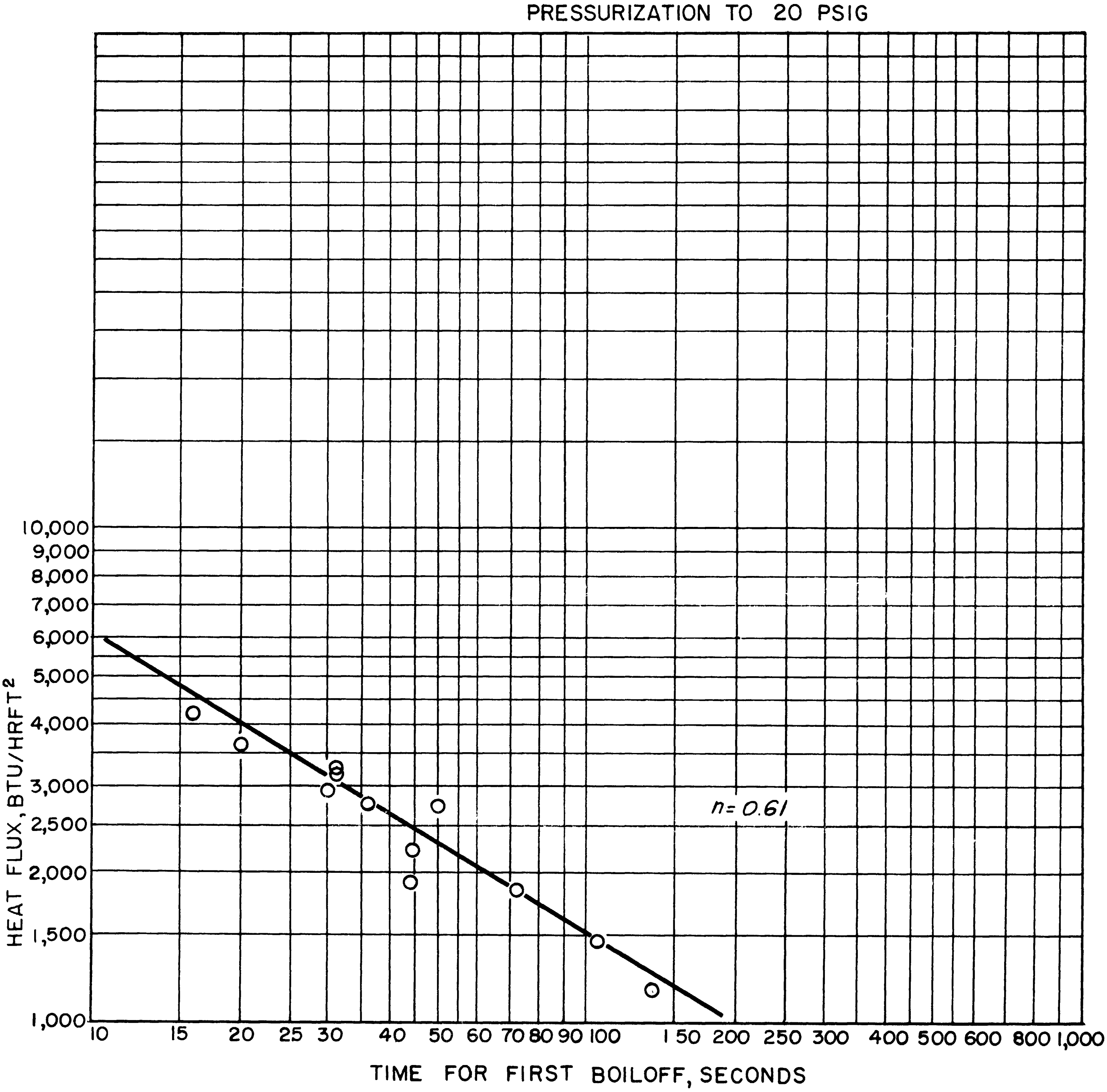

PRESSURIZATION TO 20 PSIG
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,500
2,000
1,500
1,000
HEAT FLUX, BTU/HRFT^2
n= 0.61
10 15 20 25 30 40 50 60 70 80 90 100 150 200 250 300 400 500 600 800 1,000
TIME FOR FIRST BOILOFF, SECONDS

FIG. 13