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PROGRESS REPORT

RELIABILITY OF THE TEMPERATURE-FLOW-CORROSION UNIT IN ANTIFREEZE STUDIES

BY

JAMES H. CONLEY

AND

CAPT. AUTHUR B. KREWINGHAUS

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AMCMS CODE NO. 5025.11.80300

DEPARTMENT OF THE ARMY PROJECT NUMBER

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ABSTRACT

The purpose of this investigation was to determine the reliability of the Temperature-Flow-Corrosion Unit (TFCU) in obtaining heat transfer data and corrosion data on antifreeze compounds.

Tests conducted in this study employed antifreeze meeting Federal Specification O-A-548a with added Federal Specification O-I-490a inhibitor. Quantitative metal ion concentration was determined by an atomic absorption spectrophotometer.

Oscillographic scanning of the wave forms produced by the electronic power unit under load indicate a fallacy in the volt meter and ammeter scale readings.

The results show that the use of the TFCU for heat transfer properties of antifreeze compounds is not feasible due to the fact that they do not necessarily correlate with results produced in an automotive cooling system.

The results also indicate that a loss in thermal conductivity resulting from corrosion product deposition on a metal surface can be overshadowed by the effect of increased surface roughness on fluid turbulence. Resultant heat transfer rates can increase as the corrosion products are deposited.

TABLE OF CONTENTS

I. INTRODUCTION

The U. S. Army Coating and Chemical Laboratory was directed by AMC Program Directive, AMCMS Code 5025.11.80300 dated 22 September 1967 to conduct research on automotive coolants.

In the past, evaluation of coolants involved three phases of qualification: glassware bench corrosion tests, simulated service tests, and vehicle field tests. The Temperature-Flow-Corrosion Unit was designed as a single unit which would rapidly produce results correlating with results received in the three phase testing presently practiced. If correlation is received the number of tests, which at present are essential to properly evaluate an automotive coolant, could be substantially reduced.

After installing the unit a program of study was initiated to determine the capabilities, mechanical variables, and test limits of the machine. The preliminary results were reported in CCL Report No. 205. This report covers the tests conducted in the interim.

II. DETAILS OF TEST

A. Apparatus - In its simplest configuration the heat transfer chamber of the TFCU consists of a vertical 10.6 inch long annulus. The outer wall is glass and has a diameter of 2.25 inches. In the center is a heat rejecting rod, 0.25 inch in diameter, made of a special alloy steel which exhibits corrosion properties similar to the cast iron used in an automotive engine block. The heat rejecting rod is electrically heated to maintain a constant temperature at a chosen spot along its length. The rod has a 1/8 inch diameter axial hole in which thermocouples are placed. The position of these thermocouples can be varied at will. The coolant is continuously recirculated at a controlled flow rate, inlet temperature, pressure, and is aerated at a constant rate.

B. Procedure - All metal specimens are cleaned and weighed to the nearest 0.1 mg. The specimens are mounted in the test chamber and the thermocouples inserted into the heat rejecting specimen at the top and bottom. Nine liters of coolant are used to fill the unit. The power is turned on, the instruments activated and the pump started. On this series of tests the controls were set so that the following conditions were recorded:

Solution Temperature	- 180°F (80°C)
System Pressure	- 7.6 lbs
Flow Rate	- 5 to 8 gallons/min.
Aeration	- 0.75 Standard cubic feet per hour.

After the solution temperature reaches 180°F the specimen heat is turned on and gradually increased until a maximum temperature of 257°F (125°C) is reached. The heat exchanger valve is partially opened so that the auxiliary heater is in operation approximately 30 seconds every minute. Thermocouple readings of solution temperature and heat rejecting specimen temperatures are recorded continously on an electronic recorder. Periodic readings of these temperatures are taken for calculation of heat data. Flow rates are varied from 5 gallon/minute to 8 gallon/minute and temperature profiles are recorded. Readings of the voltage are also taken periodically for the calculation of heat transfer data (see Table I). Periodic solution samples are also taken for analysis of metal ions. The test is continued for a total of 500 hours.

After the test period the chamber is opened, the strips are removed, scrubbed clean with a soft bristle brush and soap, rinsed with water, rinsed with acetone, and dried. After they reach equalibrium in a desiccator they are again weighed to the nearest 0.1 mg.

III. RESULTS OF TESTS

Results of the preliminary tests gave indications that the voltage and amperage readings taken from the A.C. meter may have been incorrect. An oscilloscope was used to check the wave forms. It was found that the curves were not true sine waves. Portions of the waves were flattened out due to the stepless silicon rectifier control device. The wave forms were photographed with a Polaroid camera and retraced onto graph paper. Areas under the curves were calculated and the true root mean square (rms) or effective voltage and amperage values obtained for calculation of the heat transfer data (see Graphs II and III).

Results in Graph I show curves of metal ion concentration in two runs with O-A-548a antifreeze and O-I-490a inhibitor. The graph shows that the metal ion concentration increased in the beginning of the run and then decreased to values even lower than the initial sample. Analyses were made utilizing the Perkin-Elmer Model 303 Atomic Absorption Spectrophotometer.

Results in Graphs IVa thru VIb and Tables II, III, and IV show that even though the temperature profiles for each test at any given time were as expected, the change of heat transfer coefficient (h) with time and corrosion is contrary to results received in automotive cooling systems and conflicts with the results obtained by the designers of the Temperature-Flow-Corrosion Unit on their own apparatus.

The heat transfer coefficients were calculated based on the assumption that the heat flux rate is the same for all points along the length of the rod. However, on the two specimens used boiling occurred at the top one third of the rod and progressed down until about one half of the rod exhibited boiling at the surface and the heat flux rates were most probably not the same.

Tables II, III, and IV show measurements made on identical liquid mixtures under similar conditions of heat rejecting specimens of differing degrees of surface rougnness. Sample S was polished with crocus cloth until it was shiny and smooth. The surface of sample R was sandblasted and was extremely rough. At the entrance where no boiling is occurring, with the thermocouples placed at distances D = .075 and D = .200 the ratio of $h_R/h_S = 1.47$. Thus the roughness increases h by 47%.

At the other end of the rods where nucleate boiling is occurring, under conditions of equal degree of superheat and equal positions on the rods the ratio of $h_R/h_S = 1.64$. Here the roughness increases h by 64%.

This roughness effect could also be caused on the heating rod by corrosion and a subsequent increase in boiling sites causing an apparent rise in heat transfer rate, with the insulating effect of the corrosion products being overshadowed by the roughness.

IV. DISCUSSION

Boiling is defined as the formation of the vapor phase within a liquid. The TFCU demonstrates three different regions or types of boiling. An examination of a typical boiling curve, Figure 1, clearly shows the locations of the regions and transition points where the type of boiling changes. (see ref. 7).

In the region, AB, heat is transferred from the heater to the liquid by natural convection only. At B however, bubbles first start to appear in the liquid immediately adjacent to the heater surface, and the slope of the boiling curve sharply increases. The overall heat transfer coefficient in boiling is defined by:

$$h = {}^{Q}/_{A}\ (T_w - T_{sat.})$$

Where T_w is the heater surface temperature and T_{sat} is the saturation temperature of the liquid. As soon as the bubbles form, the heat transfer rate begins to increase rapidly with no change in the wall or fluid temperatures, thereby resulting in similar increases in h. There are two possible explanations as to what happens to cause this increase. As the bubbles form and leave the surface cooler liquid continuously replaces them causing better liquid circulation and thus an increase in h; or as the bubbles break cooler liquid is drawn into the areas where the bubbles were causing the increase in h.

In the region, BC, bubbles form continuously at the heat surface at increasing frequency and over a larger area. The two-phase boundary layer adjacent to the heating surface is stable. This region is called the <u>nucleate boiling region</u>. In this region the surface heat transfer coefficient increases continously with heater temperature.

At point C, the two-phase boundary layer becomes unstable due to large relative velocities between the bubbles, or columns of vapor rising from the surface, and the columns of liquid approaching the surface. This phase begins at point C and continues to point D where a stable vapor blanket covers the whole surface. The region, CD, is called the <u>transition boiling region</u>. In this region an increasing percentage of the heater surface is insulated from the liquid and the surface heat transfer coefficient drops rapidly with increasing heater temperature. At point D, the surface is completely insulated, and boiling takes place at the gaseous/liquid interface.

In the region, DE, the stable vapor blanket persists and the solid surface heat transfer coefficient increases only very slowly from its minimum value at D, due to slightly higher convection rates inside the vapor blanket and an increasing radiation component. This type of boiling is called <u>film boiling</u>.

In most engineering applications, the boiling curve will follow the broken line CC', shown in Figure 1. For boiling water under normal conditions, the value of the heat flux (Q/A) at point C is approximately 1×10^6 BTU/ft^2 hr. and the surface superheat is 100°C. If the heat flux is increased beyond this value, the boiling curve follows the broken line to point C' where the surface superheat under film boiling conditions may be 5000°F and the heating surface may burn out. For this reason, C is referred to as the <u>burn out point</u>. In actual boiling processes, a value of 50% of the burn out flux may be considered to be the maximum safe operating flux.

Therefore, the area bounded by the incipient boiling point, B, and the burn out point, C, is the area where very high and very stable heat transfer coefficients are obtained.

When heating or cooling a fluid by a forced convection such as we find in an automotive engine and part of the TFCU, it has been found by dimensional analysis and experiment that the following relationships hold (see reference 4):

$$N_{Nu} = C(N_{Re})^m (N_{Pr})^n \qquad (1)$$

where C, m and n are experimentally determined empirical constants and N_{Nu}, N_{Re}, and N_{Pr} are dimensionless numbers known as Nusselt's, Reynold's, and Prandtl's numbers respectively. These numbers are obtained by the following ratios:

$$N_{Nu} = \frac{hD}{K}, \quad N_{Re} = \frac{VD\rho}{u}, \text{ and } \quad N_{Pr} = \frac{u\,C_p}{K} \qquad (2)$$

where h is the coefficient of heat transfer, D is a diameter or length characteristics of the flow cross section, k is the thermal conductivity of the fluid, V is the velocity of flow of the fluid, ρ is the density of the fluid, u is the dynamic viscosity at the mean temperature, and C_p is the specific heat of the fluid at constant temperature. By dimensional analysis for forced convection we obtain,

$$\frac{hD}{K} = C \frac{VD_p}{u}^{m} \quad \frac{uC_p}{k}^{n} \qquad (3)$$

When the Reynold's number exceeds 2300, turbulent flow is obtained. In engine cooling systems this situtation prevails in both the heating phase and the cooling phase. This also exists in the TFCU, and N_{Re} of about 7500 are prevalent at the entrance region. It has been found experimentally that the value of C is 0.023, and for m it is 0.8. The value for n is 0.4 for heating and 0.3 for cooling. Substituting these constants in equation (3) we obtain,

$$h = 0.023 \frac{(V_p)^{0.8} \; k^{0.6} \; C_p^{0.4}}{u^{0.4} \; D^{0.2}}$$

The above correlations applies only for force convection and does not hold for the heat transfer mechanism occurring on the top of the rod, namely nucleate boiling.

Various correlations have been proposed for nucleate boiling heat transfer coefficients, among them, those of Rohsenow (8) and Forster (9). These models are quite complicated and will not be included here for sake of brevity. However, it should be noted that many other additional properties are relevant in nucleate boiling aside from those incorporated into the conventional Reynold's and Prandtl number. For example, the surface tension, latent heat of vaporization, liquid and vapor densities, saturation temperature, and other properties of both liquid and vapor phase must be introduced to adequately describe the heat transfer occurring.

The heat transfer coefficient for a liquid flowing past a surface is often a function of the surface roughness. This roughness can be viewed as protrusions above the surface or as cavities in the surface. In cases where the liquid is not boiling, the increase in h due to roughness is the result of the local turbulence that the protrusions cause. Previous investigators have found that the heat transfer coefficient increased in the early stages of deposition and then decreased as expected (see ref. 10). This effect is expected to be a function of the distance that the surface roughness protrudes above the surface. Initially the heat transfer coefficient tends to rise because the deposit roughness disturbs the laminar sublayer of the fluid, causing turbulence near the tube wall. This effect opposes any decrease in h due to the low thermal conductivity of the deposit. When the thickness of the deposit approaches and exceeds that of the laminar sublayer, however, the effect of the low thermal conductivity becomes predominant and the heat transfer coefficient will decrease.

In the case of a liquid undergoing nucleate boiling, the roughness plays an additional role. At any given degree of superheating, the amount of boiling that takes place is dependent on the number of nucleation sites and the radius of curvature of these sites. An increase in roughness starting from a smooth surface probably results in a larger number of sites and a smaller radius of curvature for these sites. Both of these factors would tend to increase the amount of nucleate boiling and thus increase the heat transfer coefficient.

As to whether or not the heat transfer coefficients did actually increase with corrosion deposits can not be unequivocably confirmed at this time. As mentioned earlier, all calculations were based on the assumption that the heat transfer rates were equal at all points along the rod. This is a questionable assumption since two types of heat transfer mechanisms were observed on the heating element. Thus, as deposits accumulated on the rod and boiling progressed down the length of the rod, the effect of the higher rates due to nucleate boiling could mask a drop in the forced convection transfer rates at the rod's bottom. Thus, further analysis should be made on existing data to clarify this situation.

V. CONCLUSIONS

A. The temperature profiles curves at any given time, set point temperature, and flow rate on either a smooth or rough rod follows the expected pattern.

B. The chief cause for concern in the use of the TFCU for heat transfer data of automotive coolants lies in the fact that the data does not correlate with results produced in automotive cooling systems.

C. Corrosion data, as evidenced by the analytical determination of metal ion concentration in the coolant after test, does not correlate with results received in glassware or simulated service tests.

VI. RECOMMENDATIONS

It is recommended that further analysis of the data be performed.

VII. REFERENCES

1. AMC Program Directive AMCMS Code 5025.11.80300 dated 22 September 1967.

2. Federal Specification O-A-548a, Antifreeze, Ethylene Glycol, Inhibited, Type I, dated 30 Dec 1958.

3. Federal Specification O-I-490a, Inhibitor, Corrosion, Liquid Cooling System, dated 26 April 1965.

4. Max Jacob and G. A. Hawking, Elements of Heat Transfer and Insulation, 1950, John Wiley and Sons, New York.

5. CCL Report No. 90, Heat Transfer Efficiency of Automotive Engine Coolants, 17 March 1960.

6. CCL Report No. 205, Preliminary Investigation of the Temperature-Flow-Corrosion Unit as a Tool for Coolant Evaluation, June 1966.

7. Royal Aircraft Establishment Technical Note No. Mech. Eng. 391, Comments of Nucleate Boiling Heat Transfer, Correlations and Suggestions for Evaluating the Number of Boiling Sites, October 1963.

8. Rohsenow, W. M. Trans. A.S.M.E. 74, 969 (1952).

9. Foster, H.K. and Greif, R.J., Heat Transfer 81, 43 (1959).

10. Smith, J.D., The Effects of Deposits on Heat Transfer to Aviation Gasoline, to be published in I&EC Product Research and Development.

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APPENDIX A

TABLE I

MATHEMATICAL RELATIONSHIPS USED TO CALCULATE HEAT DATA

$Q' = \text{Volts}_{\text{rms}}\ \text{Amperes}_{\text{rms}}$ = Power in Watts

A = Surface Area in sq. feet = $0.06545\ \text{ft}^2$

3.41304 = Conversion factor to convert to BTU

T_2 = Temperature drop at Metal Surface = $0.008191\ Q'$

$T_s = T_1 - T_2$ = Temp. reading at a given point - Temp. drop in °F

$Q = 3.41304\ \frac{V \times I}{A}$ = Heat Flux Rate in BTU/ft^2 hr.

$h = 3.41304\ \frac{V \times I}{A\ (T_s - T_B)} = \frac{\text{Heat Flux Rate}}{T_s - \text{Liquid Bulk Temp}}$ = heat transfer coefficient in BTU/ft^2 hr. °F

TABLE II

TYPICAL HEAT DATA FOR A ROUGH ROD

T.C. #5 inches from Top	T.C. #3 inches from Bottom	T.C. #1 Soln Temp at Bot. °F	T.C. #2 Soln Temp at Top °F	T.C. #3 Spec. Temp Bot. °F	T.C. #5 Spec. Temp Top °F	Volts	Amps X200
5	4	180	181	216	257	1.85	2.38
5	5	180	181	226	257	1.85	2.38
5	6	180	181	237	256	1.85	2.35
5	7	180	181	245	256	1.85	2.35
5	8	180	181	250	257	1.85	2.38
5	9	180	181	252	257	1.85	2.38
5	10	180	181	254	257	1.85	2.38
4	11	180	181	255	257	1.85	2.38

TYPICAL HEAT DATA FOR A SMOOTH ROD

T.C. #5 inches from Top	T.C. #3 inches from Bottom	T.C. #1 Soln Temp at Bot. °F	T.C. #2 Soln Temp at Top °F	T.C. #3 Spec. Temp Bot. °F	T.C. #5 Spec. Temp Top °F	Volts	Amps X200
5	4	179	181	213	258	1.73	2.25
5	5	179	181	221	257	1.72	2.22
5	6	179	181	231	257	1.72	2.22
5	7	179	181	239	257	1.72	2.22
5	8	179	181	247	257	1.72	2.22
5	9	179	181	252	258	1.73	2.24
5	10	179	181	255	258	1.73	2.24
4	11	179	181	256	258	1.73	2.24

TABLE III

VARIATION OF POWER WITH TIME AND FLOW RATE

Smooth Specimen

Time (hrs)	G.P.M. Flow Rate	V_{rms}	$Amps_{rms}$	(Q') Power (watts)
23	5	1.17	298	348
72	5	1.20	306	367
169	5	1.27	322	409
215	5	1.36	354	481
238	5	1.43	370	529
24	8	1.40	372	521
73	8	1.37	350	480
170	8	1.50	396	594
216	8	1.57	420	659
238	8	1.70	456	773

Power increase at 5 GPM - 348 to 529 watts.
Power increase at 8 GPM - 521 to 773 watts.

VARIATION OF H WITH TIME AND FLOW RATE

Smooth Specimen

	h. (high temp end)	h. (low temp end)	T_2	Flow G.P.M.	Q
23	248 BTU/ft^2 hr°F	582 BTU/ft^2 hr°F	2.8	5	18,148.3
72	262 BTU/ft^2 hr°F	580 BTU/ft^2 hr°F	3.0	5	19,138.1
169	294 BTU/ft^2 hr°F	525 BTU/ft^2 hr°F	3.4	5	21,328.2
215	348 BTU/ft^2 hr°F	658 BTU/ft^2 hr°F	3.9	5	25,082.6
24	366 BTU/ft^2 hr°F	913 BTU/ft^2 hr°F	4.3	8	26,228.7
73	362 BTU/ft^2 hr°F	832 BTU/ft^2 hr°F	3.9	8	25,030.7
170	430 BTU/ft^2 hr°F	858 BTU/ft^2 hr°F	4.9	8	30,975.4
216	494 BTU/ft^2 hr°F	913 BTU/ft^2 hr°F	5.4	8	34,365.1

TABLE IV

VARIATION OF POWER WITH TIME AND FLOW RATE

Rough Specimen

Time (hrs)	G.P.M. Flow Rate	V_{rms}	$Amps_{rms}$	(Q')Power (watts)
170	5	1.50	490	735
214	5	1.50	494	741
263	5	1.67	436	728
335	5	1.75	460	805
431	5	1.82	476	866
506	5	1.90	514	977
217	8	1.95	544	1061
265	8	2.00	560	1120
337	8	2.02	576	1164
432	8	2.10	602	1264

Power increase at 5 GPM - 735 to 977 watts.
Power increase at 8 GPM - 1061 to 1264 watts.

VARIATION OF H WITH TIME AND FLOW RATE

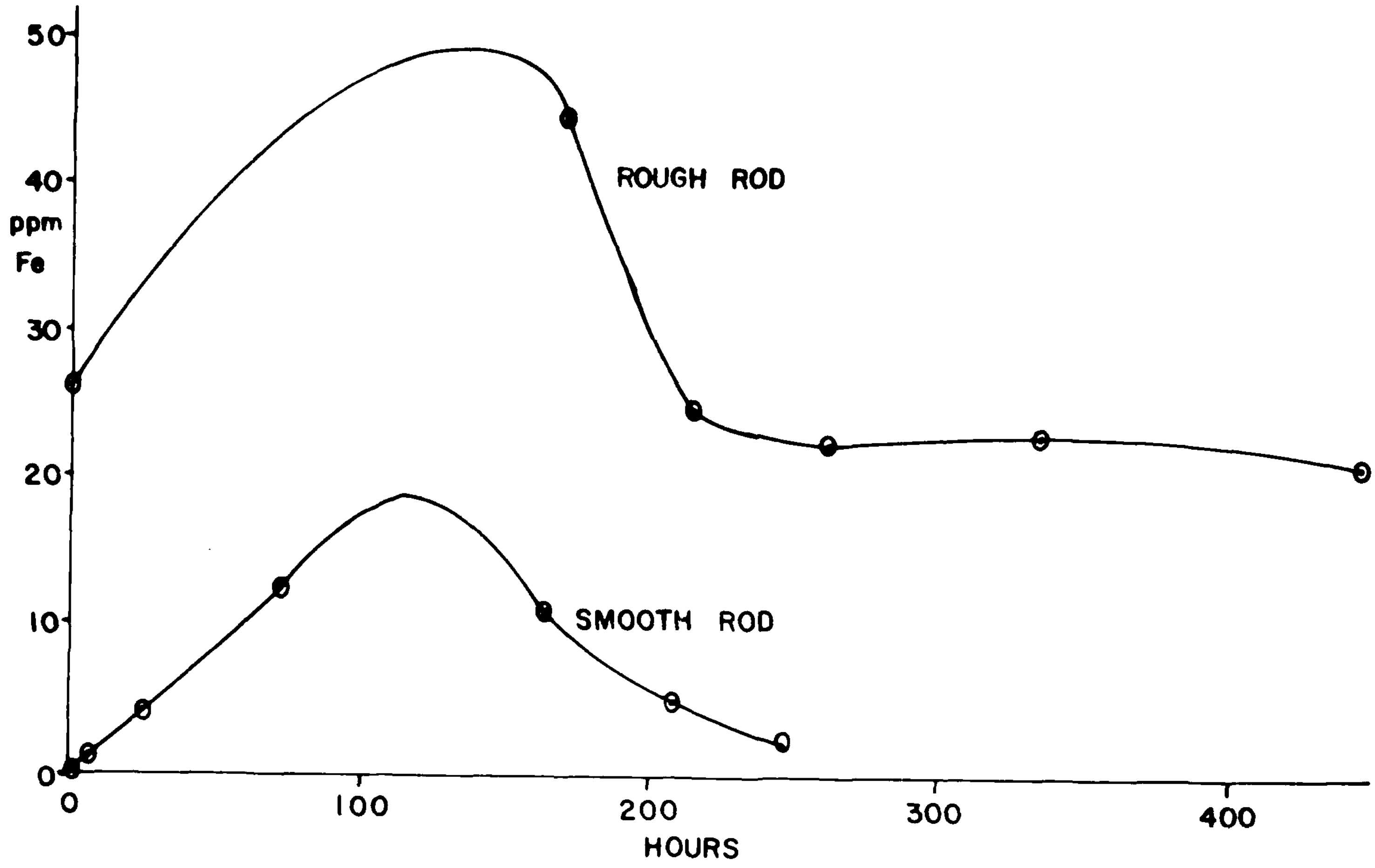
Rough Specimen

Time	h.(high temp end)	h.(low temp end)	T_2	Flow G.P.M.	Heat Flux Q
170	555 BTU/ft^2 hr°F	935 BTU/ft^2 hr°F	6	5	38,328
214	553 BTU/ft^2 hr°F	968 BTU/ft^2 hr°F	6.1	5	38,641
263	542 BTU/ft^2 hr°F	791 BTU/ft^2 hr°F	6.0	5	37,953
335	582 BTU/ft^2 hr°F	830 BTU/ft^2 hr°F	6.6	5	40,963
431	655 BTU/ft^2 hr°F	923 BTU/ft^2 hr°F	7.1	5	45,160
506	728 BTU/ft^2 hr°F	999 BTU/ft^2 hr°F	8.0	5	50,948
217	903 BTU/ft^2 hr°F	1278 BTU/ft^2 hr°F	8.7	8	55,328
265	874 BTU/ft^2 hr°F	1304 BTU/ft^2 hr°F	9.2	8	58,405
337	899 BTU/ft^2 hr°F	1303 BTU/ft^2 hr°F	9.5	8	60,699
432	961 BTU/ft^2 hr°F	1478 BTU/ft^2 hr°F	10.4	8	65,914

APPENDIX B

GRAPH I

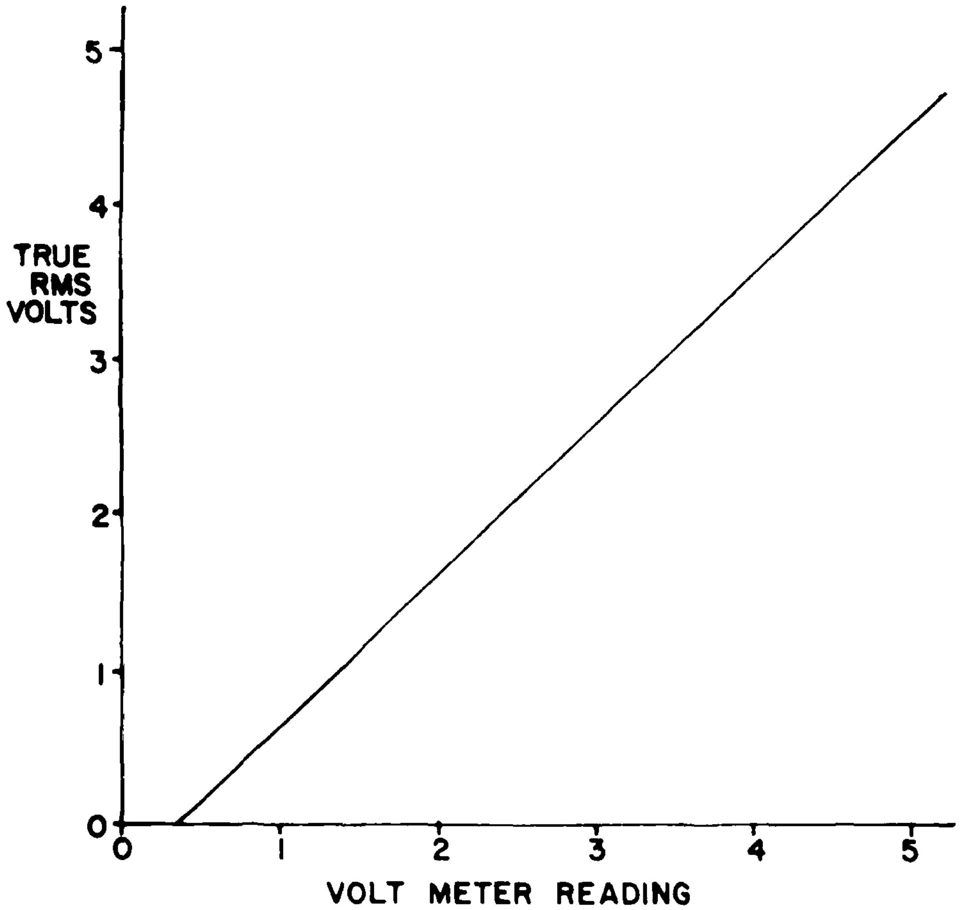
VARIATION OF IRON CONCENTRATION WITH TIME

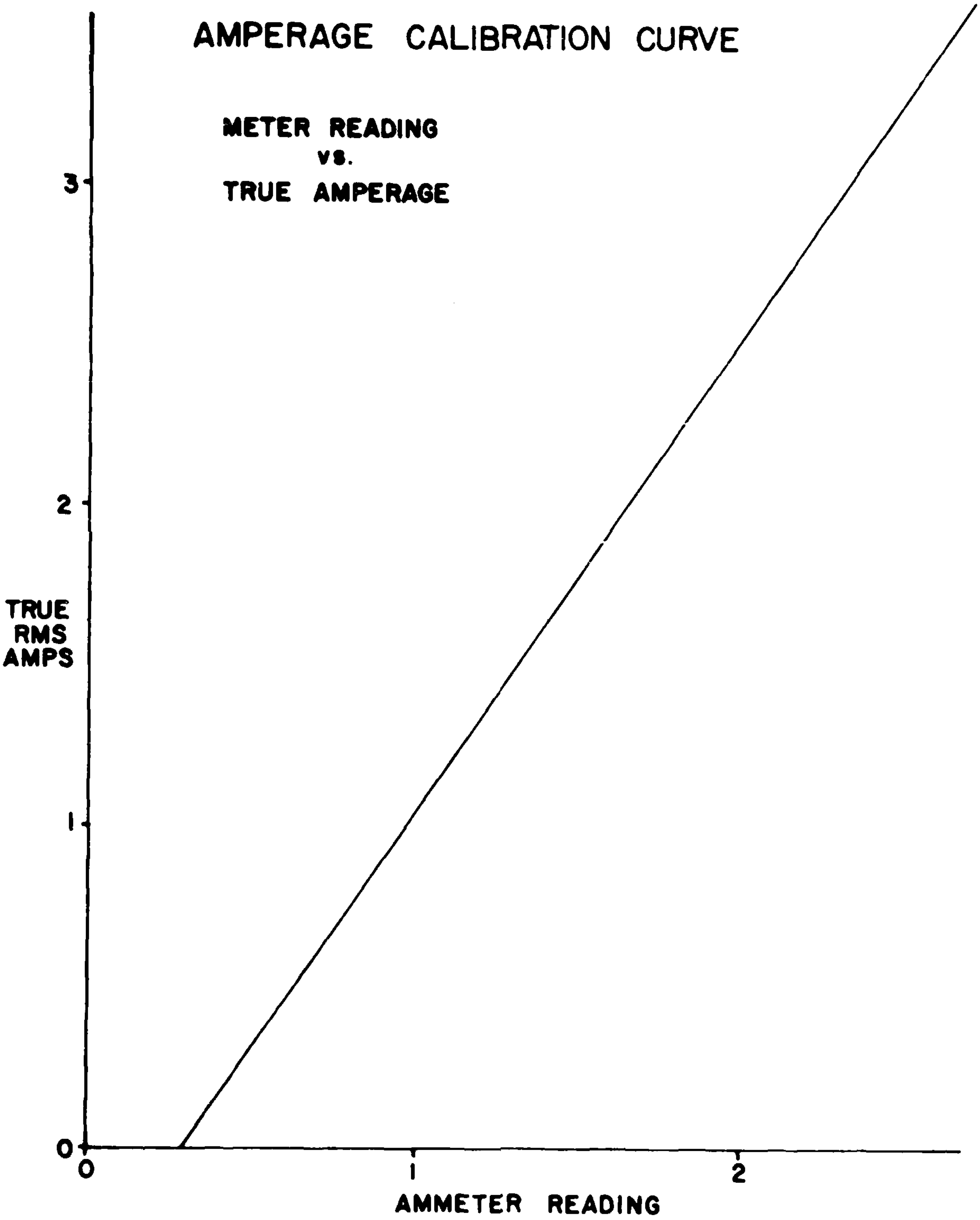


GRAPH II

VOLTAGE CALIBRATION CURVE

METER READING
VS.
TRUE VOLTAGE



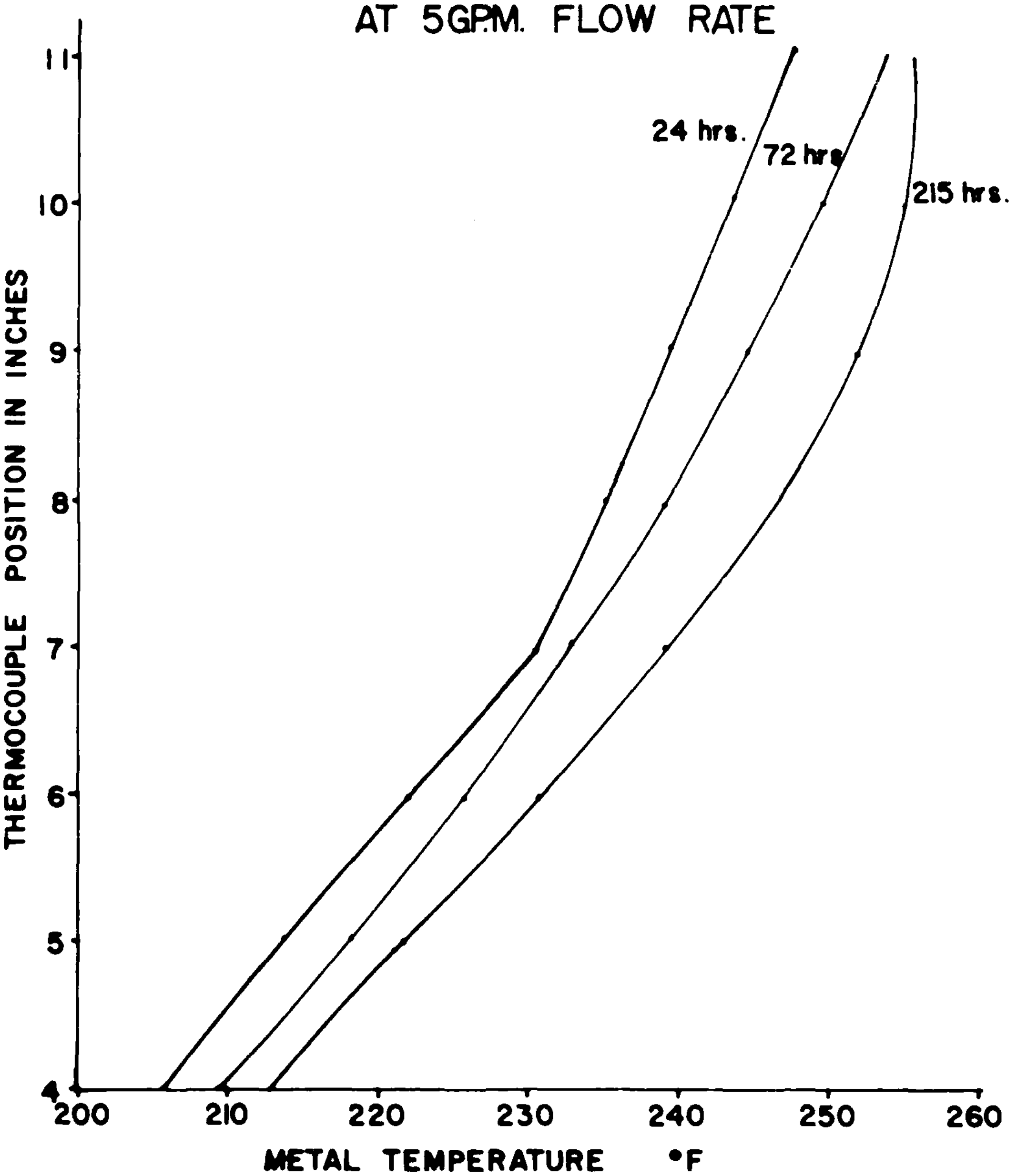


GRAPH III

AMPERAGE CALIBRATION CURVE

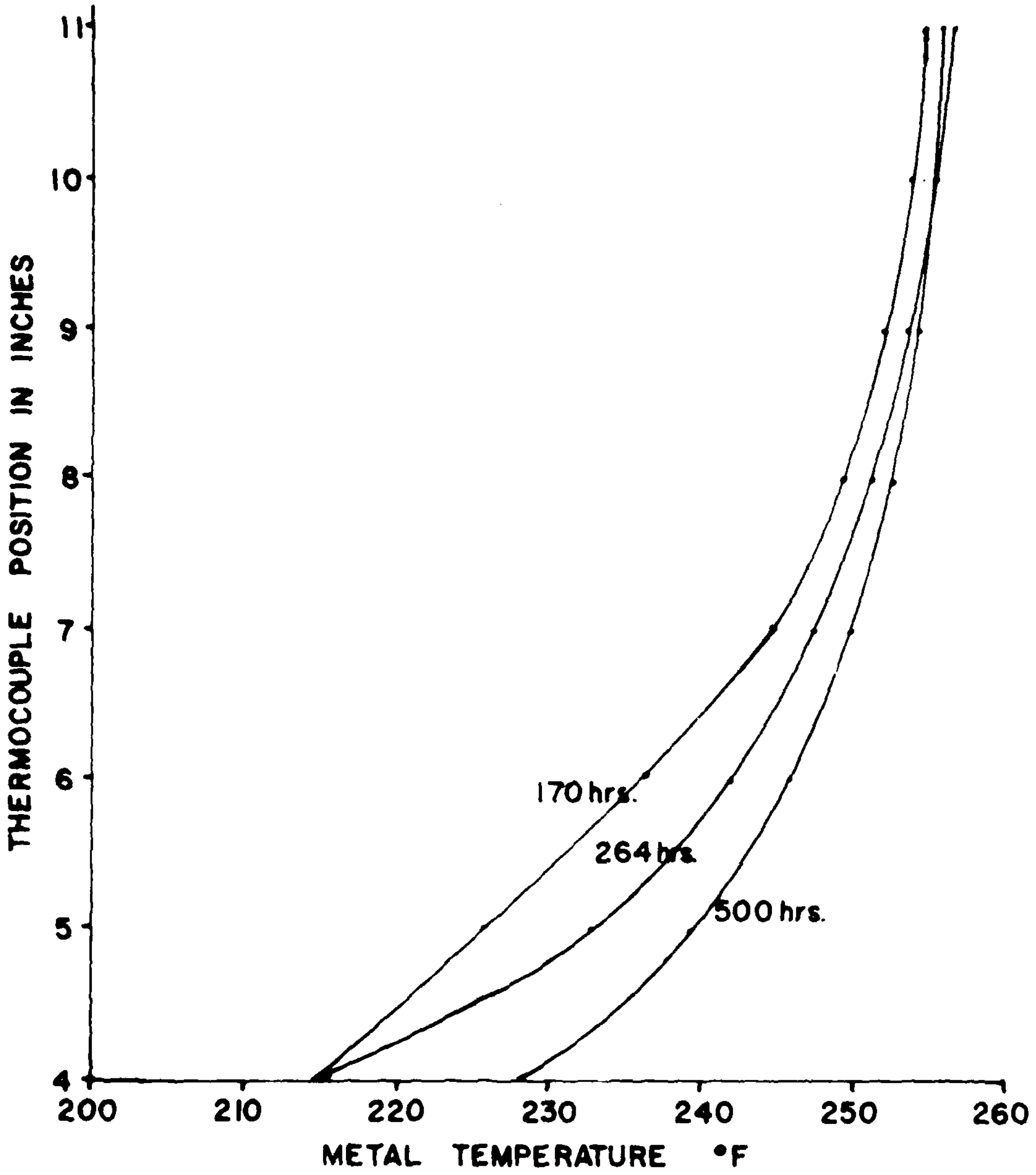
GRAPH IV a

SMOOTH SPECIMEN TEMPERATURE PROFILES AT 5GPM. FLOW RATE



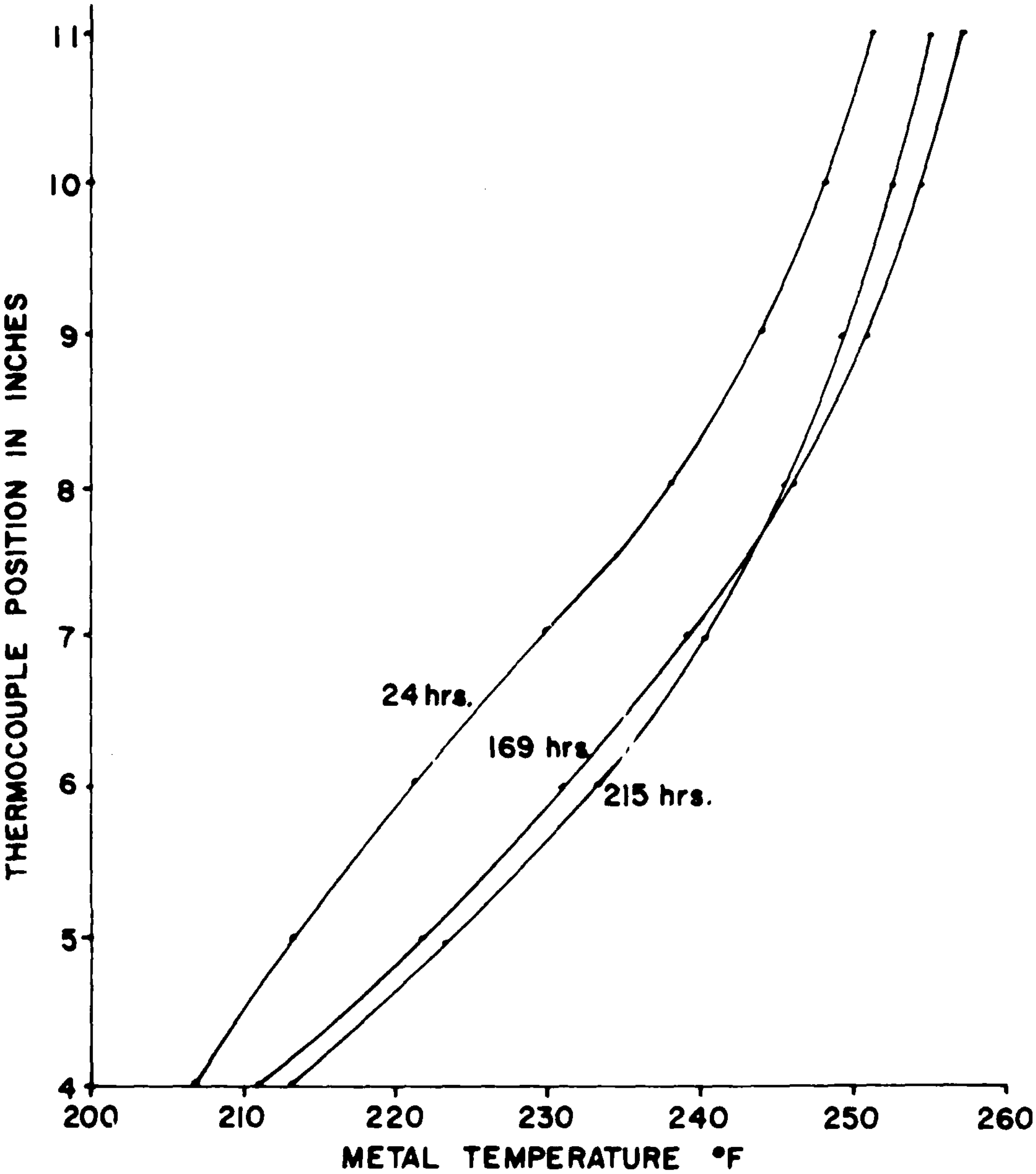
GRAPH IV b

ROUGH SPECIMEN TEMPERATURE PROFILES AT 5 G.P.M. FLOW RATE



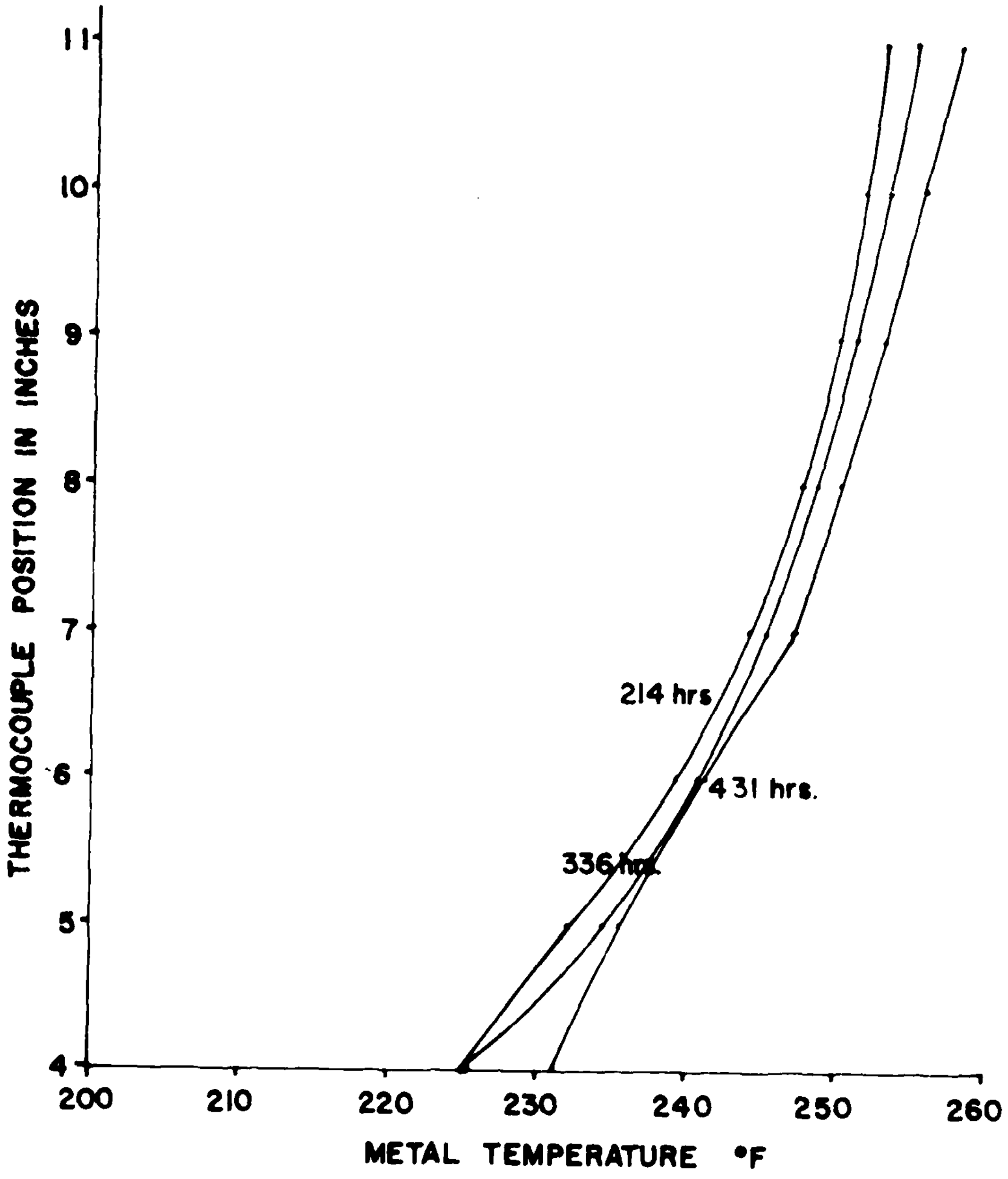
GRAPH Ⅴa

SMOOTH SPECIMEN TEMPERATURE PROFILES AT 8 G.P.M. FLOW RATE



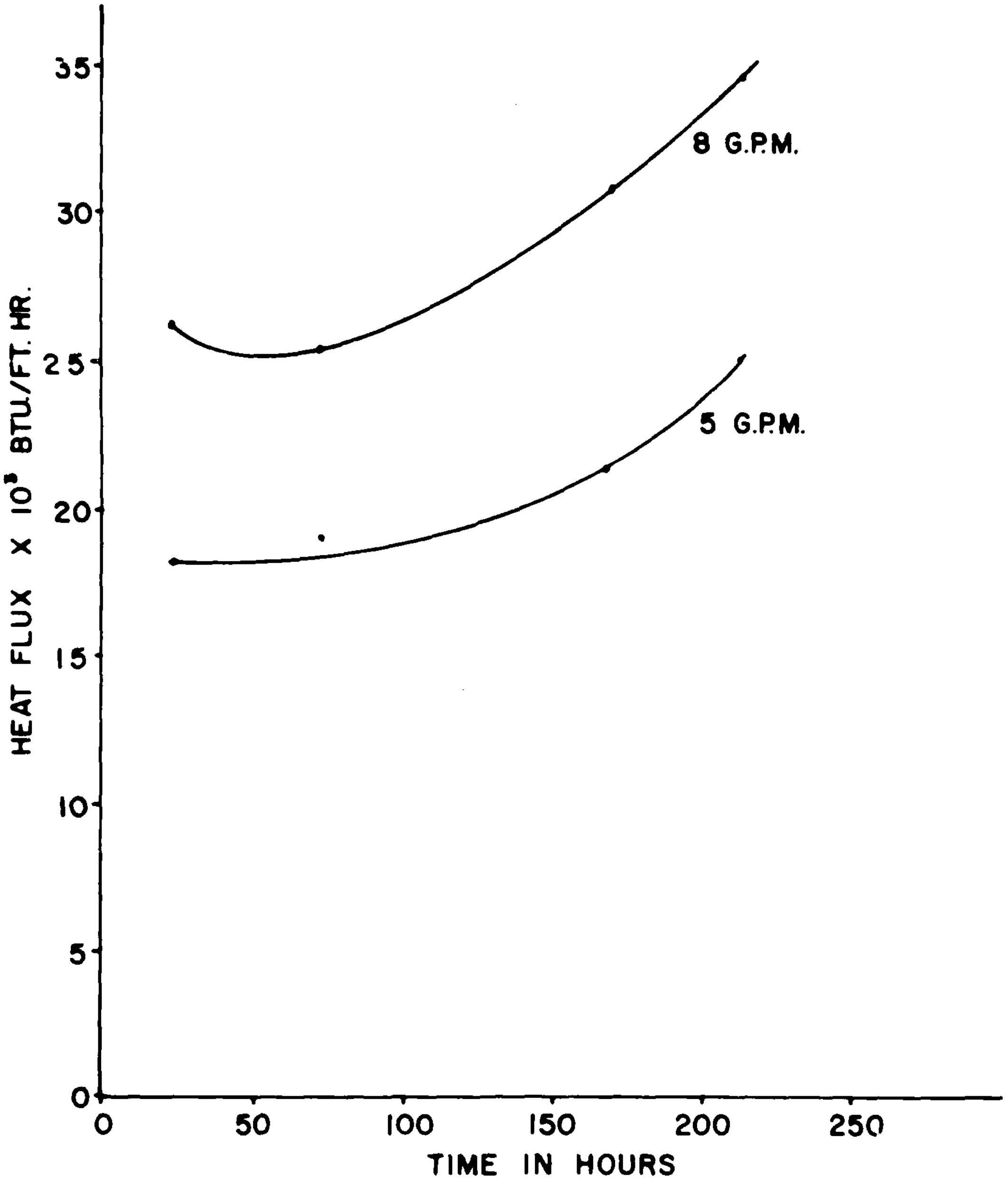
GRAPH V b

ROUGH SPECIMEN TEMPERATURE PROFILES AT 8 G.P.M. FLOW RATE



GRAPH VI a

VARIATION OF HEAT FLUX RATE WITH TIME
SMOOTH SPECIMEN



GRAPH VI b

VARIATION OF HEAT FLUX RATE WITH TIME
ROUGH SPECIMEN

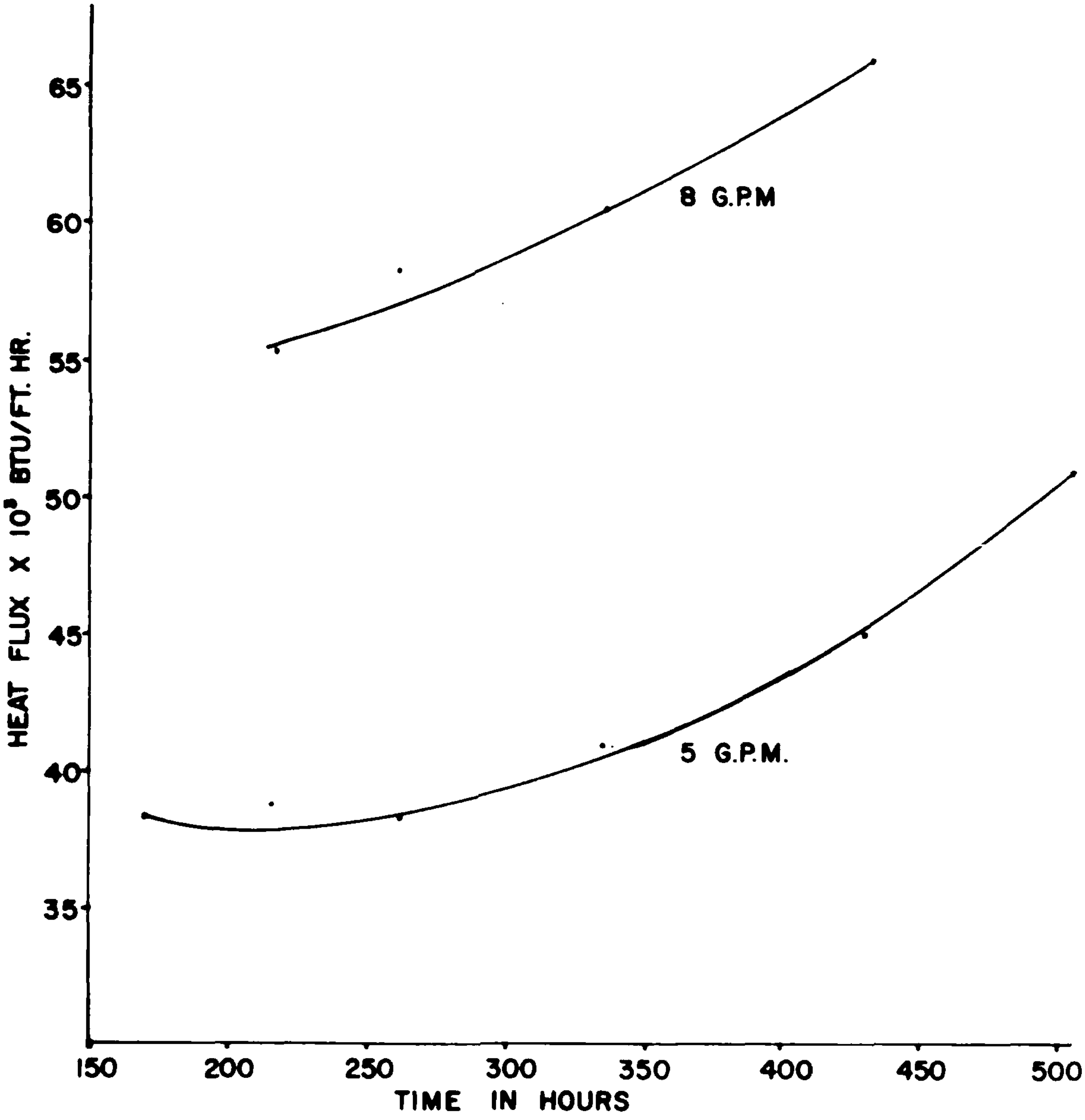
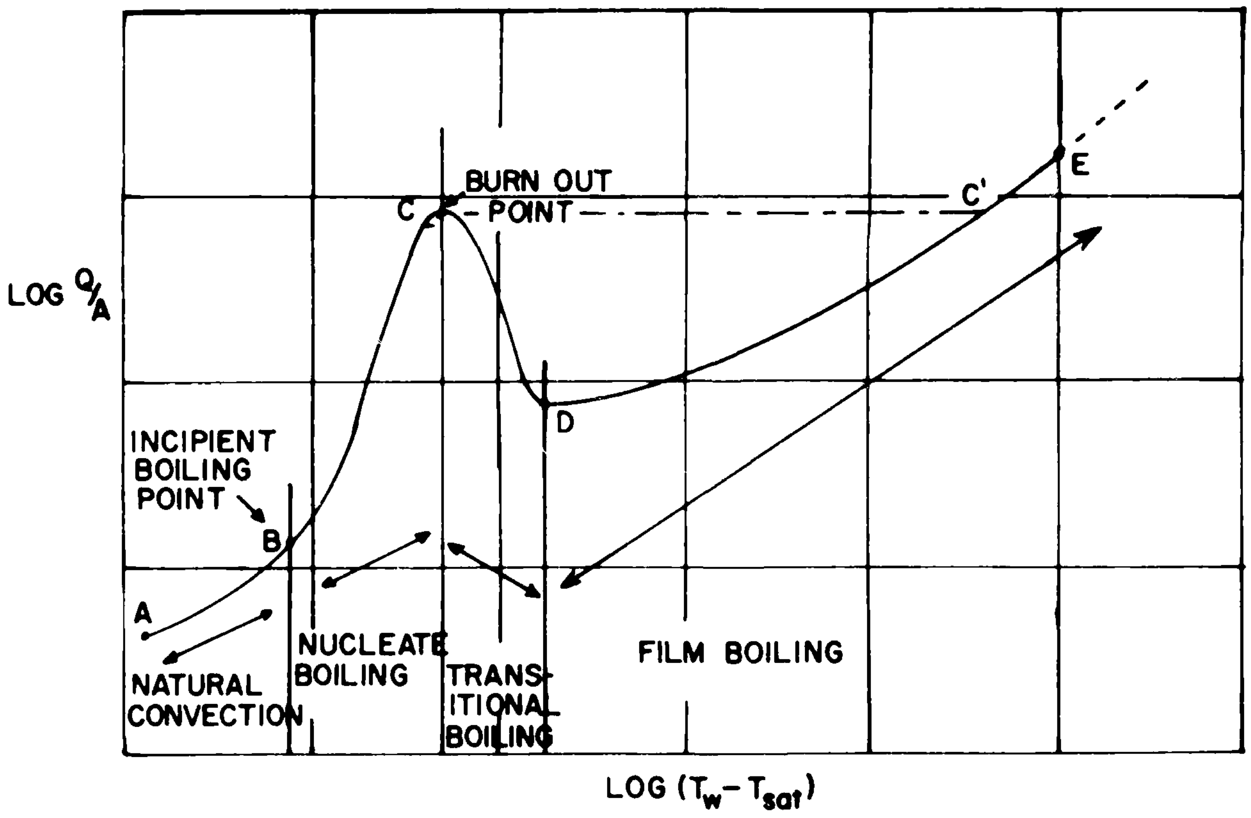


FIG. 1

TYPICAL BOILING CURVE



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RELIABILITY OF THE TEMPERATURE-FLOW-CORROSION UNIT IN ANTIFREEZE STUDIES

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12 SPONSORING MILITARY ACTIVITY

U. S. Army Materiel Command
Washington, D. C.

13 ABSTRACT

The purpose of this investigation was to determine the reliability of the Temperature-Flow-Corrosion Unit (TFCU) in obtaining heat transfer data and corrosion data on antifreeze compounds.

Tests conducted in this study employed antifreeze meeting Federal Specification O-A-548a with added Federal Specification O-I-490a inhibitor. Quantitative metal ion concentration was determined by an atomic absorption spectrophotometer.

Oscillographic scanning of the wave forms produced by the electronic power unit under load indicate a fallacy in the volt meter and ammeter scale readings.

The results show that the use of the TFCU for heat transfer properties of antifreeze compounds is not feasible due to the fact that they do not necessarily correlate with results produced in an automotive cooling system.

The results also indicate that a loss in thermal conductivity resulting from corrosion product deposition on a metal surface can be overshadowed by the effect of increased surface roughness on fluid turbulence. Resultant heat transfer rates can increase as the corrosion products are deposited.

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14 KEY WORDS	LINK A ROLE	LINK A WT	LINK B ROLE	LINK B WT	LINK C ROLE	LINK C WT
Temperature-Flow-Corrosion Unit						
Heat Transfer						
Antifreeze						
Surface Roughness Effect						
Turbulence						
Heat Flux						

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