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THE USE OF WATER COCLING FOR PROTECTION AGAINST THERMAL RADIATION FROM A NUCLEAR WEAPON DETONATION

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An experimental study was completed to determine the effectiveness of water cooling plates which are being exposed to the thermal radiation pulse of a nuclear weapon detonation. Heat transfer rates were measured on heated plates on which water was either sprayed or allowed to flow downward in a thin sheet. The plates in the experiments where cooling water flows over the plate were

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20. ABSTRACT (Cont.)

simultaneously heated by igniting a sheet of rocket propellant which had been placed behind the plate. The plates in the spray cooling experiments were preheated to approximately 300°C and data was taken as the water cooled the plate. One flow rate was used in the flow cooling test (1.0 GPM/foot width) and two flow rates (1.10 GPM and 0.25 GPM/square foot of area) were used in the spray cooling tests. Heat transfer data from both the spray cooling and flow cooling tests were used in a computer program to compute the effectiveness of water cooling aluminum plates on ships exposed to the thermal radiation pulse of a nuclear weapon detonation. The value of water cooling is shown by comparing the maximum plate temperatures with and without cooling for weapon yields of 100 and 1000 kilotons, aluminum plate thicknesses between 1/8" and 1/4", and ship to weapon distances corresponding to peak airblast overpressures up to 15 psi.

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THE USE OF WATER COOLING FOR PROTECTION AGAINST THERMAL RADIATION FROM A NUCLEAR WEAPON DETONATION

This effort is part of a continuing study to find methods of protecting shipboard elements from the combined thermal radiation and airblast effects of a nuclear weapon explosion. This report presents the data from heat transfer experiments to measure the heat transfer rates of a water flow or spray on a heated plate. These data were used to predict the effectiveness of water cooling the exposed aluminum plating on ships which are subjected to a nuclear weapon detonation. This work was sponsored by the Naval Ship Systems Command under Task Number SHIP-11292/SF52-553-001.

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By direction

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I INTRODUCTION

This report presents the results of a study to estimate the effectiveness of water cooling an aluminum plate which is being exposed to the thermal radiation pulse of a nuclear weapon explosion. In a practical cooling system for shipboard use, water can be applied either by an open nozzle which allows downward flow in a thin sheet ("washdown system") or by a spray nozzle which breaks the flow into droplets ("spray system"). Separate experiments were performed for both types of cooling. The spray-flow experiment consisted of initially instrumenting a small (3" by 3") thin aluminum plate with thermocouples. This plate was pre-heated to approximately 300°C and cooled by water spray from a single nozzle. Tests were conducted for two water flow rates, 0.25 and 1.10 GPM per square foot of plate. The sheet flow experiment consisted of initially instrumenting a large (24" long by 2.5" wide) thin aluminum plate with thermocouples. This plate was heated by igniting a thin sheet of propellant which was placed 2.5 inches behind the plate and simultaneously cooled by a sheet of water flowing down over the front surface. A flow rate of 1.0 GPM per foot of width was chosen because this was the minimum flow rate which would completely wet the plate. In each of these experiments, heat transfer rates were deduced from the plate temperature histories as the cooling water passed through convective and boiling phases. These experimentally determined heat transfer coefficients, heat transfer rates, and locations of boiling regimes were used to predict the temperature histories on plates exposed to the thermal pulse of a nuclear weapons explosion. That is, computer programs were written using numerical methods to construct in a stepwise manner the temperature histories of plates which are simultaneously heated and cooled. The effectiveness of water cooling in protecting ships structures against the thermal pulse of nuclear weapons was estimated by computing the maximum temperature in typical aluminum plates for representative values of weapon yields and peak airblast overpressures (i.e., distance from weapon to ship). The results of these computations predict for aluminum the degree of thermal protection against the thermal pulse of a nuclear weapons explosion for either spray or flow cooling.

The combat capability and effectiveness of surface ships would be enhanced by protecting them against the effects of nuclear weapon explosions. Although surface ships cannot be expected to withstand a well targeted direct attack, they should be capable of withstanding effects of a weapon directed against a high priority target in their vicinity.

A nuclear weapon detonation in the vicinity of surface ships will cause the following weapons effects, (1) structural heating, stresses, and strains, due to the thermal radiation pulse, (2) effects of electromagnetic pulse and transient radiation on electronic equipment, (3) physical damage due to the blast wave, and (4) contamination due to radioactivity and radioactive fallout. The immediate threat to ship or ship systems survivability is destruction caused by the blast and/or the thermal waves. The thermal pulse travels much faster than the blast wave and in situations of interest, most of the total thermal energy has arrived at the target prior to the arrival of the blast wave. Thermal energy heats any exposed ship structure or system and may cause failure by melting or combustion or may weaken and stress a structure so that it is more vulnerable to the oncoming blast wave. Because thermal effects of a nuclear explosion are felt first, a consideration of immediate ship survivability should begin with an analysis of the effect of the thermal energy pulse. The irradiance history of the thermal energy pulse is characterized by a very rapid rise to a peak irradiance followed by a gradual decay to zero irradiance. The time to maximum irradiance(tm) increases with weapon yield and is of the order of 0.1 seconds for a 10 kiloton yield weapon and 1.0 second for a 1000 kiloton (1 megaton) yield. The thermal pulse, normalized about its maximum, is illustrated in Figure 1 (reference (1)). The fact that thermal radiation exposure commences almost instantaneously after a nuclear weapons explosion makes it difficult to protect structures with water cooling since it takes a finite amount of time to activate a water cooling system. For the purposes of this report, it is assumed that a system can be designed to sense a nuclear weapon explosion and apply either spray or flow water in 0.5 seconds. The sensing time of 0.5 seconds was also used in the calculation of structure temperatures mentioned above.

The present work is part of a continuing effort to determine the susceptibility of shipboard structures to thermal radiation and thermal radiation-airblast interaction effects and to develop techniques to protect against these effects. This report concludes the initial effort to provide a thermal protection method for the thermal pulse of a nuclear weapon explosion. Past efforts centered on the calculation of temperature histories in structural elements such as plates (reference (2)), cylinders (reference (3)), or rectangular sections ("box beams") (reference (4)). Future work includes an examination of the effectiveness of coating systems in reducing temperature rises resulting from thermal pulse heating to acceptable levels.

II SYMBOLS

A = area

C = constant (see equation (2))

C_p = specific heat at constant pressure

E = percent statistical error in temperature measurement

h = heat transfer coefficient

h_{fg} = heat of vaporization

H_{max} = maximum irradiance

g = acceleration of gravity

L = reference length

N = number of samples

P = pressure

q = heat transfer rate per unit area

q_c = constant heat transfer rate per unit area

q_{in} = heat transfer rate that goes into heating a plate

Q_T = total water flow rate in the boundary layer

S = unbiased standard deviation

t = thickness

T = temperature

T_c = critical (Leidenfrost) temperature

T_i = temperature data point

T_{max} = time at which maximum irradiance of nuclear weapon thermal pulse occurs

T_s = surface temperature of aluminum plate model

T_w = water temperature

u = velocity component in the x direction

U = water velocity

U_e = water velocity at edge of boundary layer (water surface velocity)

U_m = maximum water velocity at edge of boundary layer

U_o = initial water velocity

U_r = reference velocity

v = velocity component in the y direction

V = volume

W = mass flow rate

x = coordinate direction down and along the inclined plate model

y = coordinate direction perpendicular to and outward from the inclined plate model

yb = boundary layer thickness

μ = statistical true mean temperature

γ = viscosity

ρ = density

σ = water surface tension

T = time

Superscripts

/ = dimensional quantities (in Appendix A)

– = average value

Subscripts

f = liquid water

g = gaseous water vapor

III HEAT TRANSFER THEORY FOR WATER COOLING WITH BOILING

The use of water as a cooling medium for a structure being heated by a relatively large heat source, such as a nuclear weapons thermal pulse, is characterized by an early transition to boiling. For example, water existing on an open surface (i.e. at atmospheric pressure) will commence boiling when the surface temperature reaches about 150°C (reference (5)). There are two basic types of boiling; namely, pool boiling and flow boiling. Pool boiling occurs when the surface being heated is submerged in a pool of initially stationary liquid. Flow boiling is boiling in a flowing stream of fluid. Although only flow boiling is considered herein, pool boiling is mentioned because many of the theories for flow boiling begin with results from pool boiling. The determination of boiling heat transfer rates are complicated by the existence of several regions of boiling. That is, once boiling is initiated, the heat transfer rate does not continue to increase as the external heating is increased until the fluid is completely evaporated. Instead, the heat transfer rate will increase or decrease depending on the governing boiling phenomena.

The following is a description of the heat transfer regions encountered by water flowing over a surface which is being rapidly heated. A more detailed description of these regions is given in references (5) and (6). First is the non-boiling region where the heat transfer rates are characterized by a forced-convection coefficient, h.

$$q = h(Ts-Tw) \tag{1}$$

This heat transfer coefficient can be derived analytically using boundary layer theory (reference (7)), or it is given by older empirical formulas (reference (8)). Next is the forced convection-surface boiling or partial nucleate boiling region where the heat transfer rates continually increase as external flux increases due to an increasing amount of nucleate boiling. Initially, nucleate boiling occurs only in patches along the heated surface while forced convection exists in between. As the external heat flux increases, more nucleation sites are activated and the boiling patches increase in size until the entire surface is in boiling. This is the beginning of the fully developed nucleate boiling region and the heat transfer rate increases as still more nucleation sites are activated until a critical (maximum) heat flux is reached. For fully developed nucleate boiling the data of many investigators can be represented by a simple equation,

$$q = C(Ts-100)^n \tag{2}$$

for atmospheric pressure and Ts in degrees centigrade. The surface minus saturation temperature difference, Ts-100, has been found to be the governing parameter for nucleate boiling phenomena. The heat transfer rates for the region of partial nucleate boiling are usually estimated by constructing an empirical boiling curve which connects the region of forced convection (equation (1)) to the region of fully-developed nucleate boiling (equation (2)). Formulas for this region are defined in several ways in references (5) and (6).

Many attempts have been made to predict the maximum heat transfer rate of the nucleate boiling region. These have necessarily been based on empirical or semi-empirical explanations of nucleate boiling due to the extreme complexity of the boiling process. One successful theory is the additive prediction method described in reference (9) . Here the maximum heat transfer rate for nucleate boiling is assumed to consist of two terms. One term represents the boiling contribution to the heat transfer rate in the absence of forced convection (i.e. pool boiling), and the second term is a contribution for the equivalent forced convection in the absence of boiling. The pool boiling factor is found by multiplying the maximum heat flux for a fluid at its saturation temperature by a factor to account for subcooling. In the present work, water is subcooled if its temperature is below 100°C (the saturation temperature at atmospheric pressure). The Zuber-Kutateladze equation (reference (10)) for flow on a flat plate can be used to predict the maximum heat flux for saturation pool boiling.

$$qmax1 = .0131 \cdot hfg \cdot (\rho_g)^{1/2} \sqrt[4]{\sigma \cdot g \cdot (\rho_f - \rho_g)} \sqrt{1 + \frac{\rho_g}{\rho_f}} \quad (3)$$

and the factor to account for subcooling is given in reference (10) by

$$F = 1 + \left(\frac{\rho_f}{\rho_g}\right)^{0.8} \left(\frac{Cp \cdot (Ts-Tw)}{15.38 \cdot hfg}\right) \quad (4)$$

The forced convection heat transfer rate is given by equation (1) with the surface temperature replaced by the temperature at which the critical (maximum) heat transfer rate occurs (i.e. the Leidenfrost temperature). This contribution can be expressed as:

$$qmax2 = h(Tc-Tw) \quad (5)$$

The final correlation for peak nucleate heating is given by a combination of equations (3), (4), and (5).

$$qmax = F \cdot qmax1 + qmax2 \quad (6)$$

The attainment of a critical or maximum heat transfer rate for nucleate boiling leads to an unstable region termed partial film boiling or transition boiling when the surface temperature is raised above the Leidenfrost temperature by the external heat flux. Instability results when so many bubbles of water vapor are formed on the surface that they interfere with the oncoming liquid. This vapor thus forms an insulating layer which forms moving patches on the surface. The onset of instability is termed boiling crisis or burnout and is characterized by a large drop in heat transfer rate accompanied by a rapid rise in surface temperature. The unstable region of partial film boiling leads to the region of stable film boiling when the vapor layer completely covers the surface and insulates it from the cooling water. At this point the heat transfer rates are at a local minimum and further increases in the surface temperature cause increases in the heat transfer rate only because the thermal radiation rate increases from the hot surface. The partial film boiling region where film boiling and nucleate boiling exist side by side is unstable and difficult to measure. No satisfactory theory exists and the usual practice in estimating heat transfer rates is to assume that stable film boiling occurs immediately after the critical flux is exceeded. The stable film boiling region can be analyzed using boundary layer theory and this has been done for laminar flows (reference (11)), turbulent flows (reference (12)), and subcooled flows (reference (13)). Stable film boiling is described by a heat transfer coefficient which is characteristic of convective heat transfer (see equation (1)) as opposed to the power of the temperature differences law for nucleate boiling (see equation (2).

It is possible that the sequence of events from nucleate boiling to critical nucleate boiling to film boiling will not occur at all. That is, if boiling is occurring in a sufficiently thin layer with turbulence at the liquid-vapor interface, heat may be conducted through the liquid layer and evaporation take place at the liquid surface instead of the structure surface. This type of a heat transfer mechanism is termed "forced convection vaporization" and is characterized by a heat transfer coefficient. Theories yielding these coefficients are discussed in reference (6). However, this type of boiling usually occurs only in high void annular flows where the core vapor velocity is high enough to cause sufficient turbulence at the liquid-vapor interface.

The heat transfer regions for water cooling a surface have been extensively studied, principally through experimentation, but also analytically using boundary layer and other theory. It has been found that the boiling regions are very difficult to describe accurately because they are affected by many parameters such as fluid flow rate, pressure and viscosity, structure material and especially structure surface conditions. For example, the exponent, n, in equation (2) has been found to range between two and five for nucleate boiling due primarily to choice of structure material and condition of the surface. Also, in the examination of data for maximum nucleate boiling, it was found that equation (6) correlates the experimental values only within a deviation of 40 percent. These uncertaintities make the obtaining of a complete and accurate description of the heat transfer rates throughout all of the regions of convection and boiling a nearly impossible task for a given system. Thus, two sets of experiments were run in the present work, one for a washdown type system where the water flows down over the surface, and secondly for a spray type system where the water impinges normal to the surface.

IV EXPERIMENTAL ARRANGEMENT

Experiments were run to determine the heat transfer behavior of water cooling on heated models. The data from these experiments will be used to determine the effectiveness of water cooling ship structures which have been simultaneously heated by the thermal radiation pulse of a nuclear weapon explosion. Two different types of water cooling tests were run. In the first experiment, a sheet of water is allowed to run down and over the plate being heated. This is termed a washdown type of cooling. In the second experiment, water is sprayed perpendicularly onto a heated plate using a spray nozzle. This is termed spray cooling. The plate models, cooling and heating methods and data recording equipment, are described below for each experiment.

Washdown Heat Transfer Tests

The experimental setup for measuring the heat transfer coefficient for an aluminum/water flow interface consists basically of a water supply, an aluminum trough instrumented with thermocouples, and a chemical heating source with a rapid uniform ignition and the capability of transferring at least 10 calories/cm^2-sec to a 1/2 square ft target. A picture of the experimental equipment is found in Figure 2.

The trough was made from a 1/4" x 3" x 24" 6061-T6 aluminum plate. A 1/8" x 2 1/2" x 24" center section was milled out of the plate leaving a trough with 1/8" high walls. Fourteen Iron-Constantan thermocouples were spot welded to the front surface and then connected directly to Burr-Brown instrumentation amplifiers and recorded on an FM taperecorder. The cold junction was at room temperature which was monitored.

The trough was mounted to a transite sheet baseplate (1/4" x 5" x 24") with side brackets so that there was a 2 3/8" separation between the back surface of the trough and the baseplate. A 1/4" x 3" x 24" piece of propellant (heating source) was attached to a 1/8" x 3 1/2" x 24" transite sheet two inches from the back surface of the trough as a means of producing a uniform rapid ignition. The propellant was instrumented with a nichrome wire grid and painted with a pyrotechnic ignition mixture. The nichrome wire was electrically heated to incandescence. This caused the AlA pyrotechnic mixture to ignite at many points. These isolated burning areas rapidly spread out over the entire surface (within one second) yielding a reasonably uniform ignition of the propellant surface.

The paint was a combination of nitrocellulose, butyl acetate, and a zirconium-based pyrotechnic ignition powder (AlA). The basic mixture by weight was 3-5 percent nitrocellulose and 95-97 percent AlA with butyl acetate used as a thinner. The use of this mixture as a preignitor was necessary because of the relatively slow burning rate of 1-1 1/2 ft/minute of the propellant at atmospheric pressure. Only at much higher pressures as would be found in rocket engines would ignition rates be on the millisecond time scale. The burn rate of the paint is at least an order of magnitude greater. A detailed description of the use of propellant heating is given in reference (14).

The flow was initiated by forcing a water-photoflow mixture through a closed tube with a single line of equally spaced 1/16" diameter holes. A water-photoflow mixture of 2000:1 was used in order to reduce channeling of the flow down the trough. The water poured onto the trough from a height of 2" and became a nearly uniform flow (constant layer thickness both across and down the trough) by the time it had traveled 2" down the trough. There was some turbulence at the edges but these were spatially damped toward the center of the trough.

An attempt was made to measure the maximum flow velocity at different flow rates. The rate was controlled by a valve in series with the pump and the tube. Then 1/4" diameter disks were dropped onto the flow near the top of the trough and photographed with a 16 mm movie camera with a frame rate of 128 frames/sec. A scale had been previously etched on the trough. The film plane was placed parallel to the trough surface to reduce parallax and at a distance away so that the 24" long trough would just fill up the longest side of the 16 mm frame. The flow velocity was then simply determined by measuring the position of the disk as a function of time and differentiating with respect to time.

The flow rate was measured by dividing the amount of water collected at the bottom of the trough in a specified time (30 sec) by the collection time.

V EXPERIMENTAL MEASUREMENTS

Both the washdown and spray cooling experiments were designed to measure temperatures in the various boiling regions from which heat transfer rates would be deduced. The washdown system experiment requires a further test to measure the velocity of the thin water layer flowing down the trough. This section will present the data from each of these experiments. Analysis of these data and derivation of heat transfer laws for the boiling regions of both washdown and spray systems will be treated in succeeding sections.

Water Velocity Tests

The experiment to measure the velocity of the water layer flowing down the inclined trough was performed in support of the washdown heat transfer experiment. The nozzle flow rate (1.0 GPM/ft), the trough inclination (33^{o} from the vertical), and the trough dimensions (2.5" by 24") all duplicated the conditions of the washdown heat transfer tests. Six velocity tests were performed by dropping 1/4" diameter paper disks onto the flow and photographing their path down the trough with a 16 mm motion picture camera. By viewing the film a frame at a time, a distance-time history of the disk was obtained with data points given every .0078 (1/128) seconds. To avoid errors in measuring small distances, ten data points (approximately 2.5") were used to compute a velocity. It was found that the velocity increased to an approximately constant value for each test but that there were significant differences in the terminal velocities observed in different tests. These differences were attributed to the disks sinking to different depths in the flow as a velocity gradient exists in the thin water layer. Note that the thickness of the disks (approximately .005") is about 1/2 the expected depth of the water layer. Because these tests indicated that the flow exhibits a velocity gradient characteristic of boundary layer flow, it was decided to calculate the average velocity using boundary layer theory. The flow model assumes that the water layers begin at a single velocity with no boundary layer but that the boundary layer grows until it includes the entire flow. The boundary layer cannot grow further and the velocity profile remains fixed due to a balance between the inertial (frictional) forces and the gravitational forces. A development of boundary layer theory for this flow model and the calculation of the terminal velocity profile is given in Appendix A. An average velocity of 2.30 ft/sec was found and shown to be in general agreement with the measured velocities. This calculated value will be used in all further heat transfer data reduction schemes for the washdown heat transfer tests.

Washdown Heat Transfer Tests

The washdown heat transfer tests were performed using the propellant heating source and the instrumented inclined trough (see Figure 2) previously described. A heating test was accomplished by first adjusting the water flow rate to approximately 1.0 GPM/ft then starting the recording equipment and finally igniting the propellant. Six tests were made in this manner along with seven similar tests with no water flow. The water-off tests were run to measure the heating rates due to the propellant heating and a water-on tests for the net heat transfer rate with cooling. The difference between the propellant heating rate and the net heating rate is the amount of heating removed by the water. This quantity is important because it determines the convective heat transfer coefficient and the boiling heat transfer rates for washdown cooling. There are 11 thermocouple stations for all of these tests. The locations of these stations are given in Figure 3. The washdown heat transfer test data consisting of temperature histories for all 13 tests and 11 stations are presented in Appendix B.

An examination of the temperature histories given in Appendix B shows that there are variations in their shapes, especially for the water-on situation (see Figures B-1 through B-11). Although the tests were run under nearly identical conditions, there were apparent unforeseen differences, such as in flow behavior or in propellant heating rates. It is possible that a slight change in conditions can cause a large change in observed temperature history in a water-on experiment. This is because the boiling of thin layers may not be well-behaved when subjected to slightly-changed conditions. Because of the observed variations in experimental data, it was decided to average the observed temperature histories. These are given in Appendix B along with a statistical estimate of the error incurred by averaging. An average temperature history was obtained for each thermocouple station for both the water-on and the water-off condition.

The average temperature history is used to compute the heat transfer rate in the following manner. First, since the aluminum plate was thin (1/8"), a one dimensional heat flow equation neglecting heat conduction is used to define the heat transfer rate. This is given as:

$$q = \rho \cdot cp \cdot t \cdot \frac{\partial T}{\partial T} \qquad (7)$$

Equation (7) is integrated over small time intervals by making two assumptions. These are : (1) The variation of specific heat is linear and the density is constant with temperature; i.e.

$$Cp = A + B \cdot T \qquad (8)$$

and (2) The heat transfer rate is constant over the small time interval, ΔT. The result of integration is:

$$qc = \frac{\rho \cdot t}{\Delta T} [A \cdot \Delta T + B \ \Delta T^2/2] \qquad (9)$$

Equation (9) was applied to the temperature histories of the washdown heat transfer tests with a time increment, ΔT, of 0.1 seconds. The result is a sequence of heat transfer rates, qc, for each thermocouple station which are only piecewise continuous over the total test time. Continuous heat transfer rates were derived by fitting a third order polynomial to the individual qc values using the method of least squares. The final heat transfer rates are the polynomial values taken at 0.1 second intervals. Heat transfer rates were reduced in this manner for each thermocouple station and for both the water-on and water-off average temperature histories. These heating rates are given for the 0.1 second time increments in Tables 1 through 11 along with the average temperature history and the estimated error in temperature as defined in Appendix B. The difference in the water-off (no cooling) and water-on (with cooling) heat transfer rates will be used later to define the heat transfer coefficients and boiling heating rates for a washdown system.

The errors in temperature history given in Tables 1 through 11 are seen to be initially large but decrease to the 5-10 percent range after approximately 1.0 second of testing. These large initial errors are unimportant since the heat transfer results will be derived from later data only. The relatively small errors for boiling data (i.e., data after approximately 2.0 seconds of testing) may be fortuitous in that temperature changes due to changes in flow conditions or local propellant heating rate may be offsetting the random experimental errors. In any event, the purpose of these experiments was not to investigate the nature of the boiling process but rather to develop some heat transfer laws that will predict the temperature response of structures to rapid heating. For this purpose, averaged temperatures showing statistically small differences from individual test histories should be sufficient.

Spray Heat Transfer Tests

The spray heat transfer tests were performed by spray cooling the instrumented heated aluminum plates with water from a standard nozzle. A heating test was made by first heating the plate to approximately 300°C, then starting the recording system, and finally spraying the plate with water. Tests were made for flow rates of 0.25 and 1.10 GPM/ft^2 of sprayed area by varying the plate to nozzle distance 9" to 15". No special significance is attached to these flow rates except that they serve to study the effect of varying the flow rate. Three tests were performed for each nozzle distance to test the repeatability of the measurements. The data of these tests consists of temperature histories for the five thermocouple stations. Only water-on tests are necessary here because no heat was added during the spray cooling. The heat transfer rates were computed from the measured temperature histories by using equations (7) through (9) and the polynomial-fit method described above for the washdown heat transfer tests. The temperature histories for spray cooling will not be presented herein. Instead, heat transfer results will be given later when the convective coefficients and boiling heat transfer laws are discussed.

Spray Cooling Heat Transfer Tests

The experimental setup for the spray cooling tests consists of a water supply and spray nozzle, an aluminum plate model instrumented with thermocouples, and a heating source. The model is a square (3" x 3") plate, 0.2" thick, and cut from stock 6061-T6 aluminum. It was mounted at the center of a 12" x 24" sheet of "transite" (asbestos based fiber board). A 3" x 3" square hole was cut into the transite and the aluminum plate model mounted behind it with a 1/4" offset. The aluminum plate was instrumented with five iron constantan thermocouples welded to its backside. Four of these were placed on a 2" diameter circle with a thermocouple on each corner of the plate. The remaining thermocouple was placed at the center of the circle.

Immediately prior to a test, the plate model was heated to approximately 300°C by use of a hot air heat gun (Master Appliance model HG 751 J). A test is initiated by directing a water spray onto the center of the plate and recording the resulting temperature changes. Ordinary tap water at 25°C was used and a flow rate of 0.195 GPM measured by a rotometer was supplied to the nozzle. The single spray nozzle used for all tests was a Spraying Systems Company Fulljet Nozzle (model 1/8 GG 1). The flow rate received by the plate was varied by changing the plate to nozzle distance from 9" to 15". The flow rate on the plate was measured in separate tests. In these tests, the aluminum plate model was removed and a plastic beaker cemented to the backside of the transite completely covering the 3" x 3" hole. A small hole was bored into the side of the beaker so that water could be lead into a graduated cylinder by a tube. Three separate tests at each plate to nozzle distance were made to determine the spray flow rate received at the plate. For a nozzle to model distance of 9", a flow rate of .069 GPM strikes the plate. This will be given as a spray flow rate of 1.10 GPM/ft^2 of plate. For a distance of 15", a flow rate of .0155 GPM or .25 GPM/ft^2 strikes the plate. The spray nozzle has a conical spray angle of approximately 54° and the measured flow rates are somewhat higher than a prorated rate based on the total flow (.195 GPM) and the portion of the area intercepted by the plate.

Data Analysis

Temperature data for both the washdown and spray cooling tests were recorded on a 14 channel FM tape recorder (Precision Instrument Corporation). The outputs of the recorder channels were filtered using 4-pole low pass (20 Hertz) electronic filters and digitized sequentially at the rate of 100 samples per channel per second. An Analogic 12 bit A to D converter and a Honeywell DDP 516 computer were used to produce a digitized tape of the data. This digitized tape is compatible with the NOL CDC 6400 computer and standard programs were used to read the tape.

VI EXPERIMENTAL HEAT TRANSFER LAWS FOR WASHDOWN AND SPRAY COOLING SYSTEMS

The experimentally measured temperature histories will be used to derive the convective coefficients and boiling heat transfer rates for both washdown and spray cooling systems. Tests on the aluminum plate models for both systems are expected to show heat transfer regions having boiling characteristics. That is, water initially transfers heat by convection until the model surface temperature goes above 100°C. Next, there is partial and fully-developed nucleate boiling until a critical surface temperature is reached and boiling heat transfer rate is a maximum. Further plate heating causes partial or stable film boiling regions. These regions are characterized by large increases in model temperature and in a loss of cooling effectiveness by the water. This section will use the experimental heat transfer data previously discussed to derive heat transfer laws for both washdown and spray cooling systems.

Washdown Cooling Systems

The temperature histories and heat transfer rates given in Tables 1 through 11 for thermocouple stations on the inclined plate model will be used to derive heat transfer laws for a washdown cooling system. The first region encountered is that of forced convection cooling which is characterized by a convective coefficient, h (see equation (1)). This coefficient was derived from the data in the following manner. First, the plate model is divided into ten elements, each 2.0" in length. Then a temperature history of the cooling water is computed by making energy balances on each element and incrementally building up the water temperature as it flows down the plate. The relationship between the heat rate transferred to a flowing stream and the resulting temperature rise is:

$$\Delta q = W \cdot Cp \cdot \Delta Tw \quad (10)$$

In equation (10), Δq is the difference between the heat transfer rate with no cooling and the heat transfer rate with cooling. The time that water is at an element is the ratio of element length (1/6 ft) to the average water velocity (2.3 ft/sec), hence:

$$\Delta T = 0.0725 \text{ seconds} \quad (11)$$

In summary, the water temperature history at every element is computed in a stepwise manner from the heat transfer rates given at each element in Tables 1 through 11, the time step of .0725 seconds, and incremental temperature rises computed from equation (10). Linear interpolation was used in the tables to find heat transfer rates at times which are multiples of the time step, ΔT. Also, it was necessary to use the heat transfer rates for station 1 (x = 4") for station 0(x = 2") due to the failure of the thermocouple at station 0. At stations 4, 5, 6, and 7 where two thermocouples were on the same element, an averaged value of heat transfer rate was used. Once the water temperature had been determined at each element, the convective heat transfer coefficient was calculated by equation (1) and the experimental heat transfer rates given in Tables 1 through 11. This numerical technique to calculate heat transfer coefficients will give reasonable results only if the heat transfer data is accurate. This is true because experimentally determined heat transfer rates are used to compute both the water temperature and the heat transfer coefficients. An earlier examination of the data indicated that there might be rather large statistical errors in the first 1.0 second or so of the data. The data showed that boiling does not generally occur until about 2.0 seconds of test time. Therefore, only the test period between 1.0 and 2.0 seconds will be used to determine the heat transfer coefficients for the convective cooling region. The convective coefficients so determined are plotted in Figure 4 as a function of water temperature. For comparison, the predictions of an empirical law for turbulent boundary layers (reference (8)) and the theoretical boundary layer result of reference (7) are shown on this figure. The predictions of the theoretical method are slightly higher than the experimentally determined values but the overall agreement is quite good considering the complexity of the experiment. From the data a representative value of convective cooling coefficient is taken. It is:

$$h = 0.075 \text{ cal/sec cm}^2 \, {}^{o}\text{C} \quad (12)$$

The use of the constant value of heat transfer coefficient given by equation (12) for calculating cooling of plates with washdown systems will not cause an appreciable error because the total heat removed in the convection region is small compared to the heat removed in the boiling regions.

Boiling heat transfer data is found in Tables 1 through 11 for test times after approximately 2.0 seconds. An inspection of this data shows that the first three thermocouple stations (x = 4", 6", and 8") had a significantly higher rate of heat transfer to the water, Δq, than the remaining eight downstream stations. This was expected because the water temperature is lower for stations nearer the nozzle. In fact, the calculated water temperature of only the eight lower stations remained above 100°C after approximately 2.0 seconds of test time. This indicates that some form of boiling must be occurring at these stations. The experimental heat transfer rates are shown in Figure 5 for the upper three stations and Figure 6 for the lower eight stations. In these figures, boiling heat transfer rates are plotted against both test time and average plate temperature. An inspection of this data shows that the measured heat transfer rates do not behave like other experimental observations previously discussed (see Heat Transfer Theory For Water Cooling With Boiling). Some of the obvious differences are : (1) There is only a very small region of presumable nucleate boiling before the maximum heat flux is reached. (2) There is an actual decrease in plate temperature due to boiling and this causes the average plate temperature to drop below 100°C. (3) After the critical heat flux there is a large region of boiling which should be the partial and stable film boiling regions. This region is characterized by heat transfer rates of approximately 8.0 cal/sec cm^2 for the upper three stations and approximately 5.5 cal/sec cm^2 for the lower eight stations. These heat transfer rates are at least an order of magnitude higher than those of stable film boiling. This region may be a return to partial nucleate boiling, may be a forced-convection vaporization region, or may be a combination of boiling mechanisms including film boiling. Because of the observed differences, it is concluded that the boiling heat transfer phenomena observed during these washdown tests do not follow the boiling laws of previous investigators. Therefore, new laws will be derived from the experimental data.

The initial boiling curve is derived from data which is found between approximately 2.0 and 3.0 seconds of heating time. Because of the decrease in model wall temperature due to boiling, it is impossible to find a simple power law based on the wall minus saturation temperature difference. Instead, a power law based on test time will be found. The use of time as a characteristic parameter is justified for the present purpose of using the results of the present data to calculate the temperature rise of aluminum plates cooled by washdown systems exposed to the thermal radiation pulse of a nuclear weapon explosion. This follows because the total external heating for the washdown tests (approximately 40 cal/cm^2 in 5.0 seconds) is of the same order of magnitude as that received from a nuclear weapon thermal radiation pulse. For example, the total heating (radiant exposure) received from a 100 kt nuclear weapon explosion at a location where the peak airblast overpressure is 5 psi is approximately 39 cal/cm^2 (up to the blast arrival time of 4.3 seconds). An inspection of the data shows that there is a period of 0.3 second in which a maximum boiling heat transfer rate is reached (from test times of 2.1 to 2.4 seconds). Two expressions for heat transfer rate were derived for this region by fitting curves through the experimental data. For the upper three stations,

$$q = 18.3(T + 0.1)^{0.375} \tag{13}$$

For the lower eight stations, the heat transfer rate is given by

$$q = 12.7(T + 0.1)^{0.375} \tag{14}$$

Where T is the time of initial boiling ($0 \le T \le 0.3$ seconds). The time at which boiling commences is governed by the water temperature. When the calculated water temperature reaches 100°C, initial boiling is assumed to occur and equations (13) and (14) will be used to calculate the rates of heat transfer.

Expressions for the boiling heat transfer rates immediately after the maximum boiling heating rate were also found by fitting curves through the experimental data. These are, for the upper three stations,

$$q = 6.85(T + 0.1)^{-0.7} \tag{15}$$

For the lower eight stations,

$$q = 4.12(T + 0.1)^{-0.85} \tag{16}$$

Where T is the time of boiling after 0.3 seconds and up to 0.7 seconds. For all later values of time, the boiling data is represented by constant heat transfer rates (see Figures 5 and 6). For the upper three stations,

$$q = 8.0 \text{ cal/sec cm}^2 \tag{17}$$

For the lower eight stations,

$$q = 5.5 \text{ cal/sec cm}^2 \tag{18}$$

The boiling curves given by equations (13) through (18) are shown on Figures 5 and 6. It is seen that the boiling laws chosen represent the experiment only approximately and that there is considerable scatter in the experimental data. A calculation was made to prove that the uncertainty in the data will not prevent the expressions derived for boiling heat transfer rates from at least predicting accurately the average plate temperature. For this purpose, a computer program based on the above boiling laws and using numerical methods, was written to compute temperatures on a rapidly heated, inclined plate which is cooled by a washdown system. The details of constructing this program are given in Appendix C. A slight modification of this program to calculate the temperature of plates heated by the thermal radiation pulse of a nuclear weapon explosion is also given in Appendix C. The external heating rate for checking the experimentally derived boiling laws was supplied by specifying the heat transfer rate without cooling as given in Tables 1 through 11. Temperatures computed by the program were averaged and compared with an average of the model plate temperatures given in Tables 1 through 11 (data with water cooling). This comparison is shown in Figure 7. The excellent agreement between computed and experimental temperatures indicates that the heat transfer laws derived from the data (equations (13) through (18)) can be successfully used to predict average plate temperatures.

VII SPRAY COOLING SYSTEMS

Heat transfer results from the spray cooling tests are found to be in qualitative agreement with the boiling theory previously discussed. That is, water sprayed onto heated plates appears to go through film boiling, nucleate boiling, and convection heat transfer regions as the plate is cooled. This is encouraging because the observations of other investigators were made under radically different conditions such as flows through heated channels or pipes. The spray cooling tests are simpler to analyze because the plate was not being simultaneously heated and cooled. It is assumed that the heat transfer rates to the water for cooling a heated plate are the same as those for simultaneously heating and cooling the plate. Heat transfer rates derived from data of the three most upper thermocouples differ markedly from those found for the lower thermocouples (thermocouple locations are shown in Figure 8). This is apparently due to the build-up of a thin water film which flows down the plate as it is being sprayed. This water film is not observed on the upper thermocouples. Hence, heat transfer data will be reported for upper and lower plate regions separately and for the two nozzle flow rates, 0.25 and 1.10 GPM/ft^2. The results of three identical tests for each flow rate will be given.

The forced convection heat transfer region was reached when the plate was cooled below 100^{o}C. Forced convection heat transfer coefficients were computed from the measured heat transfer rates and equation (1). In equation (1), the water temperature is at its initial value of 24^{o}C. Forced convection coefficients are shown in Figure 9 for the upper thermocouples (i.e., without film flow) for the nozzle flow rates of 0.25 and 1.10 GPM/ft^2. These data were averaged to obtain representative convective coefficients. For a flow rate of 1.10 GPM/ft^2,

$$h = 0.262 \text{ cal/sec cm}^2 \text{ }^{o}\text{C} \tag{19}$$

For a flow rate of 0.25 GPM/ft^2,

$$h = 0.243 \text{ cal/sec cm}^2 \text{ }^{o}\text{C} \tag{20}$$

Similarly, forced convection coefficients for the lower thermocouples (i.e., with film flow) are shown in Figure 10 for both nozzle flow rates. From these data the following convective coefficients were obtained. For a flow rate of 1.10 GPM/ft^2,

$$h = 0.223 \text{ cal/sec cm}^2 \text{ }^{o}\text{C} \tag{21}$$

For a flow rate of 0.25 GPM/ft^2,

$$h = 0.220 \text{ cal/sec cm}^2 \text{ °C} \quad (22)$$

A comparison of equations (19) and (20) with equations (21) and (22) indicates that the water film is interfering with the spray to reduce the heat transfer effectiveness. However, the coefficients measured here are still much higher than those for washdown flow previously determined (see equation (12)).

The nucleate boiling region occurred in a region of plate temperatures between approximately 160°C and 100°C. This region contains a rather large section which appears to be fully developed nucleate boiling and a very much smaller section of partial nucleate boiling. The region at temperature higher than those of nucleate boiling was taken to be the region of partial film boiling. The above assumption on boiling region location is based on an examination of the data. Heat transfer rates for the nucleate and partial film boiling regions are shown in Figures 11 and 12 for upper thermocouples and the nozzle flow rates 1.10 and 0.25 GPM/ft^2, respectively. The data is such that nucleate boiling curves can be drawn through the data of the three identical runs. These are:

For a flow rate of 1.10 GPM/ft^2,

$$q = 21.0(Ts - 100)^{0.2} \quad (23)$$

For a flow rate of 0.25 GPM/ft^2,

$$q = 19.0(Ts - 100)^{0.2} \quad (24)$$

Similarly, heat transfer rates for nucleate and partial film boiling regions are shown in Figures 13 and 14 for lower thermocouples and nozzle flow rates of 1.10 and 0.25 GPM/ft^2. The nucleate boiling curves for these data are:

For a flow rate of 1.10 GPM/ft^2,

$$q = 8.36(Ts - 100)^{0.5} \quad (25)$$

For a flow rate of 0.25 GPM/ft^2,

$$q = 6.70(Ts - 100)^{0.5} \quad (26)$$

Note that the boiling curves for the nucleate boiling region are characterized by the wall minus saturation temperature difference but that the exponents in the power law (0.2 and 0.5) are much lower than exponents obtained by other investigators of boiling phenomena (i.e., coefficients in the range 2-5). The temperature region in Figures 11 through 14 from approximately 160-220°C is assumed to be a partial film boiling region. This assumption is difficult to prove because there is no satisfactory theory for the partial film boiling region. In any case, the heat transfer rates in these regions will be assumed to vary linearly with wall temperature. For upper thermocouples, and a flow rate of 1.10 GPM/ft^2, the heat transfer rate is

$$q = 142 - 0.623 \cdot Ts \tag{27}$$

For a flow rate of 0.25 GPM/ft^2,

$$q = 120 - 0.528 \cdot Ts \tag{28}$$

Similarly for lower thermocouples, the heat transfer rates are,

For a flow rate of 1.10 GPM/ft^2,

$$q = 276 - 1.23 \cdot Ts \tag{29}$$

For a flow rate of 0.25 GPM/ft^2,

$$q = 211 - 0.942 \cdot Ts \tag{30}$$

The maximum heat flux for nucleate boiling can be computed using the additive prediction method previously discussed (see equation (6)). For the conditions of the spray cooling tests (i.e., atmospheric pressure and water at 24^{o}C), the maximum heat flux density for saturated pool boiling, given by equation (3), is 11.1 cal/sec cm^2. The factor to account for subcooling, given by equation (4), is 4.36; hence the pool boiling contribution is 48.5 cal/sec cm^2. The contribution of equivalent forced convection in the absence of boiling is given by equation (5). This is computed using the experimental heat transfer coefficients for forced convection, given by equations (19) through (20), and critical temperatures deduced from the data. For upper thermocouples, the critical temperature is about 150^{o}C (see Figures 11 and 12); and the equivalent forced convection contribution is about 30 cal/sec cm^2. For lower thermocouples, the critical temperature is about 165^{o}C; and the equivalent forced convection contribution is, coincidentally, also about 30 cal/sec cm^2. Hence, the additive prediction method predicts a maximum heat flux density for nucleate boiling of approximately 78 cal/sec cm^2 for the spray cooling tests. Examination of the nucleate boiling data shows maximum heat transfer rates of approximately 45 cal/sec cm^2 for upper thermocouples and 50 or 70 cal/sec cm^2 depending on nozzle flow rate for the lower thermocouples. Thus, the experimental measurements are in fair agreement with the additive prediction method, especially for lower thermocouples which have a water film flow component. Also, the observed critical temperatures of 150 and 165^{o}C are in agreement with other critical temperature measurements. This behavior of critical heat flux indicates that the heat transfer region between approximately 100 and 160^{o}C is a nucleate boiling region in spite of the lower than expected exponents in equations (23) through (26).

The heat transfer region at the highest plate temperatures is that of stable film boiling. Stable film boiling is characterized by a heat transfer coefficient which is calculated from the measured heat transfer rates using equation (1). In equation (1), the water temperature is at its initial temperature of 24°C. Film boiling coefficients are shown in Figure 15 for upper thermocouples and in Figure 16 for lower thermocouples. Film boiling coefficients are not expected to be influenced by the additional film water layer at the lower thermocouples because the cooling effect of the additional water flow is neutralized by the existence of the vapor layer. This expectation is borne out by an examination of the data in Figures 15 and 16. From these data, the following representative film boiling coefficient was chosen.

$$h = 0.018 \text{ cal/sec cm}^2 \text{ }^\circ\text{C} \tag{31}$$

The experimental heat transfer laws for spray cooling, given by equations (19) through (31), are used to compute the temperature rise of plate structures heated by the thermal radiation pulse of a nuclear weapon explosion. A computer program based on these laws is described in Appendix C.

VIII EFFECTIVENESS OF WASHDOWN AND SPRAY COOLING SYSTEMS

The experimental heat transfer laws and the computer programs developed for washdown and spray cooling systems will be used to determine if water cooling at the flow rates studied is an effective method of protecting vertical plates found on ship structures against the thermal radiation pulse of a nuclear weapon explosion. Both systems will be tested by computing the temperature response in aluminum plates for given nuclear weapon explosions. Aluminum plates typically found in ship structures are between 1/8" and 1/4" in thickness, hence calculations are made for plate thicknesses of 1/8", 3/16", and 14". Nuclear weapon yields of 100 and 1000 kilotons and stand-off distances corresponding to peak airblast overpressures of 8, 10, and 15 psi will be considered. These conditions are chosen to test water cooling effectiveness because the thermal energy received up to the blast arrival time is sufficient to cause damaging temperatures in unprotected aluminum structures of these thicknesses (i.e., plate temperatures above 300°C).

A consideration of the effectiveness of a water cooling system requires that the required amount of cooling water be specified. In the washdown tests, it was found that a water flow rate of approximately 1.0 GPM/ft width was required before the water would completely wet the inclined plate (uncooled). This finding is in agreement with the theory of reference (15) for predicting the minimum flow rate to avoid the occurrence of dry patches on a vertical plate. This theory predicts a minimum flow rate of 248 lbs/hr-ft (approximately 0.5 GPM/ft) for the gravity flow of water. Furthermore, experimental measurements on a polished stainless steel plate, also reported in reference (15), required a minimum flow rate of 0.5 GPM/ft to wet the plate. Thus the minimum flow rate of 1.0 GPM/ft for unpolished aluminum plates is reasonable. There is little danger in boiling away the flow on a tall structure being cooled by a washdown system. For example, even if the peak boiling heat transfer rate of

15 cal/sec cm^2 could be maintained, it would require nearly 24 feet of vertical height to boil away the 1.0 GPM flow rate on a one foot wide plate. In the spray cooling tests, the nozzle flow rates of 1.10 GPM/ft^2 and 0.25 GPM/ft^2 were arbitrarily chosen to spray the 3" x 3" aluminum plate. The experimental data showed a considerable improvement in heat transfer rate for the higher flow rate and another increase in heat transfer rate for lower plate thermocouples which contained a film flow component. For the purposes of this report, the conditions giving the maximum heat removal will be analyzed, i.e., a flow rate of 1.10 GPM/ft^2 and lower thermocouples heating rates. The lower thermocouple heating rates are expected to occur in practical problems because the liquid film was observed to be quickly established in time and space (i.e., the upper and lower thermocouples were spaced approximately 1.5" apart vertically. The flow rate of 1.10 GPM on one square foot of plate is large in comparison with the 1.0 GPM/ft flow rate of the washdown tests. For example, a spray flow rate of 1.0 GPM will only cool about 0.9 square feet at the 1.10 GPM per square foot rate while 1 GPM running down a vertical plate one foot wide could cool a very large area. A comparison as to whether washdown or spray cooling most efficiently uses available water cannot be made because lower spray nozzle flow rates were not tested.

The temperature histories in heated vertical aluminum plates were computed for both washdown and spray cooling using the experimental heat transfer laws developed herein and the computer programs described in Appendix C. The corresponding temperature histories for plates without cooling were computed using the method of reference (18). Cooling system effectiveness will be judged by comparing the temperature at the time of blast arrival for cooled and uncooled plates. This comparison is given in Figures (15), (16), and (17) for weapon yields of 100 and 1000 kilotons, plate thicknesses of 1/8" 3/16", and 1/4", and peak airblast overpressures of 8, 10, and 15 psi, respectively. Here it is seen that washdown cooling is generally ineffective in lowering the blast arrival temperature from its uncooled value. Spray cooling is seen to be effective at standoff distances corresponding to peak airblast overpressures of 8 and 10 psi but is ineffective at 15 psi overpressure. A closer examination of the experimental heat transfer laws indicate why both types of water cooling fail as higher blast overpressures (smaller stand-off distances) are reached. The experimental laws predict maximum boiling heat transfer rates of approximately 70 cal/sec cm^2 for spray cooling and 17 cal/sec cm^2 for washdown cooling. These maxima occur only for the peak boiling rate and the heat transfer rate drops drastically when the plate temperature is increased (see Figures (5) and (11)). Therefore, water cooling effectiveness is drastically affected by an increase in the maximum irradiance (Hm) absorbed by a structure. For example, if the absorptance of a surface is 0.8, the maximum absorbed irradiance for a 1000 kiloton weapon is 77 cal/sec cm^2 at 10 psi peak overpressure but this increases to 125 cal/sec cm^2 at 15 psi peak overpressure. The larger maximum irradiance at 15 psi overwhelms the cooling capacity of the water and causes higher plate temperatures. The higher temperatures cause lower heat transfer rates which leads to still higher temperatures, etc. Hence, washdown cooling will not begin to be effective until the 5 psi peak overpressure region where the maximum absorbed irradiance is about 33 cal/sec cm^2 (1000 kiloton weapon). Also, spray cooling at the .25 GPM/ft^2 flow rate (maximum heat removal rate of about 50 cal/sec cm^2) is effective up to an overpressure of 8 psi (maximum absorbed irradiance rate of 63 cal/ sec cm^2). The above remarks on effectiveness apply strictly to thin plates, i.e., structures having wall thicknesses of 1/8" or less, and to the specific flow rates considered herein.

IX SUMMARY AND CONCLUSIONS

The purpose of this study was to investigate the use of water cooling systems to protect ships structures against the thermal radiation pulse of a nuclear weapon explosion. Heat transfer laws were experimentally derived and used in computer programs which calculate temperature rises in plates subjected to the nuclear weapon thermal radiation pulse. Calculations were made to test both spray and washdown types of cooling aluminum plates typically found on ships. The results of these calculations showed, for the flow rates tested, that washdown cooling is generally ineffective. Spray cooling was found to be adequate at stand-off distances where peak airblast overpressures are 10 psi or less. This estimated flow rate is expected to be low because it was assumed that all of the incident water could be boiled, thereby ignoring the existence of boiling regimes. The present work shows that a spray flow rate of 0.25 GPM/ft^2 was insufficient but that 1.10 GPM/ft^2 would protect an aluminum plate structure from thermal radiation exposures at stand-off distances corresponding to 10 psi peak overpressures.

The large spray flow rates required to effectively cool aluminum structures imply that spray cooling is impractical except in protecting critical equipment. For example, radar antennas which are commonly constructed of thin aluminum members could be protected by spray cooling. The use of water cooling may be generally practical if combined with other thermal protection methods. For example, assume that an aluminum structure was painted with a reflective coating that would reduce its average absorptance for thermal radiation from 0.8 to 0.4. Then, the maximum absorbed radiation for a 1000 kt explosion would be about 62 cal/sec cm^2 at 15 psi overpressure and 38 cal/sec cm^2 at 10 psi. Thus, the spray rate of 1.1 GPM/ft^2 would be effective at 15 psi and the rate of 0.25 GPM/ft^2 is effective at 10 psi.

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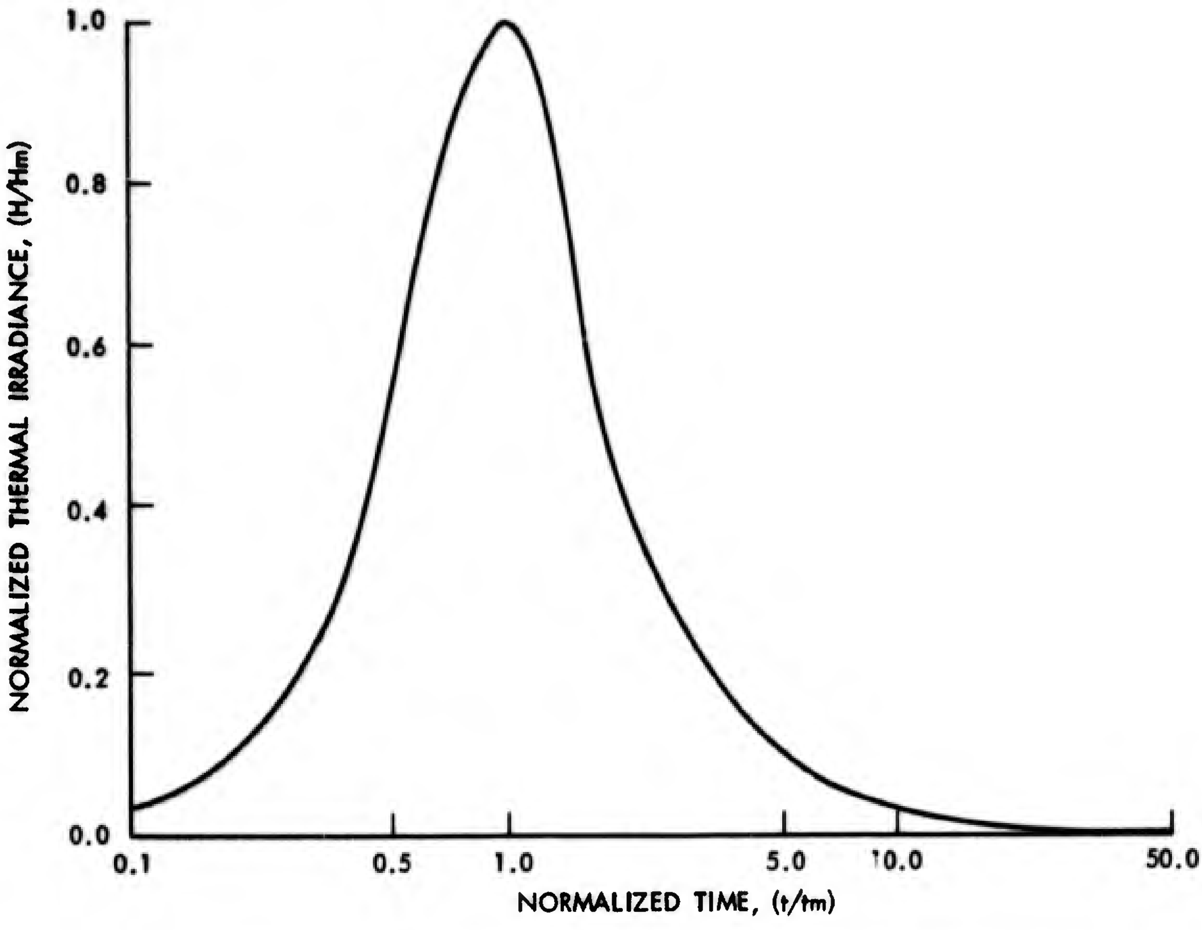

FIG. 1 NORMALIZED NUCLEAR WEAPON THERMAL RADIATION PULSE

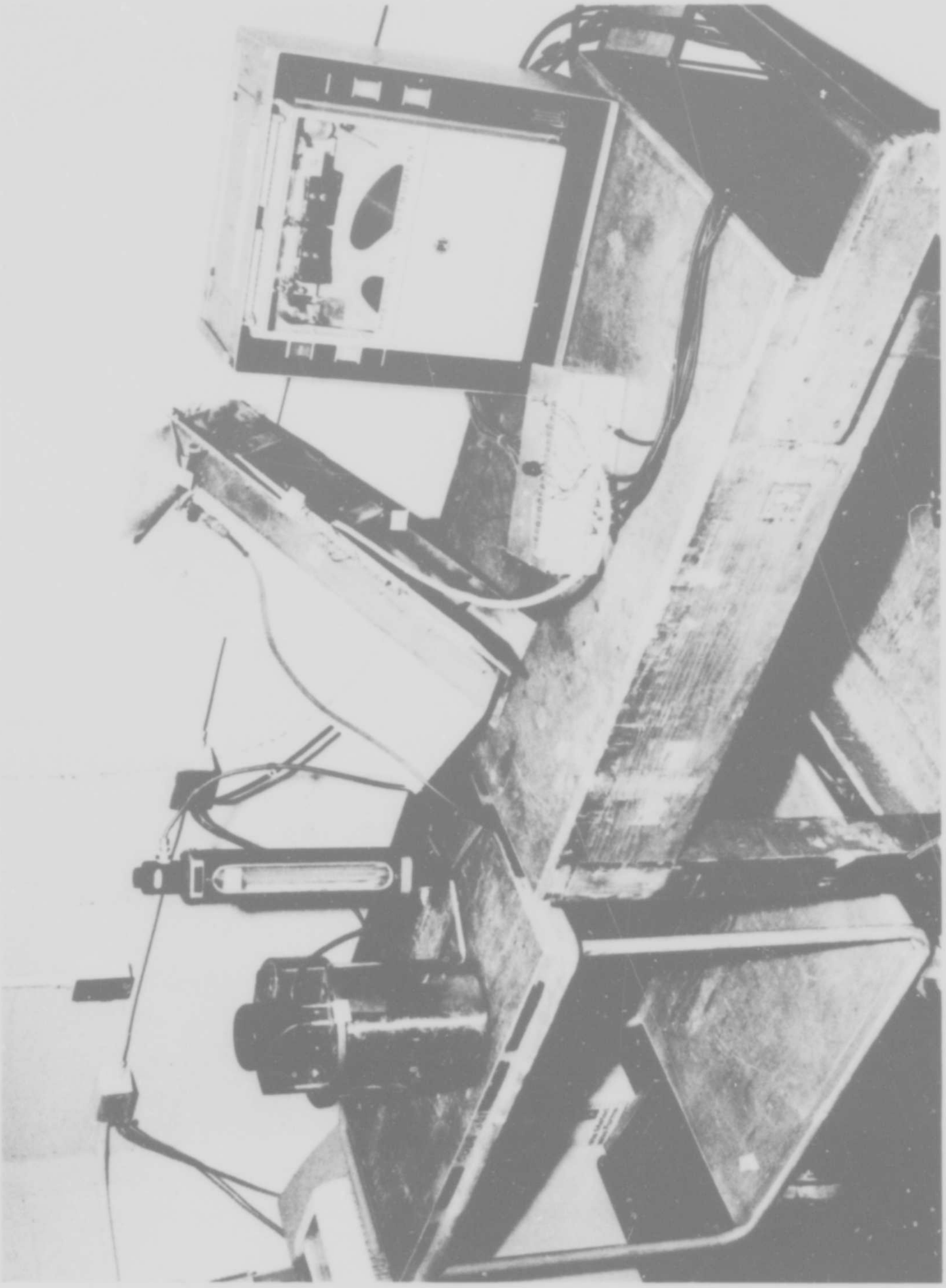

FIG.2 EXPERIMENTAL EQUIPMENT FOR WASHDOWN HEAT TRANSFER TESTS

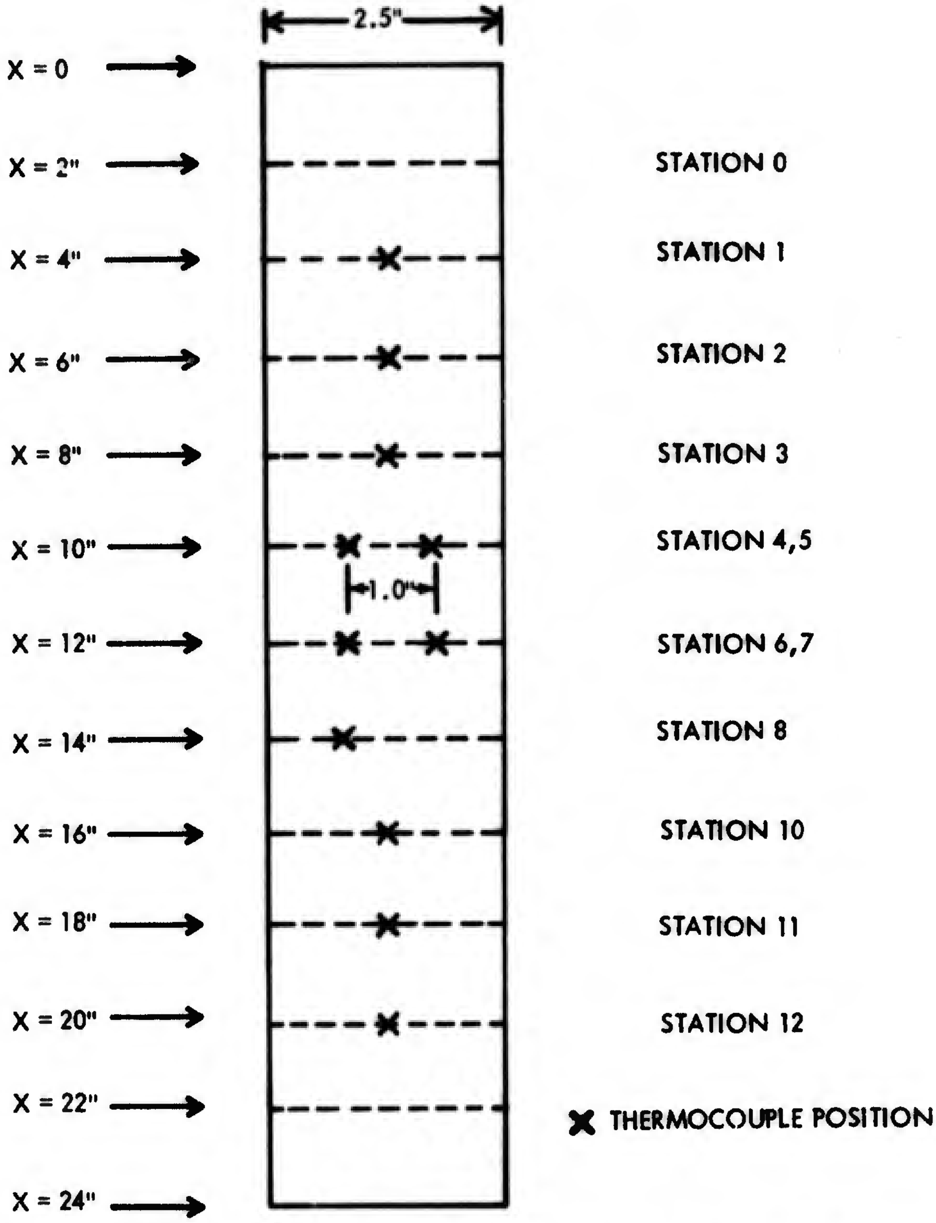

FIG. 3 LOCATION OF THERMOCOUPLE STATIONS FOR WASHDOWN HEAT TRANSFER TESTS

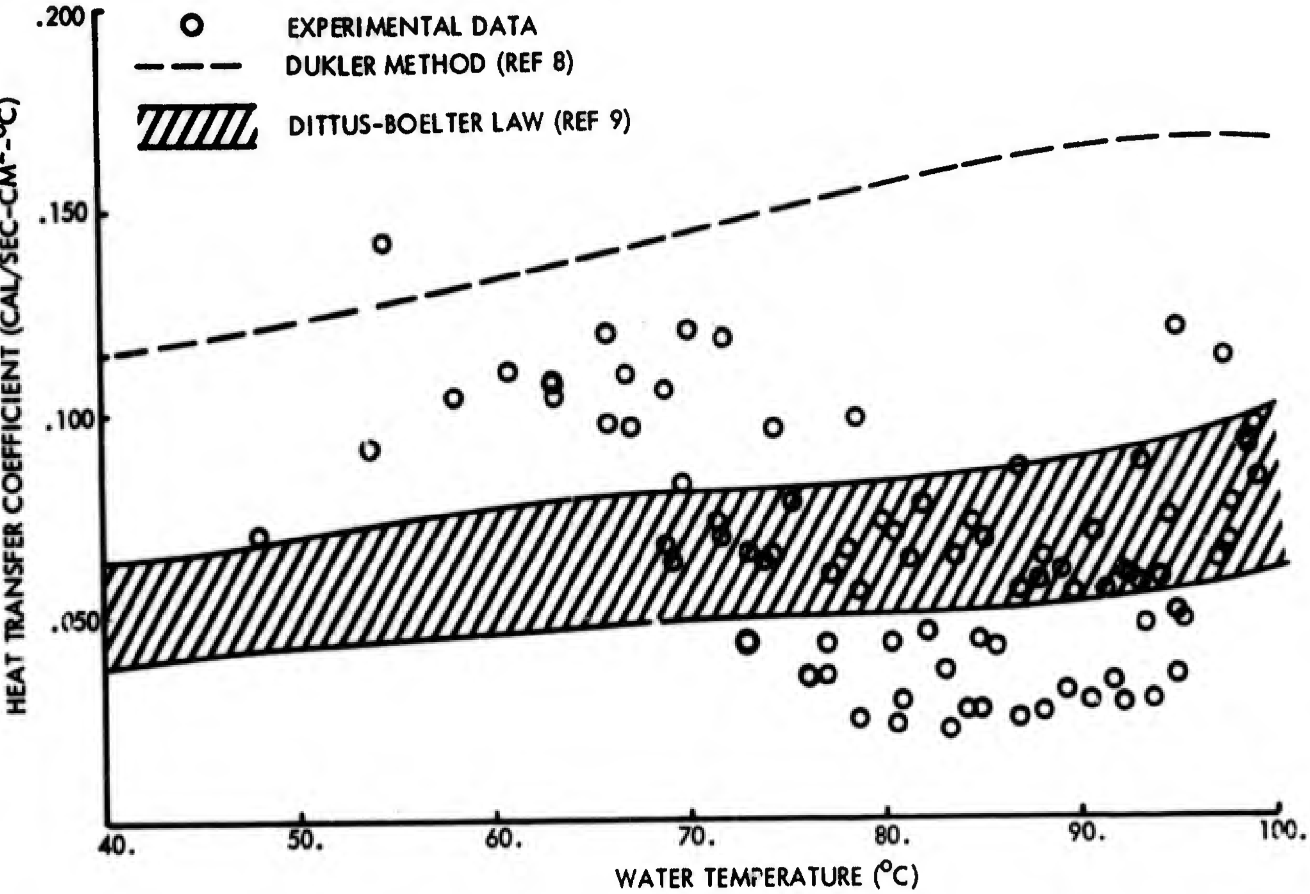

FIG. 4 FORCED CONVECTIVE COEFFICIENTS FOR WASHDOWN COOLING

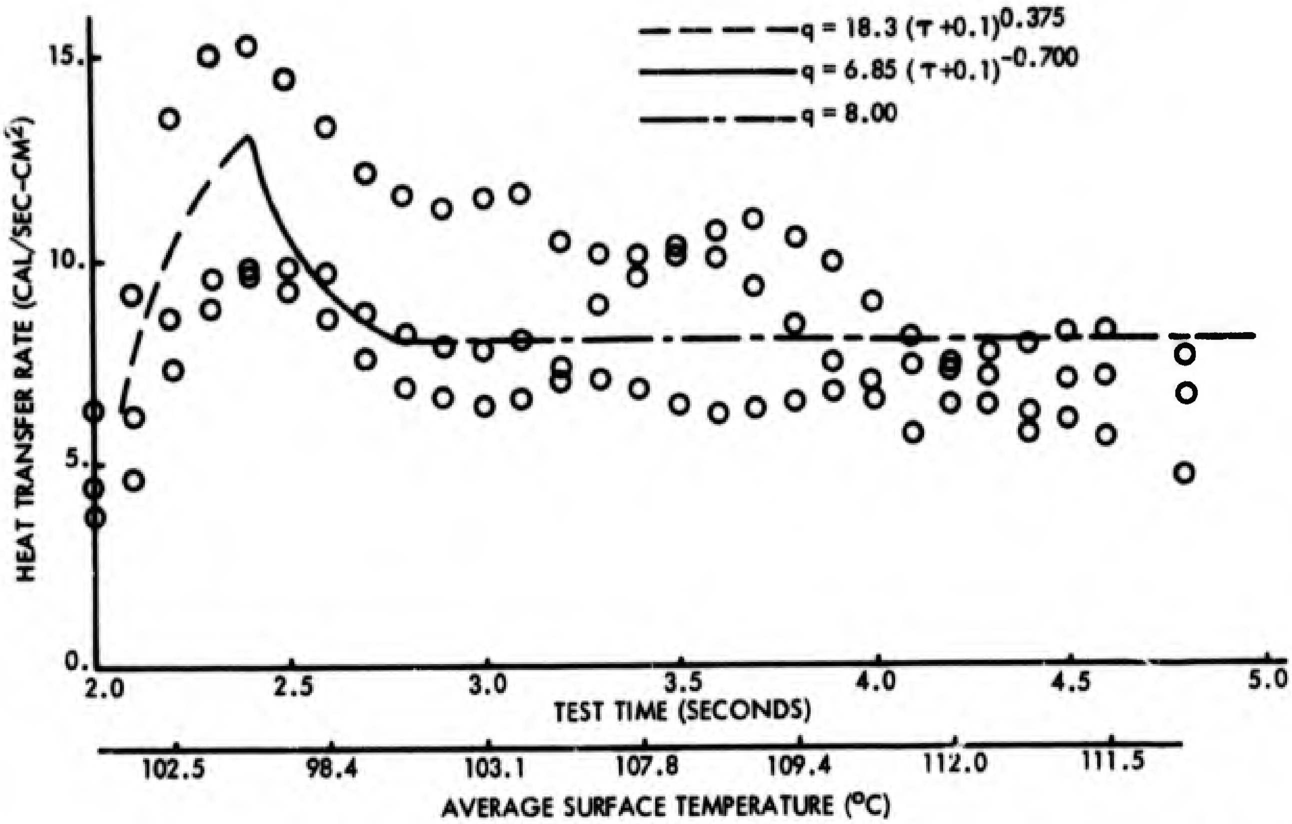

FIG. 5 BOILING REGIONS OF WASHDOWN HEAT TRANSFER TESTS (UPPER 3 THERMOCOUPLES)

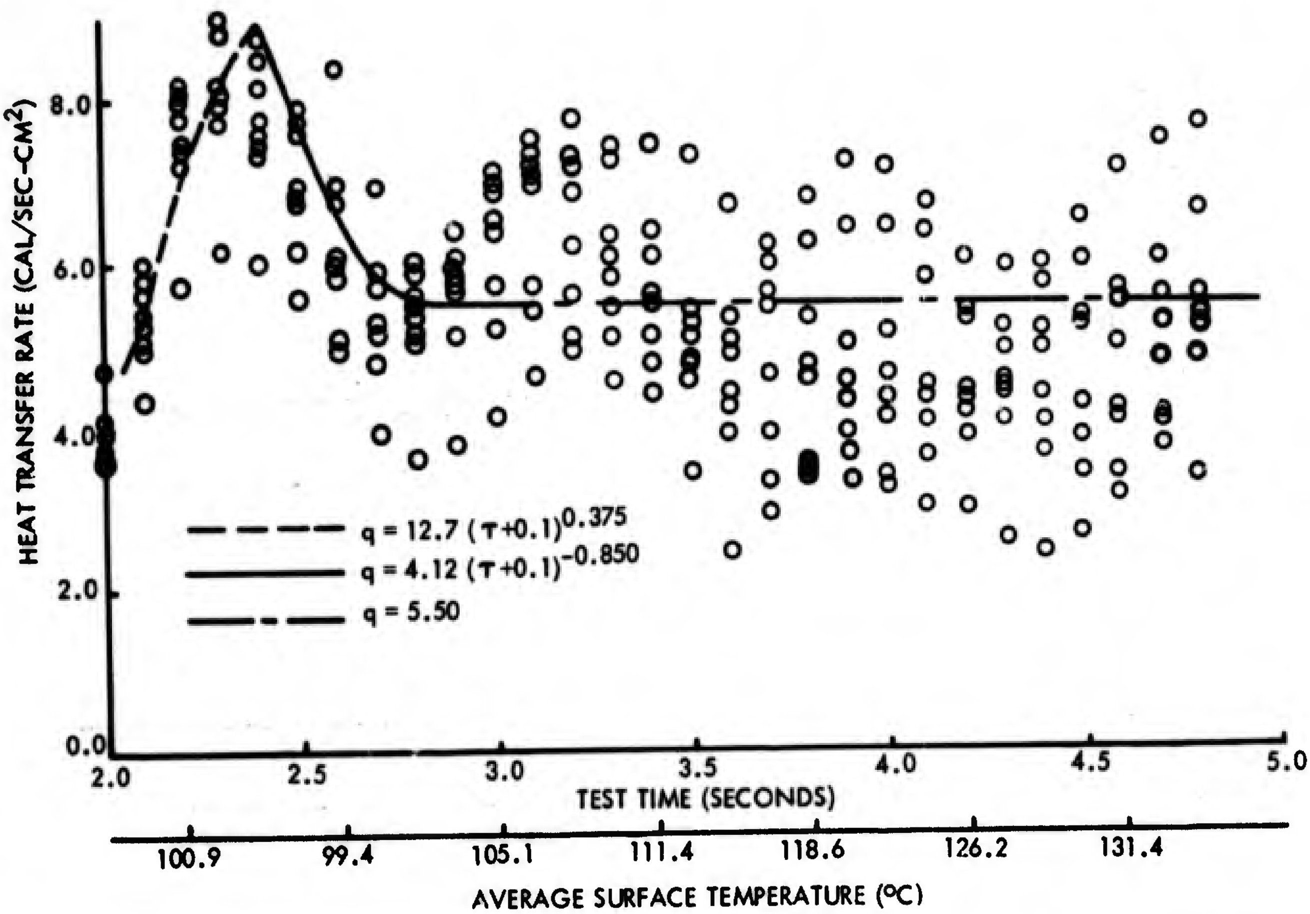

FIG. 6 BOILING REGIONS OF WASHDOWN HEAT TRANSFER TESTS (LOWER 8 THERMOCOUPLES)

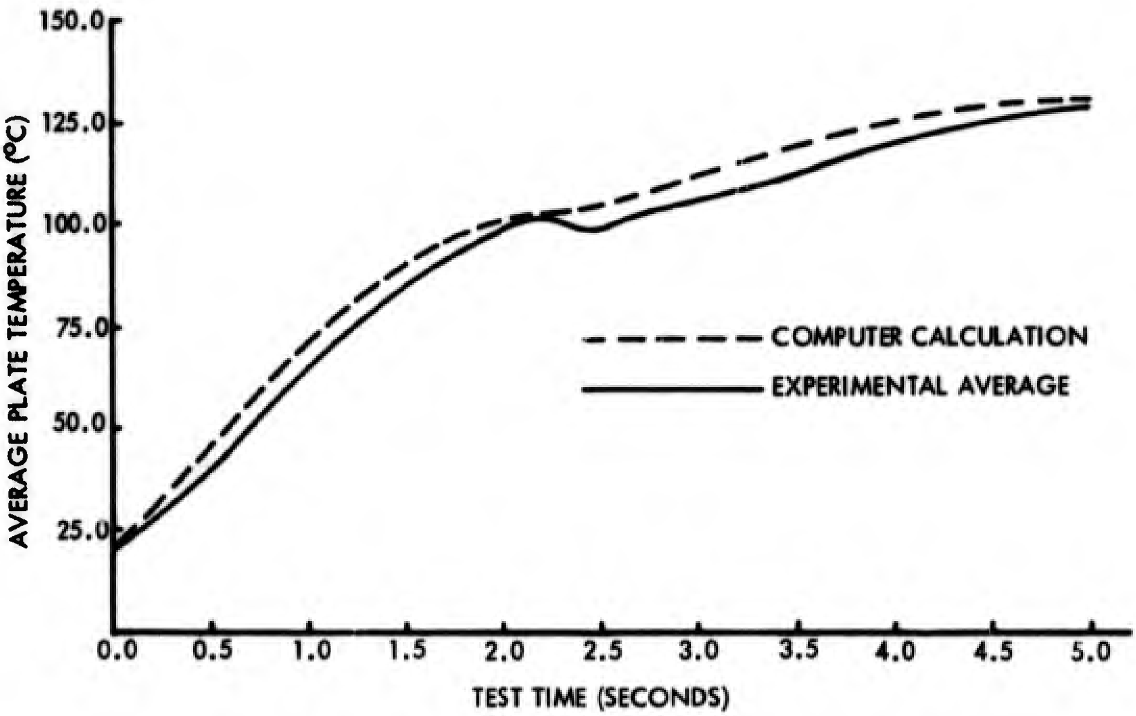

FIG. 7 EXPERIMENTAL AVERAGE PLATE TEMPERATURE COMPARED WITH COMPUTER CALCULATION FOR WASHDOWN HEAT TRANSFER TESTS

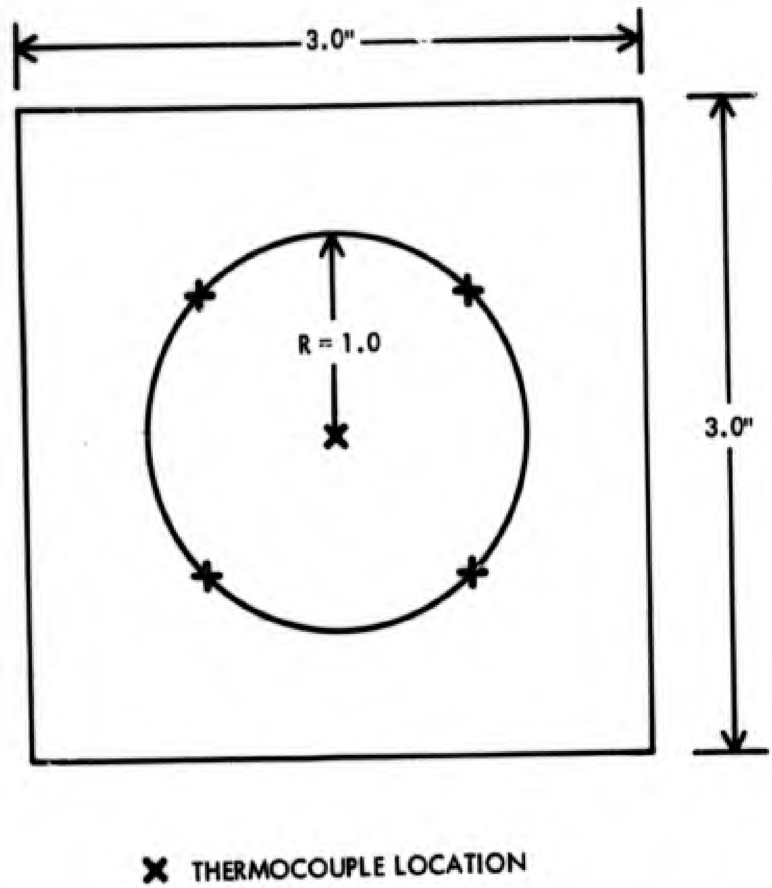

FIG. 8 THERMOCOUPLE LOCATIONS ON SPRAY COOLING TEST MODELS

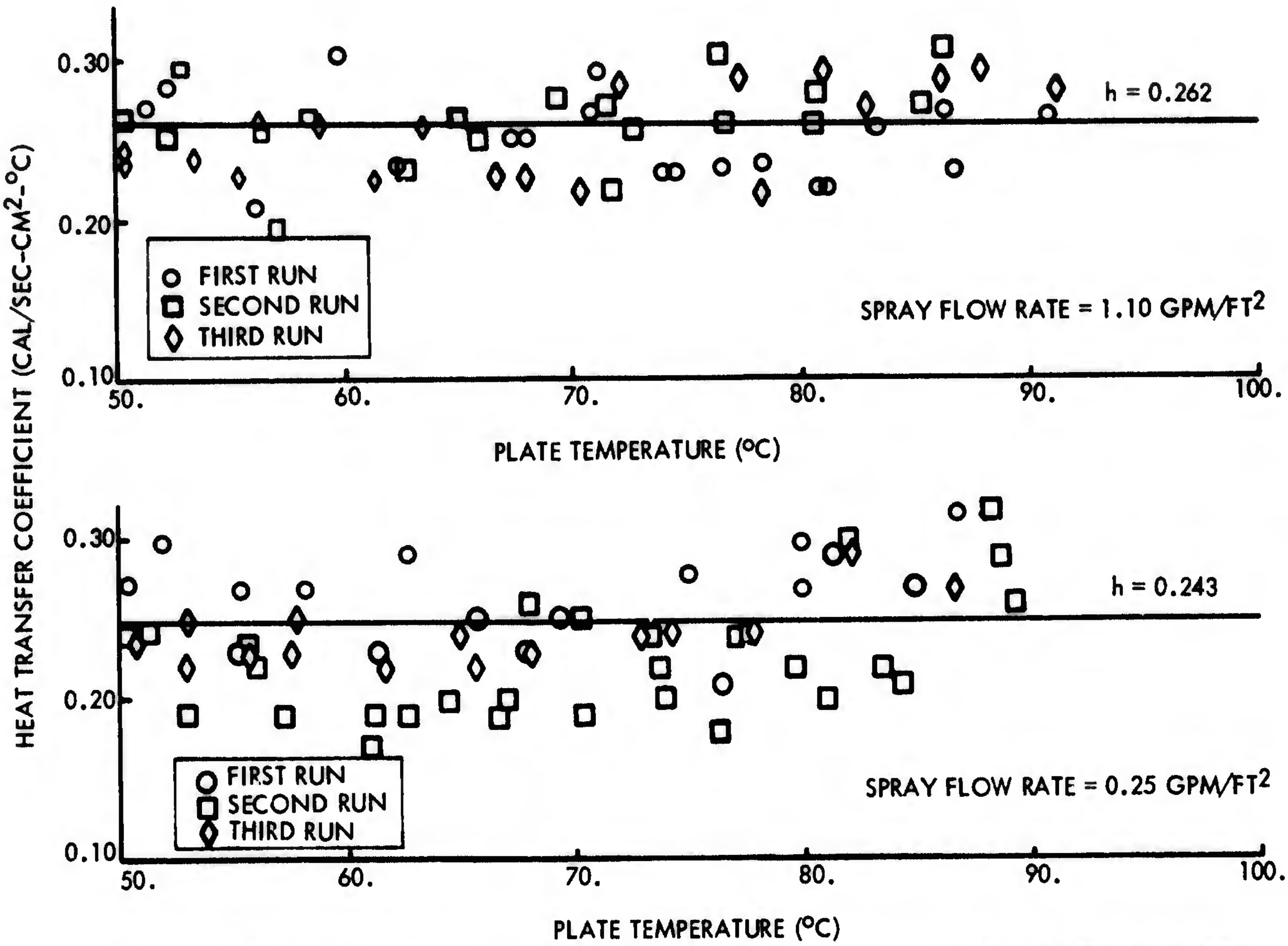

FIG. 7 FORCED CONVECTIVE COEFFICIENTS FOR SPRAY COOLING (UPPER THERMOCOUPLES)

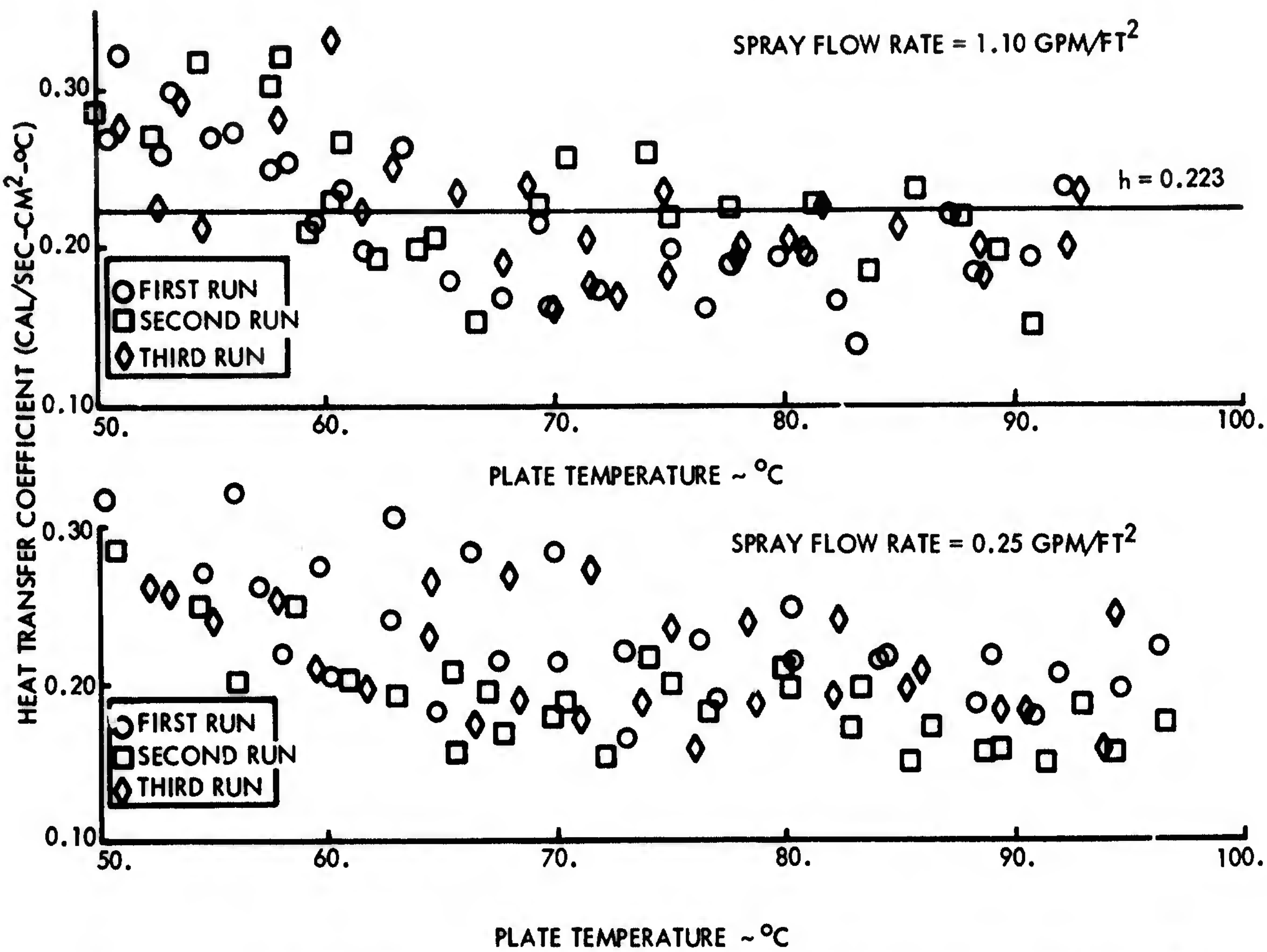

FIG. 10 FORCED CONVECTIVE COEFFICIENTS FOR SPRAY COOLING (LOWER THERMOCOUPLES)

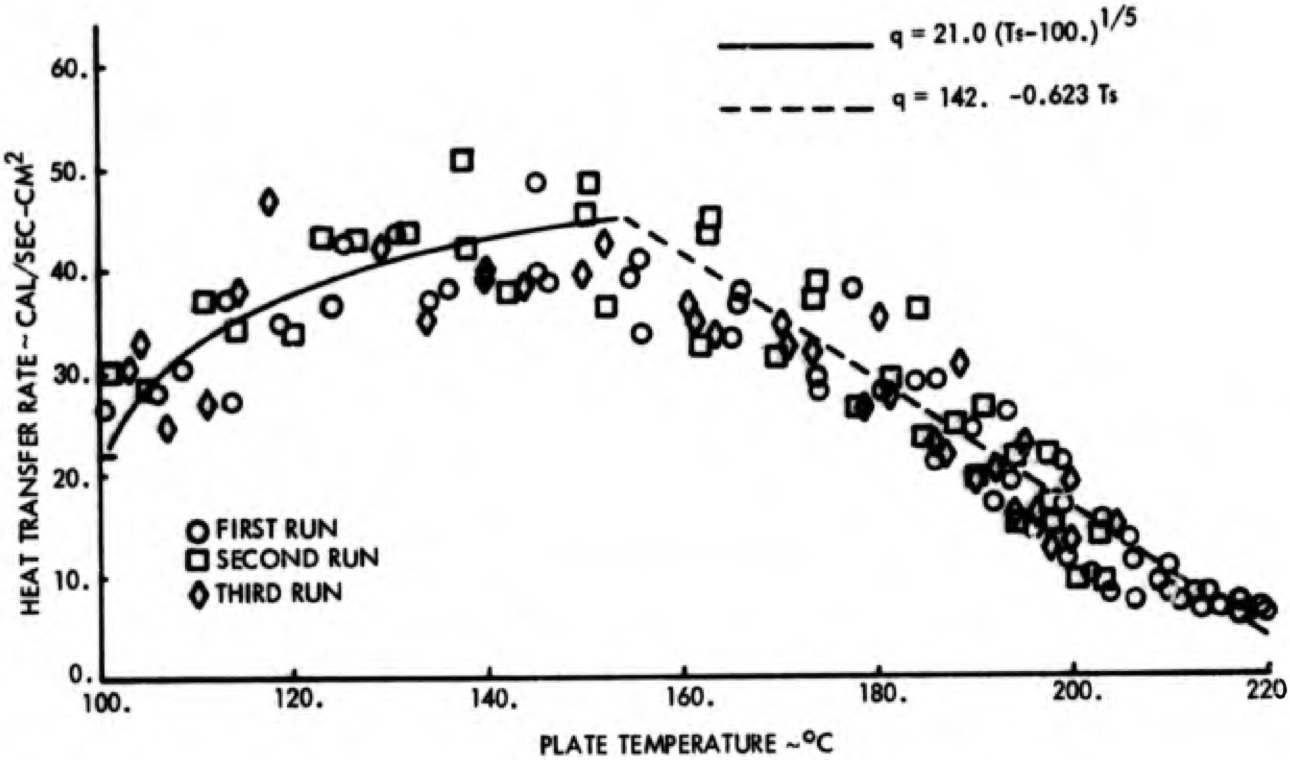

FIG. 11 BOILING REGIONS OF SPRAY HEAT TRANSFER TESTS FOR SPRAY FLOW RATE OF 1.10GPM/FT^2 (UPPER THERMOCOUPLES)

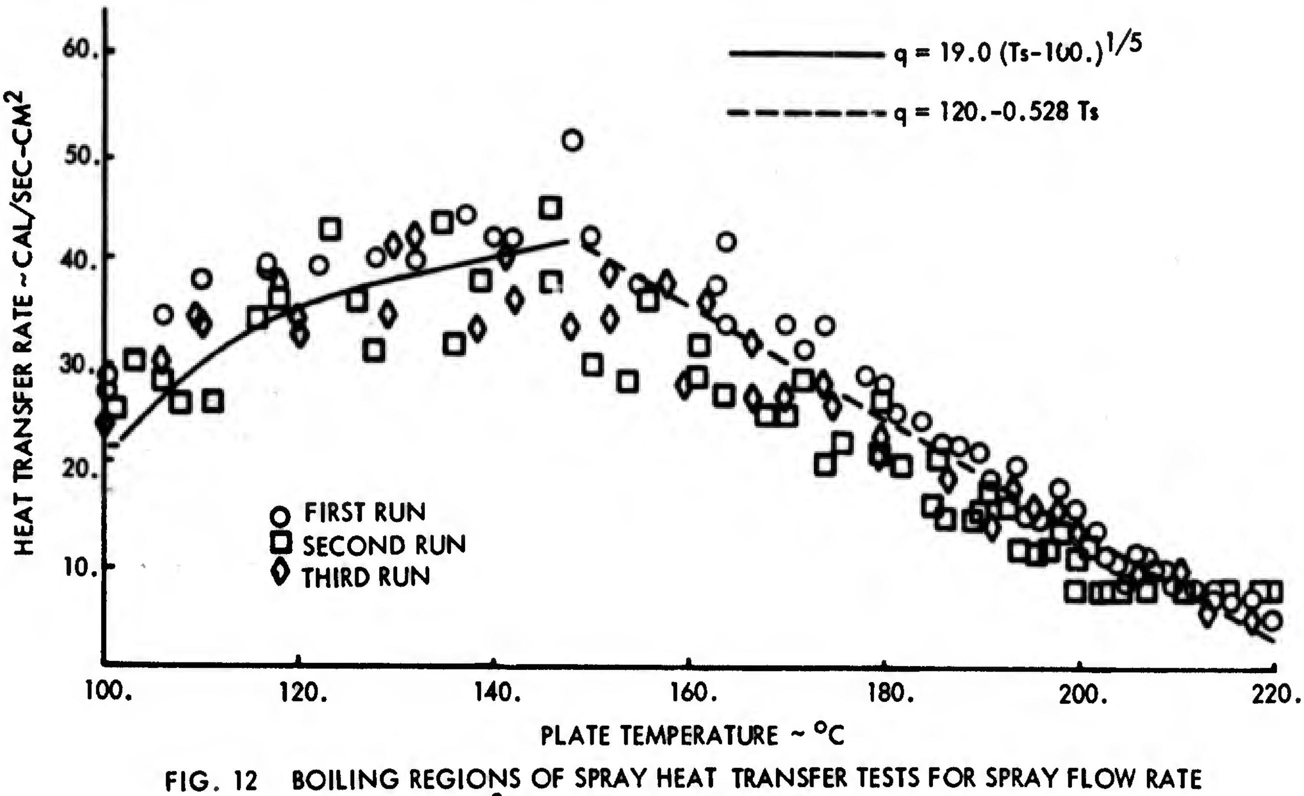

FIG. 12 BOILING REGIONS OF SPRAY HEAT TRANSFER TESTS FOR SPRAY FLOW RATE OF 0.25 GPM/FT^2 (UPPER THERMOCOUPLES)

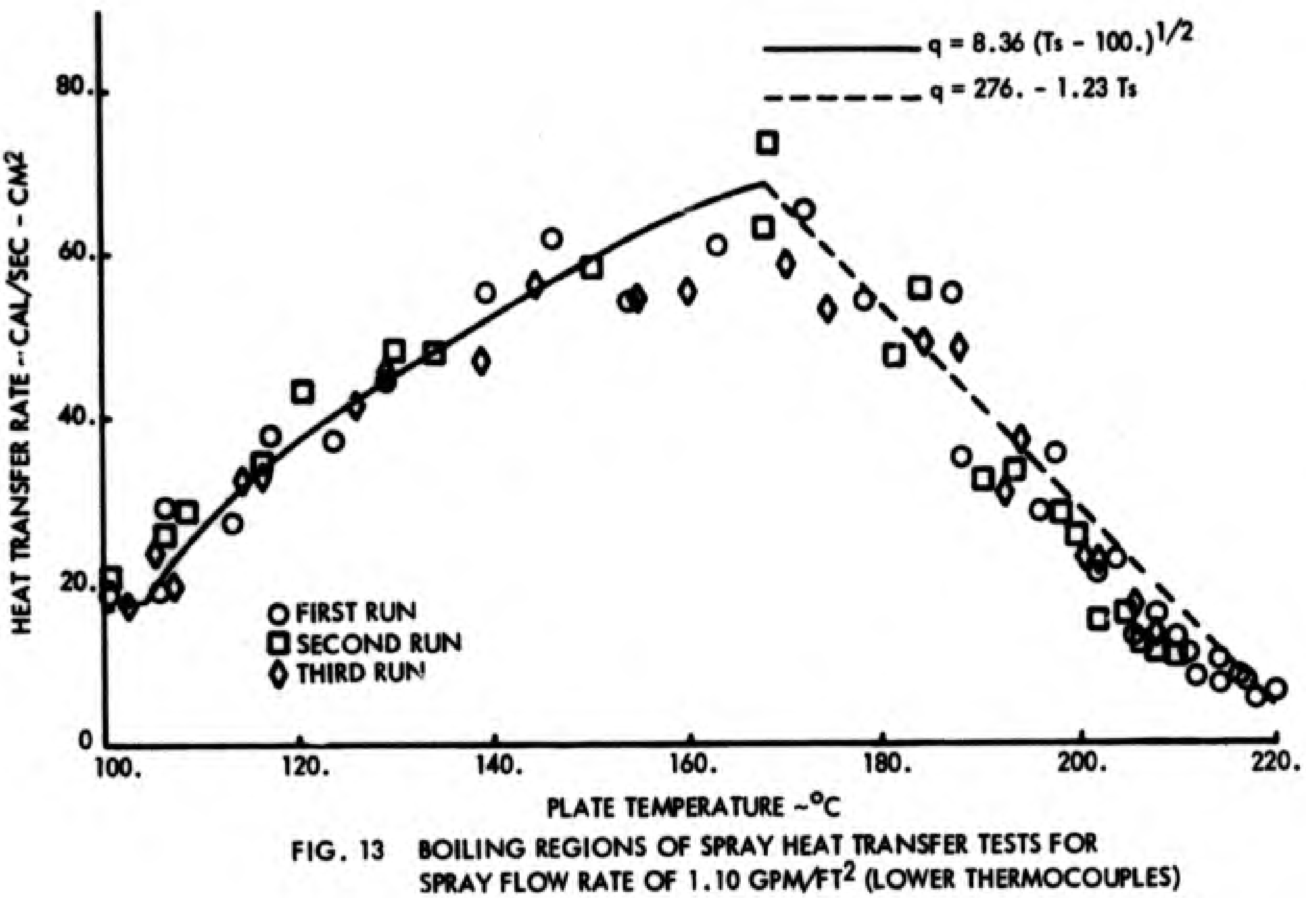

FIG. 13 BOILING REGIONS OF SPRAY HEAT TRANSFER TESTS FOR SPRAY FLOW RATE OF 1.10 GPM/FT^2 (LOWER THERMOCOUPLES)

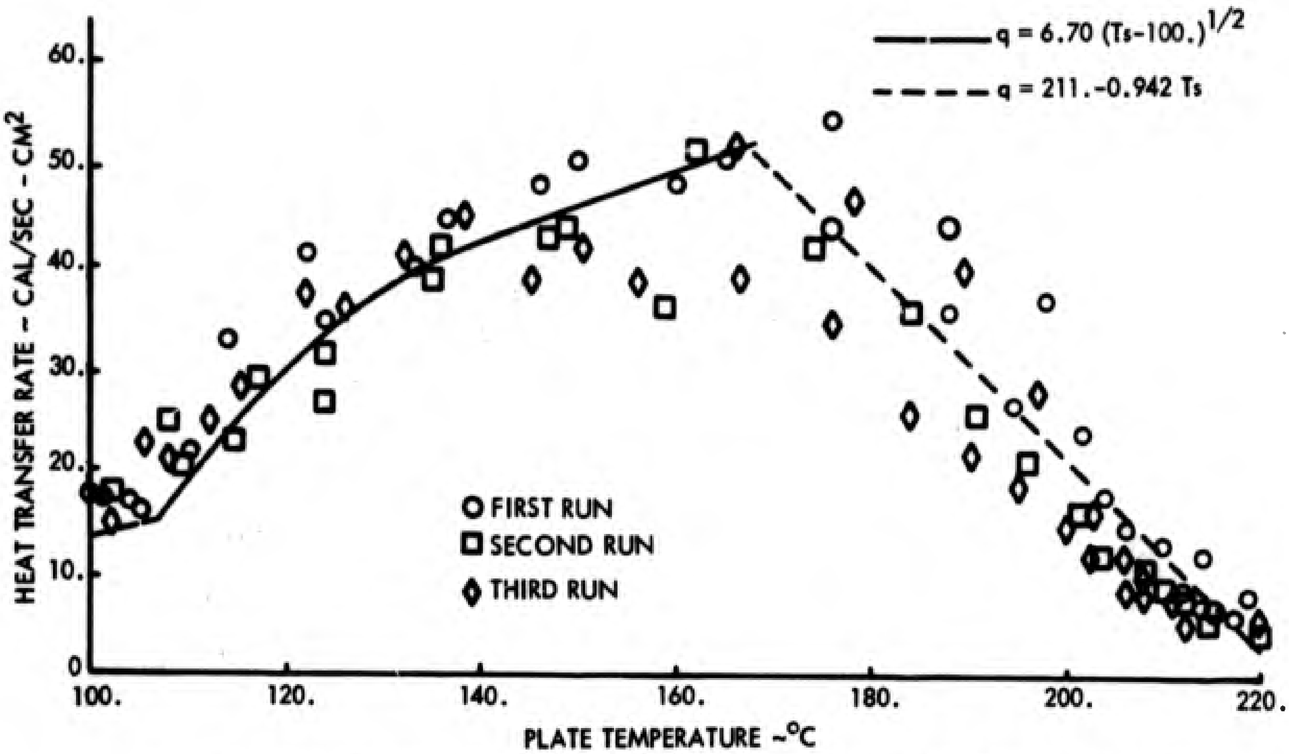

FIG. 14 BOILING REGIONS OF SPRAY HEAT TRANSFER TESTS FOR SPRAY FLOW RATE OF 0.25 GPM/FT^2 (LOWER THERMOCOUPLES)

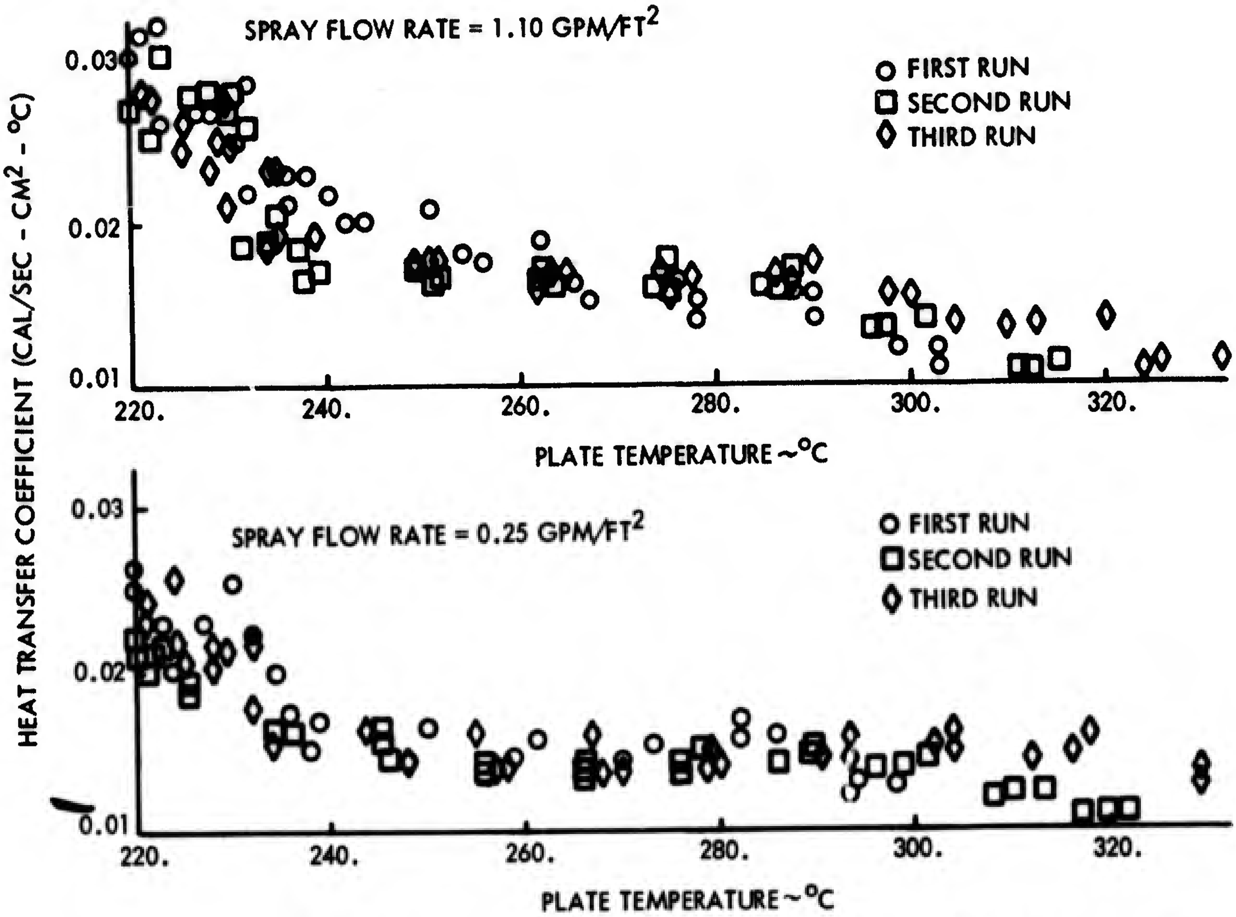

FIG. 15 FILM BOILING COEFFICIENTS FOR SPRAY COOLING (UPPER THERMOCOUPLES)

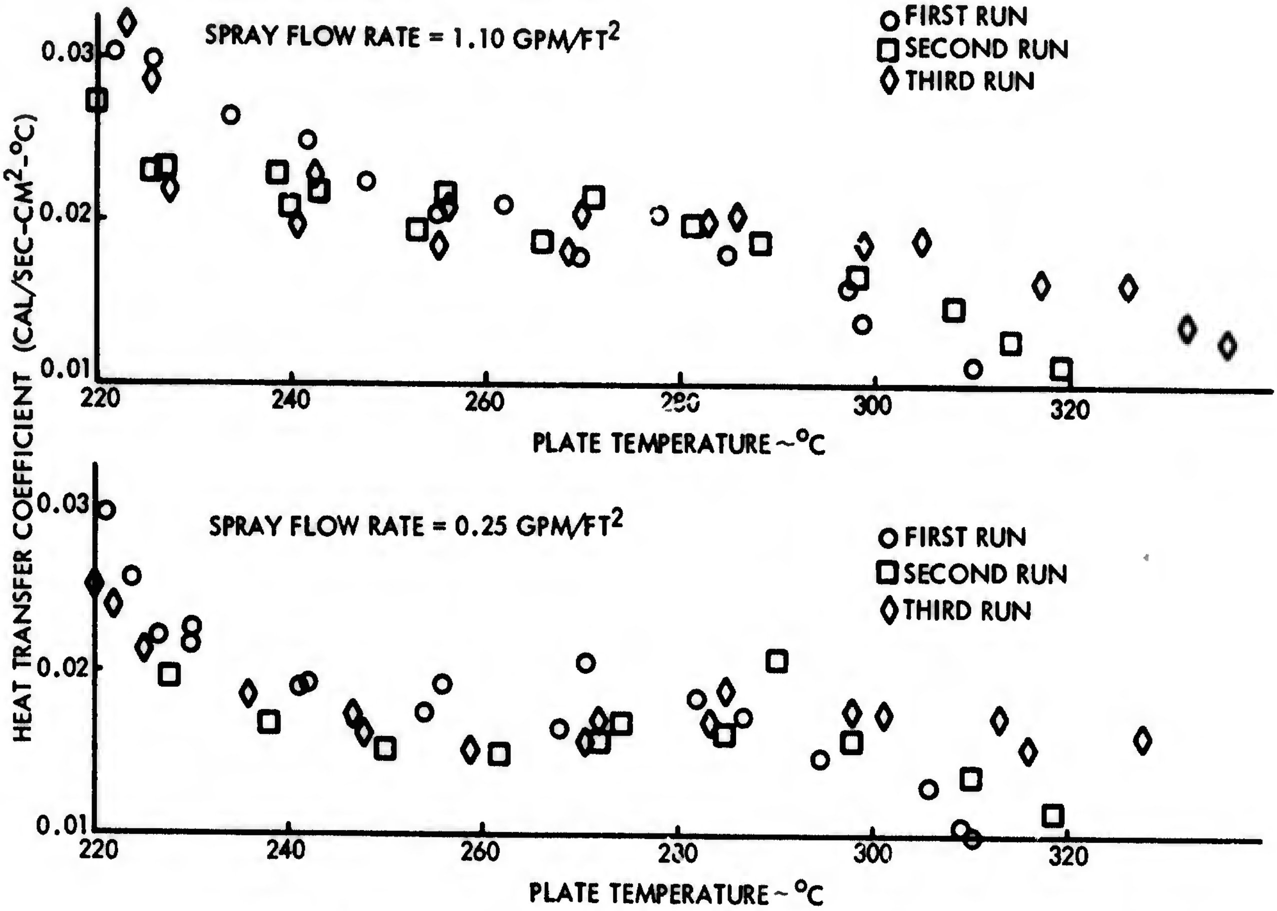

FIG. 16 FILM BOILING COEFFICIENTS FOR SPRAY COOLING (LOWER THERMOCOUPLES)

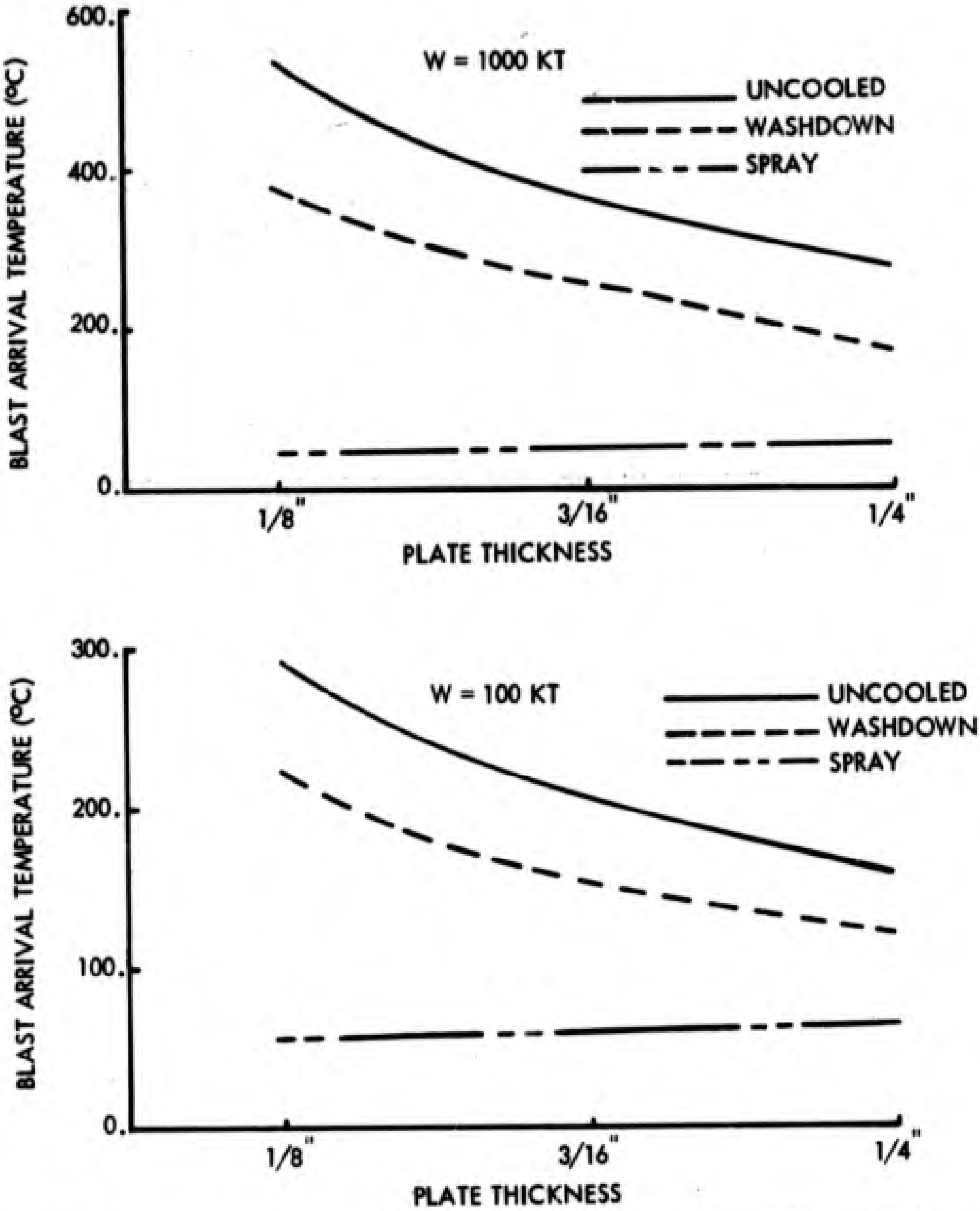

FIG. 17 COMPARISON OF COOLING SYSTEM EFFECTIVENESS AT 8 PSI OVERPRESSURE RANGE

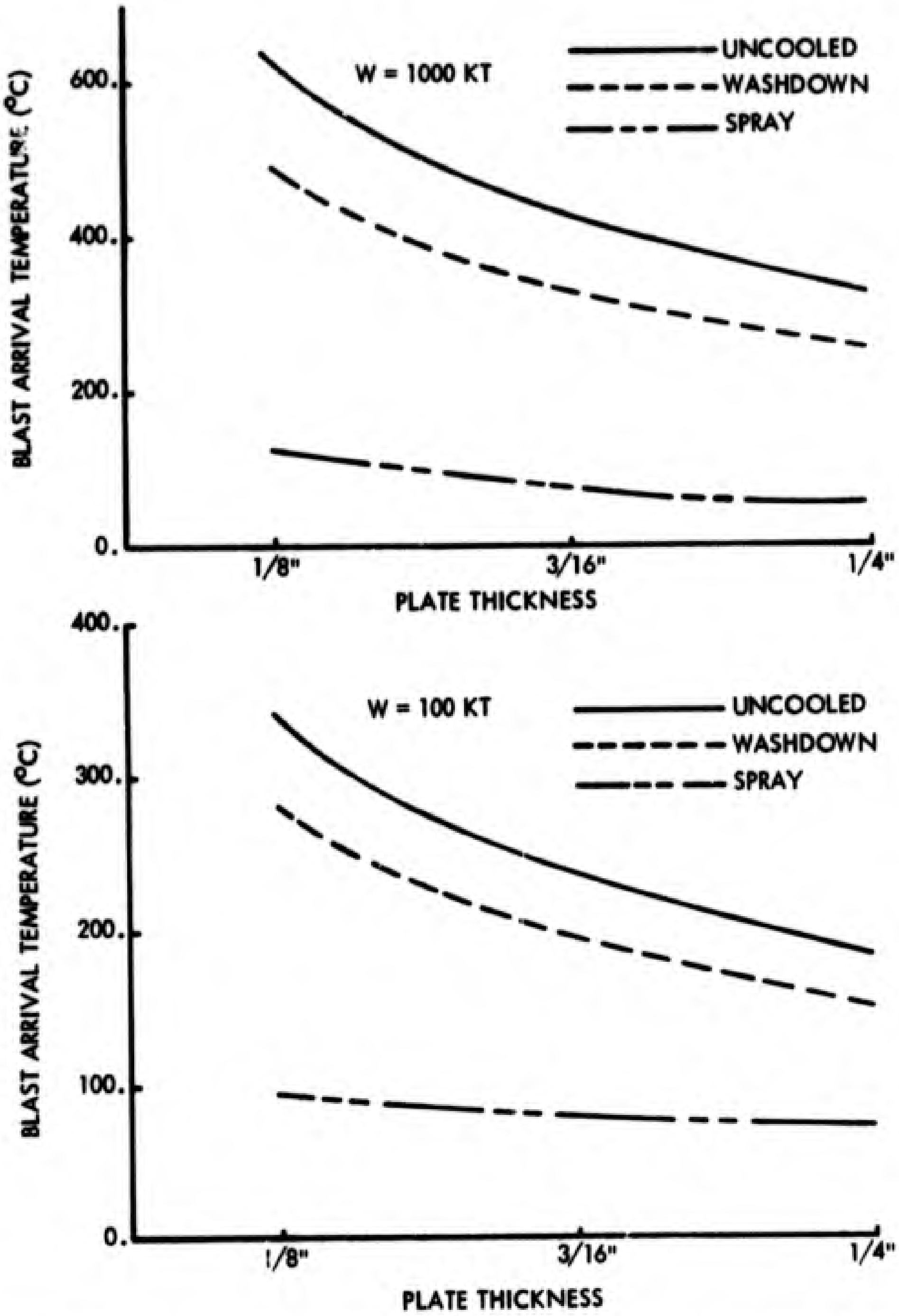

FIG. 18 COMPARISON OF COOLING SYSTEM EFFECTIVENESS AT 10 PSI OVERPRESSURE RANGE

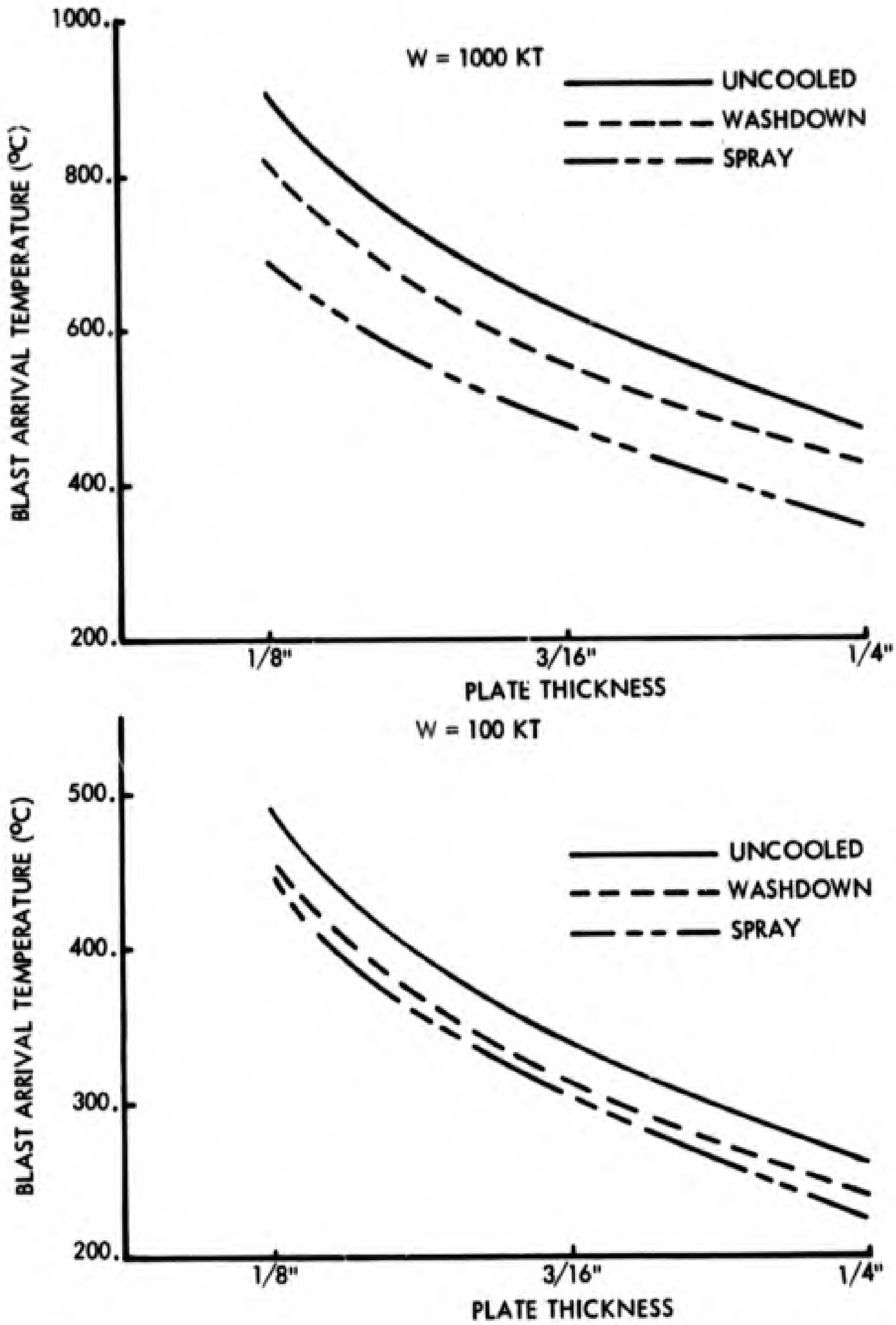

FIG. 19 COMPARISON OF COOLING SYSTEM EFFECTIVENESS AT 15 PSI OVERPRESSURE RANGE

TABLE 1 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 1 (X = 4 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	22.05	25.64	7.179	17.95	25.47	1.556
.1	27.58	25.80	9.596	20.17	11.84	4.960
.2	32.87	27.79	11.383	24.72	2.92	7.695
.3	38.03	29.55	12.588	28.53	12.69	9.739
.4	46.89	26.88	13.263	32.84	37.94	11.067
.5	55.26	23.36	13.455	40.53	17.93	11.655
.6	62.13	15.98	13.214	48.53	20.13	11.478
.7	68.43	12.52	12.433	53.86	23.11	9.885
.8	75.81	10.47	11.511	58.16	21.29	8.460
.9	82.16	9.03	10.546	63.39	19.88	7.075
1.0	87.54	8.40	9.640	67.08	20.45	5.886
1.1	92.52	7.47	8.899	69.13	17.69	5.049
1.2	97.33	6.06	8.499	71.59	16.72	5.527
1.3	101.74	6.07	8.383	74.50	18.13	5.437
1.4	106.25	5.62	8.461	78.66	15.95	5.281
1.5	111.27	5.62	8.628	81.43	15.26	5.120
1.6	116.19	6.20	8.780	84.71	15.74	5.016
1.7	121.37	6.60	8.687	86.45	14.09	5.428
1.8	125.62	7.12	8.486	89.32	13.15	5.305
1.9	130.44	7.18	8.250	91.47	13.20	4.909
2.0	135.26	7.32	8.052	95.68	14.16	4.230
2.1	139.67	6.34	7.969	97.93	14.04	3.252
2.2	143.76	6.50	8.182	100.19	13.06	.809
2.3	148.38	6.61	8.534	99.88	11.97	-.201
2.4	152.99	6.29	8.979	97.93	10.03	-.540
2.5	158.83	5.97	9.455	97.32	7.98	-.349
2.6	164.06	6.02	9.899	98.14	6.25	.227
2.7	169.29	6.25	10.153	98.55	6.91	1.442
2.8	175.44	6.52	10.353	99.67	6.57	2.188
2.9	181.28	6.38	10.497	101.11	6.75	2.655
3.0	186.40	6.20	10.598	102.08	6.31	2.852
3.1	192.56	6.50	10.669	104.08	7.20	2.789
3.2	199.22	7.00	10.591	105.93	6.86	2.347
3.3	204.14	6.25	10.746	106.75	6.40	1.889
3.4	210.18	5.71	10.962	108.08	7.01	1.405
3.5	217.05	5.06	11.175	107.98	6.43	.959
3.6	223.61	4.56	11.324	108.59	5.79	.616
3.7	228.36	4.58	11.674	109.10	5.80	.646
3.8	235.49	4.26	11.280	109.51	6.52	.647
3.9	242.51	4.16	10.527	110.03	6.79	.642
4.0	247.67	4.55	9.559	109.47	5.62	.594
4.1	252.71	4.55	8.519	109.27	4.74	.462
4.2	257.13	4.53	7.421	111.29	3.75	.053
4.3	261.04	4.61	6.741	111.33	4.15	-.304
4.4	264.23	4.54	6.342	109.74	4.40	-.623
4.5	267.30	4.38	6.156	109.10	3.89	-.851
4.6	271.13	4.29	6.116	109.37	3.46	-.934
4.7	275.21	4.74	6.155	108.79	5.36	-.819
4.8	278.61	5.10	6.206	108.67	6.33	-.452
4.9	281.13	4.65	6.201	108.19	5.89	.220

TABLE 2 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 2 (X = 6 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²
0.0	20.54	36.29	1.026	17.71	21.74	4.134
.1	23.82	29.73	7.525	23.12	31.51	7.929
.2	28.37	30.69	11.882	27.57	32.61	10.451
.3	33.30	26.79	14.410	33.37	41.60	11.854
.4	44.70	18.49	15.425	38.78	56.61	12.296
.5	55.14	20.66	15.239	45.87	27.41	11.933
.6	62.88	16.82	14.166	54.71	29.20	10.920
.7	68.87	13.38	11.737	60.95	27.09	8.519
.8	75.25	15.04	10.316	63.41	19.39	7.259
.9	80.28	13.68	9.336	66.02	13.26	6.471
1.0	85.11	13.27	8.664	69.65	13.44	6.044
1.1	90.14	13.29	8.168	73.27	12.69	5.868
1.2	94.20	13.40	7.561	77.19	11.59	6.166
1.3	98.84	15.15	7.035	80.81	13.26	5.958
1.4	102.51	16.29	6.663	83.42	12.41	5.595
1.5	105.80	17.69	6.511	87.05	11.33	5.233
1.6	109.86	20.65	6.649	89.37	12.47	5.026
1.7	112.95	24.24	7.437	91.98	11.68	6.511
1.8	117.01	27.84	8.242	95.02	10.35	6.220
1.9	122.42	28.20	9.115	97.78	11.72	5.184
2.0	128.61	26.95	10.003	103.00	14.23	3.609
2.1	133.25	22.27	10.854	104.74	14.71	1.702
2.2	139.44	15.90	11.741	105.75	14.67	-1.878
2.3	146.78	13.88	12.201	102.56	12.46	-2.929
2.4	153.16	12.82	12.456	98.94	10.38	-2.798
2.5	161.09	13.45	12.581	98.21	8.23	-1.897
2.6	167.86	16.01	12.657	97.92	8.52	-.639
2.7	174.43	18.57	12.864	98.65	8.34	.719
2.8	181.78	21.15	12.970	99.08	8.57	1.294
2.9	188.93	25.59	13.008	100.39	8.50	1.420
3.0	195.12	29.24	12.920	100.10	8.35	1.262
3.1	203.24	31.35	12.651	101.69	10.70	.980
3.2	211.94	32.46	11.331	101.26	8.66	.916
3.3	216.77	35.35	11.024	101.69	9.33	.862
3.4	222.18	36.21	11.020	103.00	6.72	.858
3.5	227.21	35.77	11.115	103.14	7.02	.908
3.6	231.85	34.56	11.103	103.29	7.65	1.016
3.7	242.65	135.53	10.860	104.16	8.64	1.555
3.8	247.00	127.90	9.961	104.45	9.25	1.561
3.9	251.93	122.32	8.744	105.17	7.18	1.344
4.0	256.27	104.64	7.466	107.93	4.61	.989
4.1	259.42	106.98	6.382	107.77	3.93	.579
4.2	262.87	104.05	6.378	108.21	3.32	.038
4.3	267.52	98.07	6.245	105.73	5.99	-.138
4.4	270.42	97.34	6.131	108.01	3.74	-.130
4.5	272.92	95.47	5.969	106.10	6.78	-.030
4.6	275.82	90.26	5.691	108.08	3.91	.067
4.7	279.82	87.69	5.230	107.58	4.50	.068
4.8	284.92	72.02	4.519	106.73	7.02	-.121
4.9	283.52	78.67	3.489	107.05	7.50	-.592

TABLE 3 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 3 (X = 8 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	22.02	19.05	-2.360	18.79	32.01	2.855
.1	24.32	17.45	2.657	23.05	24.30	5.771
.2	25.98	11.78	6.774	26.61	23.28	7.911
.3	29.72	13.21	10.000	30.06	25.74	9.338
.4	36.05	18.86	12.349	35.12	38.11	10.114
.5	43.63	21.66	13.829	41.44	18.57	10.302
.6	52.75	21.21	14.454	48.34	25.05	9.965
.7	60.51	21.70	13.746	53.70	25.45	8.630
.8	67.98	20.54	12.968	57.92	22.69	7.757
.9	74.21	18.53	12.010	60.89	19.71	7.053
1.0	80.63	16.14	10.984	64.72	19.71	6.525
1.1	86.67	14.69	10.003	68.75	19.06	6.178
1.2	91.85	12.49	9.427	72.10	18.66	6.311
1.3	97.12	10.36	8.817	75.65	17.77	6.192
1.4	101.91	9.18	8.295	78.81	18.06	6.015
1.5	106.03	8.19	7.889	82.36	17.06	5.828
1.6	110.25	7.82	7.623	85.81	16.00	5.677
1.7	114.75	8.07	7.582	88.11	16.19	6.411
1.8	119.06	8.81	7.660	91.84	15.17	5.919
1.9	122.90	9.29	7.833	94.05	14.35	4.920
2.0	127.50	8.82	8.067	98.74	14.86	3.604
2.1	132.00	7.91	8.330	100.18	14.79	2.164
2.2	136.89	7.06	8.550	101.81	13.14	-.012
2.3	141.58	6.21	8.808	99.70	10.90	-.696
2.4	146.38	5.27	9.061	98.36	9.70	-.592
2.5	152.13	4.83	9.301	98.55	8.00	.064
2.6	156.82	4.57	9.520	99.51	8.36	1.037
2.7	161.90	4.55	9.673	100.85	6.93	2.211
2.8	168.13	4.67	9.880	102.10	6.33	3.008
2.9	173.88	4.79	10.042	103.05	7.79	3.500
3.0	177.81	4.69	10.104	105.17	7.74	3.676
3.1	184.04	5.26	10.611	108.42	7.40	3.521
3.2	190.27	6.19	9.752	109.96	7.82	2.650
3.3	194.58	6.02	9.066	111.01	9.47	2.028
3.4	199.75	6.28	8.251	112.26	10.51	1.489
3.5	204.74	6.36	7.492	111.87	8.82	1.075
3.6	209.82	6.57	6.978	112.83	8.93	.829
3.7	211.18	7.55	7.386	113.50	9.31	1.120
3.8	215.55	7.72	7.672	113.89	9.85	1.246
3.9	220.15	7.74	7.995	114.75	10.73	1.304
4.0	225.02	7.18	8.295	115.69	11.63	1.282
4.1	231.99	7.84	8.514	116.13	12.59	1.167
4.2	234.48	6.65	8.272	116.37	12.94	.849
4.3	238.82	6.59	8.241	117.83	13.72	.571
4.4	243.19	6.37	8.255	117.68	14.56	.317
4.5	248.40	5.73	8.268	117.37	15.59	.118
4.6	253.71	5.40	8.236	117.53	14.39	.007
4.7	258.09	4.89	8.113	116.93	12.80	.018
4.8	262.05	4.98	7.852	118.02	17.16	.183
4.9	266.09	4.44	7.408	117.77	17.35	.534

TABLE 4 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 4 (X = 10 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²
0.0	22.76	23.52	3.567	18.42	24.15	2.151
.1	27.33	24.89	5.874	21.39	17.01	4.768
.2	30.25	23.88	7.828	24.93	19.50	6.780
.3	33.61	23.60	9.414	28.24	19.53	8.232
.4	39.89	22.60	10.621	32.47	35.31	9.169
.5	47.09	19.25	11.434	37.45	15.00	9.637
.6	53.18	13.41	11.841	44.78	23.23	9.680
.7	59.56	14.13	11.710	50.01	22.93	8.938
.8	66.51	13.48	11.338	54.30	19.82	8.533
.9	72.31	12.14	10.757	57.91	17.20	8.192
1.0	77.88	10.80	10.031	62.67	17.87	7.894
1.1	83.64	10.22	9.225	67.14	18.27	7.617
1.2	88.11	9.02	8.255	71.71	18.37	7.518
1.3	92.77	8.10	7.599	76.04	18.60	7.106
1.4	97.06	8.03	7.160	79.23	18.44	6.598
1.5	100.48	6.69	6.926	83.13	18.45	6.102
1.6	103.91	5.65	6.888	86.27	17.97	5.727
1.7	108.05	5.64	7.319	89.41	17.33	6.526
1.8	112.38	5.70	7.486	92.36	17.20	6.108
1.9	116.47	6.00	7.563	95.12	16.93	5.199
2.0	121.04	5.64	7.571	99.31	18.00	3.948
2.1	124.94	5.28	7.533	101.69	17.52	2.502
2.2	129.03	5.63	7.423	103.69	16.20	-.046
2.3	133.31	5.35	7.357	101.12	12.67	-.841
2.4	137.22	5.44	7.359	99.40	9.44	-.786
2.5	141.69	5.88	7.452	99.59	7.34	-.149
2.6	145.59	6.39	7.663	100.74	6.24	.799
2.7	149.87	6.67	7.951	101.50	5.52	2.044
2.8	154.92	7.05	8.552	102.73	4.84	2.582
2.9	159.39	7.10	9.203	104.07	4.14	2.762
3.0	163.77	7.18	9.743	105.31	3.58	2.722
3.1	169.76	7.63	10.006	107.78	3.73	2.599
3.2	176.43	8.20	9.736	109.11	3.86	2.547
3.3	180.23	8.02	8.874	109.68	3.93	2.734
3.4	185.37	8.12	7.770	111.20	4.82	3.007
3.5	190.32	8.20	6.696	113.01	6.13	3.280
3.6	195.36	8.82	5.923	115.58	7.98	3.470
3.7	194.66	9.52	6.383	117.20	9.48	3.454
3.8	198.89	9.82	6.756	119.30	10.38	3.213
3.9	203.57	9.99	7.191	120.44	11.68	2.837
4.0	208.44	10.08	7.572	122.75	11.94	2.405
4.1	212.88	9.58	7.782	123.46	12.10	1.995
4.2	217.10	9.63	7.101	124.16	12.83	1.804
4.3	220.63	9.59	6.830	125.94	13.20	1.649
4.4	223.97	8.86	6.722	126.17	13.30	1.538
4.5	227.65	8.30	6.760	127.10	14.46	1.439
4.6	231.60	7.71	6.924	127.90	15.20	1.323
4.7	235.90	7.49	7.194	128.64	15.93	1.159
4.8	240.18	7.74	7.550	129.40	16.35	.917
4.9	243.82	7.49	7.973	129.49	16.41	.566

TABLE 5 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 5 (X = 10 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	25.50	19.09	1.263	18.12	34.38	2.228
.1	28.52	18.44	4.711	21.34	22.28	5.107
.2	31.64	17.13	7.608	24.25	25.20	7.237
.3	35.17	18.70	9.936	28.10	26.13	8.701
.4	42.04	17.53	11.675	33.20	37.22	9.583
.5	48.80	12.88	12.807	39.01	19.24	9.967
.6	56.90	8.66	13.313	45.51	26.20	9.935
.7	63.88	8.66	12.945	50.80	24.75	9.226
.8	71.07	7.68	12.256	55.13	21.22	8.845
.9	77.57	6.39	11.305	59.46	19.54	8.529
1.0	83.12	6.13	10.219	64.58	20.53	8.233
1.1	88.75	5.46	9.127	69.17	20.95	7.910
1.2	93.61	4.34	7.972	73.50	20.63	7.626
1.3	98.20	4.66	7.479	77.40	20.78	6.992
1.4	102.01	4.55	7.336	81.22	20.84	6.252
1.5	105.13	3.12	7.424	84.94	20.41	5.530
1.6	109.29	3.31	7.625	87.98	19.96	4.953
1.7	113.97	4.08	8.051	89.71	19.47	5.485
1.8	119.00	3.94	7.973	92.14	18.78	5.024
1.9	123.33	4.22	7.685	94.74	18.19	4.220
2.0	127.67	4.68	7.327	98.90	19.04	3.198
2.1	130.96	4.56	7.034	100.28	18.46	2.083
2.2	134.95	5.10	7.232	101.84	17.92	.044
2.3	138.85	5.29	7.356	99.85	14.24	-.342
2.4	143.09	5.36	7.530	98.90	12.59	.048
2.5	147.95	5.30	7.751	99.16	11.76	.911
2.6	152.19	5.65	8.018	100.98	11.61	1.945
2.7	156.35	6.09	8.018	102.36	11.44	2.866
2.8	161.29	6.23	8.574	104.10	11.31	3.243
2.9	165.97	6.38	9.273	105.92	11.04	3.278
3.0	170.48	6.53	9.929	107.17	11.27	3.084
3.1	176.55	7.26	10.353	109.82	11.47	2.775
3.2	182.96	8.11	10.480	110.42	11.70	2.662
3.3	187.55	8.07	9.668	111.81	11.96	2.371
3.4	193.36	8.46	8.451	113.28	12.12	2.080
3.5	198.82	8.82	7.144	114.50	12.80	1.869
3.6	203.76	8.62	6.063	115.88	13.83	1.816
3.7	203.03	8.75	5.941	116.40	14.35	2.054
3.8	207.09	8.74	6.147	117.36	15.21	2.640
3.9	211.35	9.17	6.617	118.40	16.13	3.288
4.0	215.50	9.82	7.173	122.20	16.21	3.800
4.1	220.11	9.60	7.634	123.34	17.22	3.979
4.2	224.19	9.56	7.325	125.38	17.57	3.173
4.3	227.83	9.72	7.200	127.62	17.66	2.296
4.4	231.99	8.94	7.112	129.43	17.95	1.330
4.5	235.27	8.52	7.068	128.86	18.23	.413
4.6	239.40	8.58	7.076	128.96	19.02	-.319
4.7	243.84	8.56	7.143	126.84	21.19	-.726
4.8	248.30	8.36	7.274	127.78	22.77	-.672
4.9	251.36	7.77	7.478	128.39	23.29	-.017

TABLE 6 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 6 (X = 12 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm²
0.0	22.28	15.10	1.321	17.13	14.48	-.490
.1	24.42	18.78	4.751	19.60	15.14	3.265
.2	27.00	19.62	7.461	22.51	15.30	6.241
.3	31.15	21.58	9.509	25.10	13.36	8.459
.4	38.55	24.17	10.953	29.81	34.96	9.943
.5	45.29	24.29	11.854	35.56	20.98	10.718
.6	51.28	20.40	12.269	43.23	30.66	10.805
.7	57.64	20.05	12.176	49.68	30.10	9.602
.8	64.84	17.97	11.824	53.98	26.18	8.768
.9	71.38	15.11	11.273	57.63	20.87	8.002
1.0	77.55	13.67	10.579	62.02	20.40	7.349
1.1	83.25	12.30	9.800	66.04	20.16	6.853
1.2	87.93	11.16	8.827	69.97	19.29	6.987
1.3	93.07	10.44	8.119	73.99	19.27	6.721
1.4	97.47	10.51	7.611	77.45	19.09	6.374
1.5	101.11	9.67	7.295	81.29	18.20	6.047
1.6	105.04	8.75	7.165	84.46	18.32	5.843
1.7	109.44	8.28	7.280	87.27	17.92	6.845
1.8	113.45	7.64	7.467	90.64	17.32	6.536
1.9	117.29	7.30	7.702	93.63	17.14	5.677
2.0	121.87	6.82	7.941	98.40	18.19	4.420
2.1	126.26	7.15	8.139	100.83	17.23	2.913
2.2	131.13	7.71	8.275	103.35	15.99	.067
2.3	135.43	7.87	8.198	100.27	11.99	-.787
2.4	139.91	8.31	8.037	98.86	9.06	-.709
2.5	144.96	8.38	7.870	99.05	7.69	-.016
2.6	148.80	9.01	7.774	100.17	6.58	.972
2.7	153.10	9.10	7.930	100.83	5.72	2.202
2.8	157.87	9.24	8.158	102.32	5.02	2.613
2.9	161.98	8.68	8.442	103.73	4.79	2.584
3.0	166.56	8.78	8.723	105.74	4.45	2.278
3.1	171.89	8.45	8.946	106.62	4.33	1.860
3.2	177.88	8.33	8.838	107.19	5.35	1.602
3.3	181.52	8.58	8.866	107.56	5.96	1.422
3.4	186.67	8.79	8.881	108.96	6.38	1.383
3.5	191.34	9.30	8.846	110.18	6.34	1.511
3.6	195.73	9.74	8.722	111.21	5.12	1.831
3.7	202.15	12.58	8.394	111.49	5.19	2.669
3.8	206.08	12.85	8.034	112.51	4.31	3.283
3.9	209.90	12.82	7.636	115.23	6.10	3.746
4.0	214.34	12.15	7.243	118.24	7.68	3.978
4.1	218.43	11.44	6.900	119.90	8.32	3.900
4.2	222.05	11.13	6.770	122.14	9.64	2.892
4.3	225.40	10.73	6.637	123.65	10.46	2.181
4.4	229.95	9.93	6.537	124.48	11.76	1.593
4.5	232.67	9.34	6.470	125.13	11.93	1.171
4.6	236.23	8.87	6.439	126.18	12.69	.955
4.7	240.19	8.89	6.443	125.16	11.93	.987
4.8	244.07	8.84	6.485	127.01	13.30	1.311
4.9	247.15	8.34	6.564	127.94	12.40	1.966

TABLE 7 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 7 (X = 12 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	22.14	37.20	4.510	16.67	12.04	4.866
.1	25.27	34.64	6.877	20.71	14.98	5.910
.2	29.58	22.64	8.758	24.04	17.56	6.789
.3	34.75	14.37	10.181	27.50	24.12	7.487
.4	40.88	14.99	11.173	31.66	27.00	7.990
.5	47.58	16.19	11.764	35.53	25.47	8.283
.6	54.26	11.03	11.979	41.87	30.01	8.351
.7	60.68	10.77	11.829	46.04	30.63	7.907
.8	67.11	9.81	11.366	50.10	27.67	7.676
.9	73.42	8.51	10.693	54.07	25.56	7.452
1.0	78.89	7.69	9.880	57.64	24.62	7.211
1.1	84.37	7.19	8.997	62.10	24.64	6.928
1.2	89.24	7.59	7.747	66.07	23.85	6.609
1.3	93.65	6.90	7.181	69.34	24.04	6.156
1.4	96.86	6.31	6.976	73.01	24.19	5.675
1.5	100.19	7.02	7.044	75.98	22.61	5.259
1.6	104.60	6.72	7.301	78.66	21.60	5.000
1.7	108.88	7.54	7.927	81.04	20.82	6.128
1.8	113.16	8.39	8.140	83.62	19.74	5.694
1.9	118.22	8.47	8.170	87.09	19.37	4.691
2.0	122.32	8.53	8.091	91.11	20.42	3.381
2.1	127.08	8.67	7.975	92.84	19.68	2.026
2.2	131.49	9.49	8.035	94.63	18.58	.084
2.3	135.41	9.77	7.971	91.06	12.42	-.077
2.4	139.94	9.89	7.911	90.56	9.60	.597
2.5	144.82	9.96	7.898	92.64	8.58	1.713
2.6	149.22	9.99	7.976	94.92	7.74	2.874
2.7	153.50	9.80	8.121	96.41	6.53	3.363
2.8	158.38	9.10	8.545	98.00	6.50	3.461
2.9	162.31	9.16	9.079	100.28	6.93	3.310
3.0	167.78	8.98	9.620	102.26	7.13	3.080
3.1	173.97	9.27	10.066	103.25	7.86	2.944
3.2	179.69	9.24	10.504	104.84	8.38	3.650
3.3	184.44	9.75	10.227	107.02	8.82	3.925
3.4	190.27	10.43	9.600	109.30	9.45	4.072
3.5	196.46	11.22	8.787	111.68	10.33	4.104
3.6	201.22	11.91	7.949	114.76	11.51	4.033
3.7	205.15	11.87	7.159	117.04	12.24	3.864
3.8	208.36	11.76	6.839	116.94	12.83	3.512
3.9	211.58	11.85	6.772	119.12	13.02	3.115
4.0	215.70	11.82	6.840	123.29	14.07	2.742
4.1	220.63	12.05	6.920	124.22	15.02	2.461
4.2	224.41	11.56	6.695	125.41	15.51	2.378
4.3	226.92	11.23	6.488	124.86	15.48	2.462
4.4	230.85	10.89	6.271	127.17	15.58	2.596
4.5	234.35	10.58	6.073	128.97	16.70	2.658
4.6	238.17	10.36	5.921	131.07	17.26	2.525
4.7	240.93	9.92	5.845	131.96	18.04	2.074
4.8	244.49	9.41	5.874	132.92	18.43	1.181
4.9	247.49	8.87	6.036	132.56	19.13	-.275

TABLE 8 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 8 (X = 14 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	17.58	12.18	3.740	18.76	18.06	4.964
.1	21.67	10.15	6.401	24.82	20.83	7.221
.2	25.31	20.02	8.738	28.09	18.48	8.771
.3	30.00	29.96	10.683	31.61	21.27	9.681
.4	36.82	33.35	12.166	37.30	22.96	10.020
.5	44.24	27.75	13.120	43.80	18.45	9.857
.6	51.12	19.61	13.476	50.56	23.82	9.260
.7	58.87	19.23	12.945	54.51	25.53	7.775
.8	66.87	18.77	11.964	58.44	22.91	6.843
.9	72.81	16.76	10.713	61.53	21.51	6.156
1.0	77.66	17.11	9.395	64.40	20.86	5.698
1.1	82.27	15.01	8.210	67.84	20.67	5.453
1.2	86.99	13.97	7.576	71.17	20.51	5.518
1.3	91.23	13.31	7.243	74.30	20.08	5.629
1.4	94.63	11.84	7.152	77.23	20.58	5.775
1.5	98.75	11.05	7.210	80.46	19.04	5.924
1.6	103.23	10.86	7.326	83.29	18.50	6.045
1.7	107.11	10.83	7.321	87.43	18.46	6.995
1.8	111.11	10.67	7.292	90.16	17.98	6.310
1.9	115.53	10.93	7.227	93.79	17.75	4.939
2.0	118.62	11.33	7.147	97.48	18.93	3.205
2.1	122.99	11.38	7.068	98.95	18.12	1.429
2.2	127.47	11.11	7.022	100.46	17.35	-.719
2.3	130.50	11.20	7.024	96.32	12.19	-1.150
2.4	134.50	11.21	7.060	95.71	10.30	-.721
2.5	138.87	11.29	7.136	96.27	9.25	.206
2.6	142.98	11.59	7.255	97.73	8.18	1.267
2.7	146.74	11.30	7.352	98.44	7.45	1.638
2.8	150.74	11.54	7.666	99.55	7.88	2.057
2.9	155.71	10.69	8.043	100.86	8.11	2.394
3.0	159.59	11.38	8.415	101.77	7.16	2.699
3.1	163.95	12.35	8.717	102.68	6.82	3.024
3.2	169.53	12.26	8.832	105.11	7.03	3.913
3.3	174.38	12.91	8.772	107.13	6.98	4.224
3.4	179.47	13.44	8.665	110.16	6.19	4.268
3.5	184.44	13.54	8.618	113.19	8.05	4.121
3.6	189.04	14.02	8.737	114.20	8.46	3.859
3.7	193.04	14.09	9.930	116.12	8.83	3.689
3.8	198.50	14.12	10.227	118.44	10.14	3.306
3.9	204.68	13.70	10.148	120.26	9.76	2.900
4.0	211.33	12.70	9.761	122.30	10.80	2.546
4.1	217.44	12.12	9.133	124.08	12.38	2.318
4.2	221.80	11.90	7.774	124.31	12.34	2.406
4.3	223.93	11.37	7.110	124.79	13.50	2.570
4.4	228.50	11.17	6.792	127.25	13.34	2.762
4.5	232.39	11.06	6.733	128.85	13.67	2.875
4.6	236.76	10.85	6.847	131.07	14.69	2.800
4.7	240.68	10.28	7.047	132.00	15.08	2.430
4.8	244.16	9.82	7.248	133.42	15.40	1.657
4.9	248.04	9.34	7.361	133.33	16.03	.374

TABLE 9 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 10 (X = 16 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2	Temp Deg C	% Error 100(T-μ)/T	H.T. Rate cal/sec cm^2
0.0	22.41	16.83	4.266	20.29	36.84	10.969
.1	27.25	13.26	8.255	26.04	38.38	10.277
.2	31.79	17.79	11.127	31.18	43.91	9.669
.3	37.68	25.34	12.983	37.08	45.48	9.129
.4	45.40	23.74	13.928	42.52	44.49	8.642
.5	54.48	21.65	14.062	45.84	29.32	8.193
.6	62.92	16.57	13.490	50.68	28.34	7.765
.7	69.21	15.12	11.660	54.91	27.06	7.328
.8	75.38	14.65	10.362	59.27	25.84	6.886
.9	80.95	13.65	9.278	62.65	25.39	6.470
1.0	84.57	12.56	8.419	65.92	24.53	6.109
1.1	89.29	11.96	7.798	69.55	24.62	5.831
1.2	94.26	11.90	7.572	72.82	23.35	5.770
1.3	98.49	12.09	7.441	75.48	22.10	5.711
1.4	102.12	11.58	7.401	78.87	22.45	5.646
1.5	105.87	11.04	7.412	82.01	21.20	5.547
1.6	110.35	11.41	7.435	85.28	20.57	5.384
1.7	114.46	11.74	7.545	88.24	21.04	5.310
1.8	118.82	11.93	7.394	90.72	20.92	4.770
1.9	122.93	12.42	7.140	93.27	20.83	4.023
2.0	126.44	12.97	6.871	95.08	20.03	3.168
2.1	130.56	13.60	6.671	97.26	19.29	2.307
2.2	134.31	13.88	6.774	99.08	18.65	1.035
2.3	137.33	14.25	6.928	98.11	16.02	.820
2.4	141.33	14.30	7.138	98.35	13.77	1.126
2.5	146.17	13.44	7.369	99.38	12.39	1.779
2.6	150.16	13.28	7.585	100.65	11.59	2.604
2.7	154.15	13.36	7.601	101.61	11.73	3.756
2.8	158.75	12.82	7.764	105.49	11.23	4.183
2.9	162.98	12.06	7.965	107.42	10.23	4.238
3.0	166.98	11.91	8.174	109.48	9.39	4.052
3.1	171.58	12.11	8.361	111.42	10.41	3.754
3.2	176.78	11.00	8.471	113.90	11.20	3.439
3.3	181.38	11.08	8.512	115.77	12.77	3.371
3.4	185.97	11.72	8.542	117.47	14.36	3.420
3.5	190.93	11.69	8.609	118.68	15.48	3.520
3.6	195.65	11.68	8.762	120.49	16.85	3.601
3.7	200.25	11.45	9.290	123.64	18.04	3.313
3.8	205.70	11.35	9.626	125.82	18.78	3.345
3.9	210.90	11.05	9.873	126.30	20.21	3.451
4.0	216.84	10.53	10.022	129.16	20.39	3.583
4.1	223.03	10.19	10.063	130.43	21.24	3.692
4.2	228.23	9.61	9.941	132.31	21.79	3.902
4.3	232.68	8.51	9.734	135.00	21.32	3.773
4.4	238.75	8.29	9.431	137.64	20.98	3.453
4.5	243.85	8.13	9.025	139.93	20.65	3.019
4.6	249.01	7.36	8.507	140.63	21.37	2.547
4.7	253.93	7.05	7.872	140.01	22.69	2.115
4.8	257.17	6.90	7.112	142.87	23.04	1.798
4.9	261.15	6.53	6.219	144.15	23.51	1.674

TABLE 10 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 11 (X = 18 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error $100(T-\mu)/T$	H.T. Rate cal/sec cm^2	Temp Deg C	% Error $100(T-\mu)/T$	H.T. Rate cal/sec cm^2
0.0	20.45	21.15	4.107	18.58	19.73	6.297
.1	25.21	22.21	7.917	23.51	17.82	7.250
.2	29.41	29.13	10.828	27.32	14.96	7.984
.3	35.29	35.29	12.893	31.46	15.76	8.507
.4	43.55	31.98	14.166	36.17	17.69	8.829
.5	51.95	26.01	14.701	40.75	16.35	8.958
.6	61.25	16.47	14.553	47.01	19.54	8.904
.7	67.97	13.77	13.511	51.67	19.46	8.537
.8	75.03	10.85	12.246	55.97	17.09	8.230
.9	81.75	8.96	10.864	60.03	16.23	7.910
1.0	87.57	8.66	9.527	64.27	15.98	7.587
1.1	92.39	8.14	8.395	69.13	16.34	7.271
1.2	96.98	8.03	7.790	73.14	15.90	6.930
1.3	100.67	8.17	7.534	76.31	15.55	6.692
1.4	104.37	7.98	7.533	80.14	15.24	6.507
1.5	109.30	8.46	7.668	83.87	14.38	6.373
1.6	114.00	9.31	7.824	87.14	14.28	6.288
1.7	117.92	10.36	7.625	90.78	14.26	7.185
1.8	122.40	10.66	7.573	93.95	13.98	6.513
1.9	126.63	11.31	7.556	97.31	14.14	5.203
2.0	129.79	11.82	7.568	101.28	15.79	3.542
2.1	134.83	11.98	7.606	103.38	15.10	1.814
2.2	139.20	11.13	7.674	105.15	14.29	-.458
2.3	142.79	10.95	7.767	100.49	9.11	-.965
2.4	147.49	10.89	7.895	100.11	7.00	-.593
2.5	152.53	11.08	8.081	101.37	6.56	.304
2.6	156.45	11.45	8.346	102.54	5.99	1.374
2.7	160.82	11.88	9.007	102.91	6.09	2.107
2.8	166.42	11.43	9.338	104.59	5.71	2.435
2.9	171.79	10.77	9.521	105.90	4.41	2.523
3.0	176.95	11.43	9.582	107.39	4.95	2.506
3.1	182.66	11.93	9.548	108.42	6.20	2.521
3.2	187.81	11.96	9.256	110.05	6.82	3.052
3.3	192.29	12.85	9.189	111.31	7.60	3.376
3.4	197.67	13.50	9.217	113.65	7.82	3.655
3.5	203.15	13.23	9.299	116.17	8.48	3.884
3.6	208.19	12.97	9.398	118.31	10.23	4.061
3.7	213.01	12.33	9.735	120.37	11.70	4.244
3.8	218.83	11.57	9.559	122.51	11.84	4.255
3.9	223.65	10.99	9.118	124.38	12.71	4.120
4.0	229.22	10.40	8.480	128.11	13.83	3.821
4.1	233.83	9.84	7.709	129.62	14.81	3.339
4.2	238.25	9.32	6.576	131.42	15.97	2.184
4.3	240.23	8.39	5.898	132.21	16.10	1.523
4.4	244.00	7.93	5.494	133.61	15.65	1.151
4.5	247.23	7.91	5.335	133.27	15.34	1.077
4.6	250.02	7.89	5.394	132.59	15.27	1.308
4.7	253.12	7.40	5.642	134.65	17.31	1.852
4.8	256.18	7.20	6.050	137.30	18.13	2.717
4.9	259.86	7.44	6.591	137.55	17.33	3.911

TABLE 11 - AVERAGE TEMPERATURE MEASUREMENTS AND HEAT TRANSFER RATES FOR STATION 12 (X = 20 INCHES)

	Data With No Cooling			Data With Water Cooling		
Time Sec	Temp Deg C	% Error $100(T-\mu)/T$	H.T. Rate cal/sec cm^2	Temp Deg C	% Error $100(T-\mu)/T$	H.T. Rate cal/sec cm^2
0.0	23.56	16.12	5.160	22.07	16.12	-2.359
.1	27.48	17.59	7.040	19.98	12.97	1.375
.2	30.96	11.64	8.662	22.65	10.83	4.213
.3	36.03	19.78	9.993	26.36	16.24	6.263
.4	42.70	24.00	11.004	29.61	19.00	7.632
.5	49.52	27.48	11.663	34.04	20.65	8.427
.6	55.38	21.20	11.939	40.03	23.77	8.756
.7	62.10	21.06	11.737	44.00	23.70	8.381
.8	67.79	18.67	11.144	48.44	21.16	8.321
.9	75.44	16.07	10.341	53.23	19.37	8.301
1.0	80.66	15.46	9.466	57.24	20.02	8.253
1.1	84.38	13.92	8.657	62.46	20.21	8.113
1.2	90.41	11.11	8.240	66.91	19.89	7.552
1.3	92.96	9.90	7.899	70.97	19.69	7.185
1.4	98.64	5.99	7.677	75.03	19.79	6.862
1.5	102.70	5.53	7.527	78.41	18.87	6.586
1.6	107.46	6.05	7.403	81.69	18.75	6.361
1.7	110.82	6.67	7.089	85.09	18.73	7.200
1.8	114.30	7.70	6.914	88.66	18.37	6.387
1.9	118.71	8.57	6.819	92.52	19.12	4.940
2.0	122.31	9.71	6.799	95.71	20.20	3.189
2.1	126.25	10.23	6.849	97.55	19.50	1.469
2.2	130.20	10.56	6.996	98.71	18.19	-.379
2.3	133.44	10.55	7.156	94.46	11.88	-.695
2.4	137.74	10.37	7.319	94.55	10.40	-.218
2.5	142.26	10.14	7.461	96.10	9.43	.705
2.6	146.32	9.88	7.558	97.36	9.04	1.726
2.7	150.15	9.91	7.515	98.03	8.27	2.232
2.8	154.90	9.94	7.470	99.68	8.13	2.415
2.9	158.85	9.38	7.436	100.74	6.63	2.379
3.0	162.56	9.26	7.439	101.90	5.23	2.241
3.1	166.97	8.97	7.504	103.06	4.64	2.122
3.2	171.49	8.01	7.732	104.70	4.29	2.107
3.3	175.09	8.07	7.957	105.77	4.03	2.512
3.4	179.84	8.09	8.190	107.22	3.80	3.087
3.5	184.72	7.66	8.408	107.89	4.41	3.692
3.6	189.59	8.04	8.592	111.08	7.66	4.186
3.7	193.88	8.65	8.823	113.79	9.89	4.221
3.8	199.33	8.58	8.792	115.92	10.20	4.201
3.9	203.62	8.59	8.591	118.72	11.34	4.071
4.0	208.15	9.03	8.240	120.18	10.46	3.890
4.1	213.91	8.78	7.760	121.89	11.71	3.715
4.2	217.60	9.15	6.838	124.17	11.73	3.860
4.3	220.18	9.11	6.341	126.65	11.74	3.806
4.4	224.03	8.83	6.109	128.98	11.19	3.681
4.5	228.28	8.92	6.156	131.10	10.88	3.531
4.6	231.04	9.23	6.495	132.77	10.65	3.400
4.7	235.12	9.13	7.140	133.12	9.84	3.330
4.8	238.26	8.74	8.106	136.78	6.98	3.365
4.9	244.44	9.51	9.405	138.28	5.91	3.551

APPENDIX A

WATER VELOCITY CALCULATION USING BOUNDARY LAYER THEORY

The terminal velocity for water flow in the washdown heat transfer tests will be calculated using boundary layer theory. The appropriate boundary layer equations are given below and a method for their numerical integration is indicated. This integration yields a velocity profile and from this the average terminal velocity is found. Details of the boundary layer theory and its application to the water flow problem are as follows.

Boundary Layer Theory

The boundary layer equations describe fluid flow in a thin layer near a solid boundary surface. For two dimensional flow, there is one describing partial differential boundary layer equation and an equation of continuity. For an incompressible liquid (water), the describing equations are:

$$u' \frac{\partial u'}{\partial x'} + v' \frac{\partial u'}{\partial y'} = - \frac{1}{\rho'} \frac{\partial P'}{\partial x'} + \frac{\mu'}{\rho'} \frac{\partial^2 u'}{\partial y'^2} \tag{A-1}$$

and

$$\frac{\partial u'}{\partial x'} + \frac{\partial v'}{\partial y'} = 0 \tag{A-2}$$

where the prime quantities are used to designate dimensional values. The system of partial differential equations, (A-1) and (A-2), can be solved numerically in a general way. First, these equations are made non-dimensional by introducing the following transformations.

$$u = \frac{u'}{Ur} , \quad v = \frac{v'}{\epsilon \cdot Ur} , \quad x = \frac{x'}{L} , \quad y = \frac{y'}{\epsilon \cdot L} , \quad P = \frac{P'}{\rho' \cdot Ur^2} , \quad \epsilon = \sqrt{\frac{\gamma}{Ur \cdot L}} \tag{A-3}$$

The resulting non-dimensional system of equations become:

$$u \frac{\partial u}{\partial x} + V \frac{\partial u}{\partial y} = - \frac{\partial P}{\partial x} + \frac{\partial^2 u}{\partial y^2} \tag{A-4}$$

and

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{A-5}$$

Equations (A-4) and (A-5) are attacked numerically by writing the partial derivatives in finite difference form; i.e.,

$$\frac{\partial u}{\partial x} = \frac{3\, u_{m+1,n} - 4\, u_{m,n} + u_{m-1,n}}{\partial\, \Delta x} \tag{A-6}$$

$$\frac{\partial u}{\partial y} = \frac{u_{m+1,n+1} - u_{m+1,n-1}}{\partial \ \Delta y} \quad \text{(A-7)}$$

$$\frac{\partial^2 u}{\partial y^2} = \frac{u_{m+1,n+1} - \partial \ u_{m+1,n} + u_{m+1,n-1}}{\Delta y^2} \quad \text{(A-8)}$$

$$u = \partial \ u_{m,n} - u_{m-1,n} \quad \text{(A-9)}$$

$$v = \partial \ v_{m,n} - v_{m-1,n} \quad \text{(A-10)}$$

Here "m" refers to values in the x direction and "n" to values in the y direction. In equations (A-6) through (A-10), values at "m" and "m - 1" are known and values at "m + 1" are found stepwise by solving equations (A-4) and (A-5). The details of solving these equations in a stepwise manner starting from an initial condition are given in reference (19). A computer program using this method has been written to facilitate the solution of problems. The program will not be given herein but a copy may be obtained from the author.

Velocity Profile For Water Flow Down An Inclined Plate

The numerical solution of the boundary layer equations outlined above can be used to compute the velocity profile for water flow down an inclined plate. An x-y coordinate system for this problem has the origin at the top of the plate with the +x direction being down the plate and the +y direction normal to and outward from the plate. The flow model assumed is that water arrives at the top of the plate at a single velocity and no boundary layer thickness but that the boundary layer grows until it includes the entire flow. Henceforth, the boundary layer cannot grow and the velocity profile remains fixed at its terminal value. The initial velocity is found by dividing the water flow rate leaving the nozzle (.211 GPM) by the nozzle area (nine holes of 1/16" diameter each), i.e.,

$$Uo' = \frac{.00047 \text{ ft}^3/\text{sec}}{.000192 \text{ ft}^2} = 2.44 \text{ ft/sec} \quad \text{(A-11)}$$

Since this water flows on an inclined plane, it will be accelerated due to the force of gravity. Therefore,

$$Ue' - Uo' + \sqrt{2gx' \cos\theta} \quad \text{(A-12)}$$

Equation (A-12) can be expressed in dimensionless coordinates. Let $Ur = Uo'$, $L = 1.0$ inches and since the inclination angle from the vertical is 33^o, equation (A-12) becomes:

$$Ue = 1 + 0.855\sqrt{x} \quad \text{(A-13)}$$

Bernouilli's law is used to determine the pressure gradient term in equation (A-4). For this problem and in dimensionless variables,

$$\frac{\partial P}{\partial X} = - Ue \frac{\partial Ue}{\partial X} \qquad \text{(A-14)}$$

or with the aid of equation (A-13), the pressure gradient is,

$$\frac{\partial P}{\partial X} = - \frac{.428(1 + .855\sqrt{x})}{\sqrt{x}} \qquad \text{(A-15)}$$

The velocity profile for water flow down the inclined plate is computed using equation (A-15) in the general computer program mentioned above. The initial profiles at the first two steps are given by Uo = 1 (non-dimensional value) and the velocity profile is computed for all downstream stations in a stepwise manner. The velocity profiles so computed are used to calculate the flow rate in the boundary layer, i.e.,

$$QT = \int u' dy' \qquad \text{(A-16)}$$

or in non-dimensional variables,

$$QT = Ue' \sqrt{\frac{\gamma L}{Uo'}} \int \frac{U}{Ue} dy \qquad \text{(A-17)}$$

where $\frac{U}{Ue}$ is the normalized non-dimensional velocity profile. The boundary layer calculation proceeds until the boundary layer flow rate (given by equation (A-17)) equals the total water flow rate. The total flow rate is .00047 ft^3/sec over a 2.5" wide plate, hence in two dimensions the total flow rate is:

$$QT = \frac{.00047 \text{ ft}^3/\text{sec}}{.208 \text{ ft}} = 0.0023 \text{ ft}^2/\text{sec} \qquad \text{(A-18)}$$

Both the velocity profile and the boundary layer flow rate were calculated by the computer program previously mentioned. In this calculation, steps of ΔX = .0016 and ΔY = .0016 were taken and the boundary layer flow rate reached .0023 ft^2/sec at X = .176. The resulting normalized velocity profile is given in Figure (A-1).

The boundary layer thickness is inferred from the velocity profile. That is, the standard definition of boundary layer thickness is the thickness of the velocity profile to the point where the local velocity is 99 percent of the free stream velocity ($\frac{U}{Ue}$ = .99). From Figure (A-1) the dimensionless boundary layer thickness is 1.532, hence the boundary layer thickness is:

$$yb' = yb \cdot \varepsilon \cdot L = 9.89 \times 10^{-4} \text{ ft} \tag{A-19}$$

The average velocity is then the ratio of the total flow rate to the boundary layer thickness.

$$\overline{U}' = \frac{.0023 \text{ ft}^2/\text{sec}}{.001 \text{ ft}} = 2.30 \text{ ft/sec} \tag{A-20}$$

The maximum profile velocity (freestream velocity) is given by equation (A-12) with X = .176 inches. This is:

$$U'_m = 3.33 \text{ ft/sec} \tag{A-21}$$

Figure (A-2) compares these computed average and maximum velocities with the measured velocities from the water velocity tests previously described. Here it is seen that the calculated values are in general agreement with the experimental values. The scatter in the experimental values is due in part to the interaction of the paper disks with the flow. That is, since the disks are approximately 1/2 as thick as the fluid layer, they do not float on the fluid surface but rather sink to some depth. Hence, the velocity they indicate is less than the surface velocity. The data in Figure (A-2) shows that the depth of sinking is different for different tests which was probably due to the manner in which the disks are dropped onto the flow. The experimental velocities require more distance down the plate to reach a terminal velocity than the calculation predicts. This may be due to an initial sinking and rising of the disks or it may be due to unsteadiness of the flow which is sometimes observed in the motion of the disk (i.e., veering to the left or right). Because of the uncertainties in the experimental velocities, the calculated average velocity will be used in reducing the heat transfer data of the wash-down heat transfer tests. The calculated velocity will only approximate the velocity in an actual heat transfer test because of expected flow irregularities and unsteadiness. Also, the calculated value is for water at room temperature (20°C) while the water temperature will increase in the tests as the water flows down a heated plate.

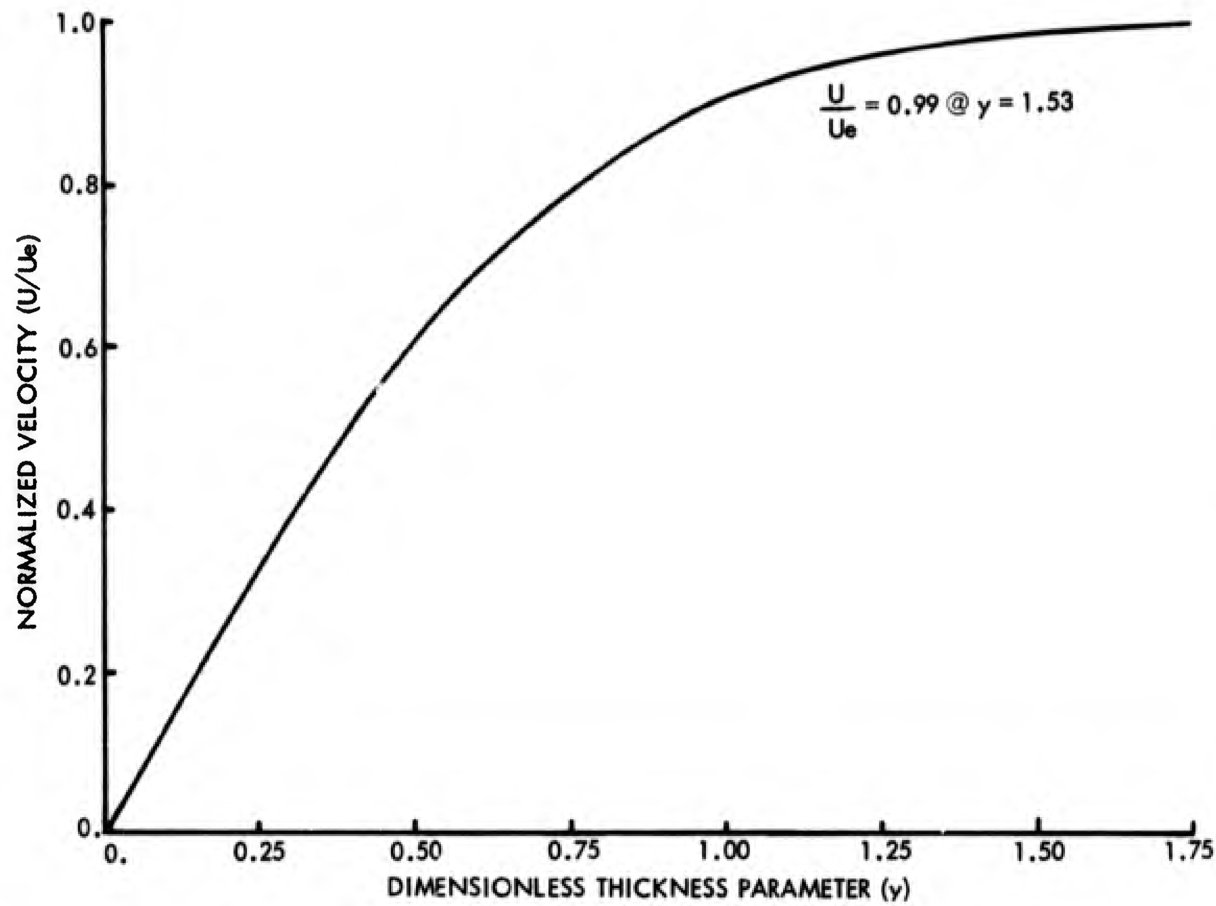

FIG. A1 NORMALIZED DIMENSIONLESS VELOCITY PROFILE FOR WASHDOWN VELOCITY CALCULATION

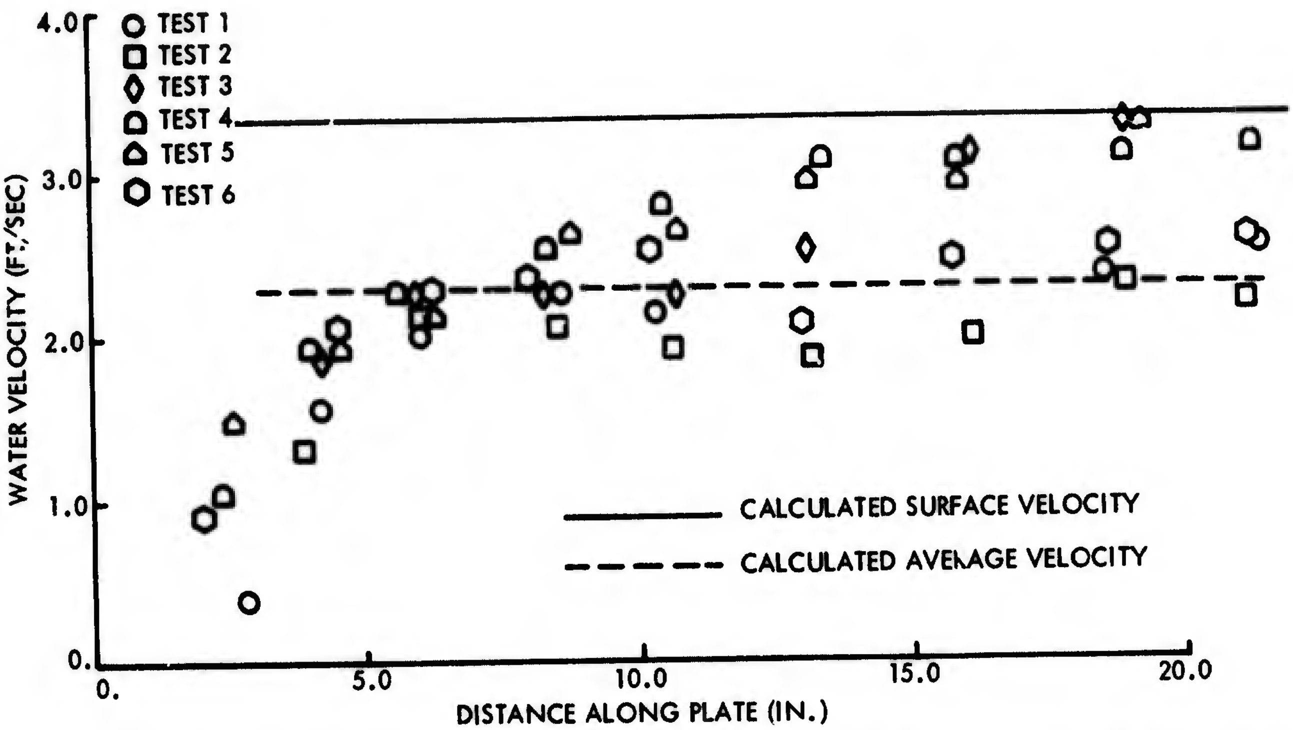

FIG. A2 COMPARISON OF MEASURED AND COMPUTED VELOCITIES OF WASHDOWN VELOCITY TESTS

APPENDIX B

TEMPERATURE DATA FOR WASHDOWN HEAT TRANSFER TESTS

The heat transfer data for the washdown heat transfer tests consists of temperature histories for each of the 11 thermocouple stations on the inclined plate model. The location of these stations on the model was given in Figure 3. There are six histories for each station of the water-on tests and seven histories for each station of the water-off tests. These temperature histories are shown in Figures B-1 through B-11 for the water-on thermocouple stations and in Figures B-12 through B-22 for the equivalent water-off stations. On some of the Figures B-1 through B-22 there are missing histories or parts of histories due to failures of thermocouples. An examination of these figures shows that there are variations in the shapes of the temperature histories for different tests at the same thermocouple station. This is especially true for the water-on tests where the largest discrepancies occur where boiling is suspected, i.e., for model temperatures slightly above 100°C. This behavior suggests that the boiling of thin layers may be greatly affected by small changes in flow or heating conditions. An alternate explanation is that heating of the water layer may cause unpredictable behavior in the flow such as the occurrence of shallow regions. In either case, it was decided that a better approach toward obtaining useful heat transfer data from these tests would be to average the temperature histories at each station for both the water-on and water-off condition.

Statistical analysis was used to indicate the error incurred in averaging data obtained under approximately identical conditions. That is, if the assumption is made that the differences in temperature histories are due to random experimental errors, statistical analysis will tell the magnitude of these errors. The error to be estimated is the error in the arithmetic mean temperature when compared with the statistical true mean temperature. This error will be found for a 95 percent confidence level and for a assumed normal (Gaussian) distribution of random experimental error. Thus, if the experimental errors occur in a normal distribution, the difference between the arithmetic mean temperature, $\overline{T}$, and the true mean temperature, μ, in 95 percent of the calculations will be found in an interval given by the following equation (reference (18)).

$$\overline{T} - \mu = D \frac{S}{\sqrt{N-1}} \tag{B-1}$$

In equation (B-1), D is a constant which varies with N, the number of samples. For the present work, D = 2.015 for the water-on tests (6 samples) and D = 1.943 for the water-off tests (7 samples). Larger values of D resulted for those stations with fewer samples due to thermocouple failure and these values may be found in reference (20). The quantity S is the unbiased standard deviation and is computed from:

$$S = \sqrt{\frac{\sum_{i=1}^{N} (T_i - \overline{T})^2}{N - 1}} \qquad \text{(B-2)}$$

Finally, the percentage error in average temperature is defined by the ratio of the Maximum expected error (for 95 percent confidence) to the average value, i.e.,

$$E = 100 \cdot \frac{\overline{T} - \mu}{\overline{T}} \qquad \text{(B-3)}$$

This error was computed for all of the temperature histories for both the water-on and water-off thermocouple stations. A tabulation of errors is given in the main section of this report.

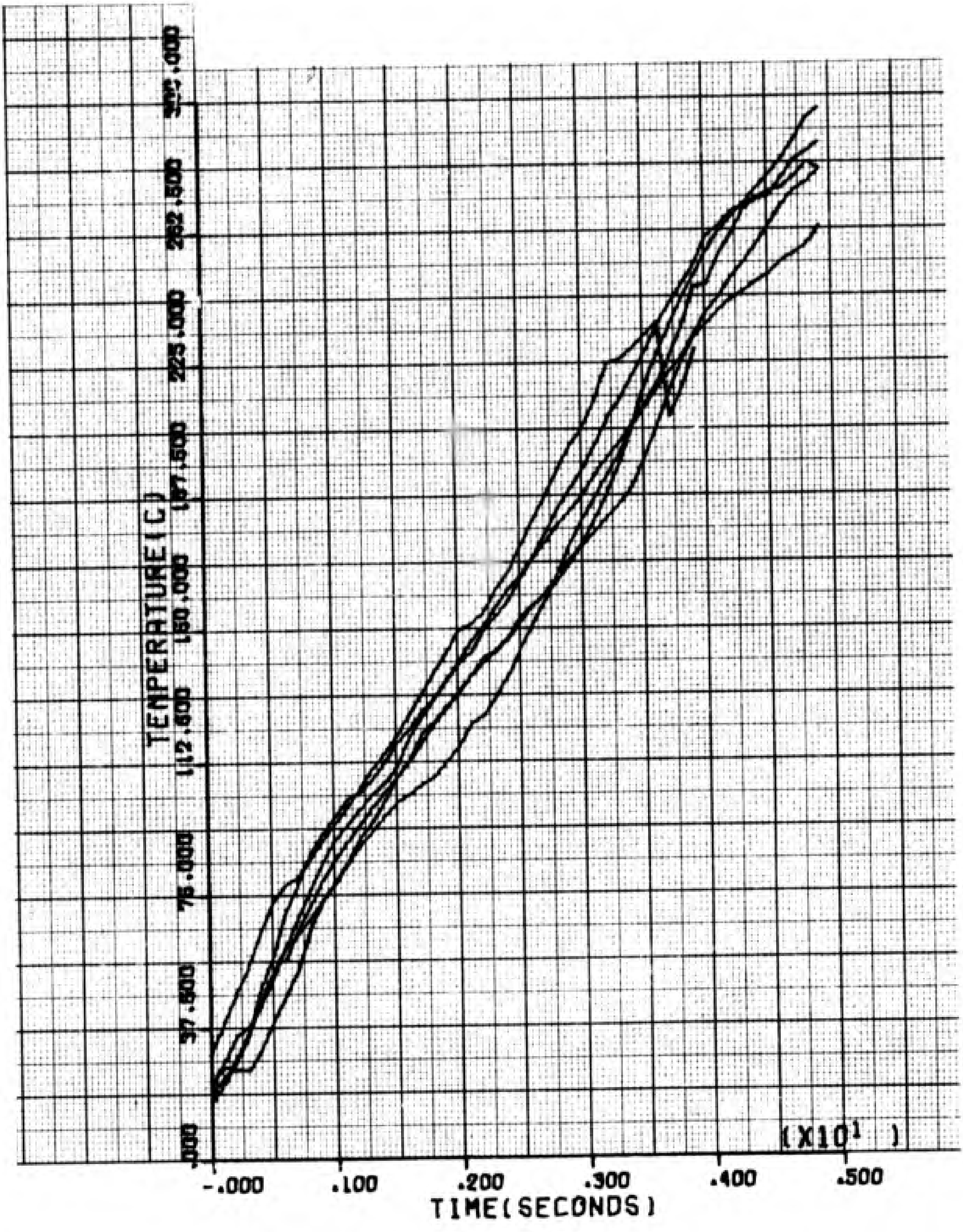

FIG. B-1 TEMPERATURE HISTORIES FOR STATION 1 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

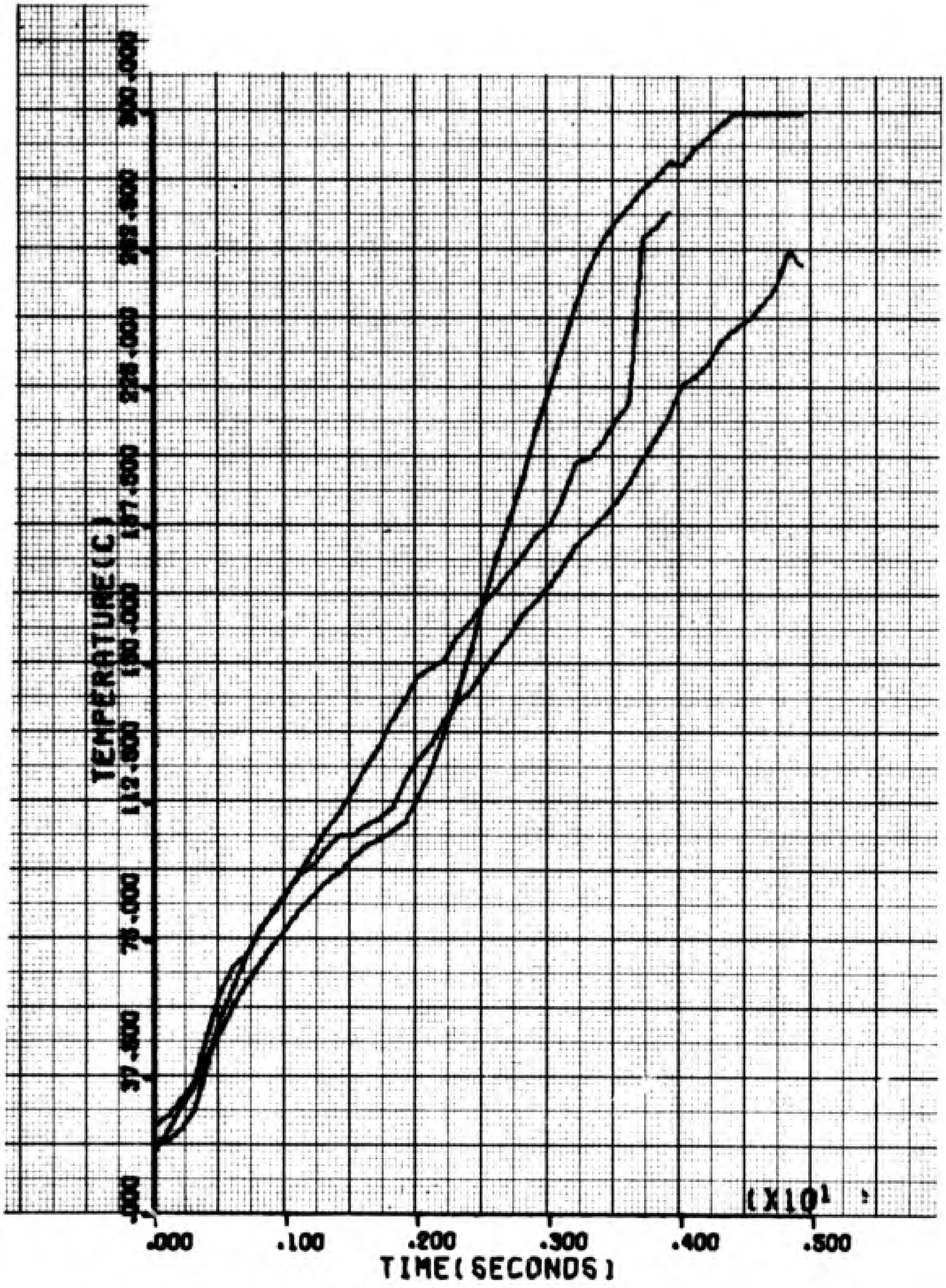

FIG. B-2 TEMPERATURE HISTORIES FOR STATION 2 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

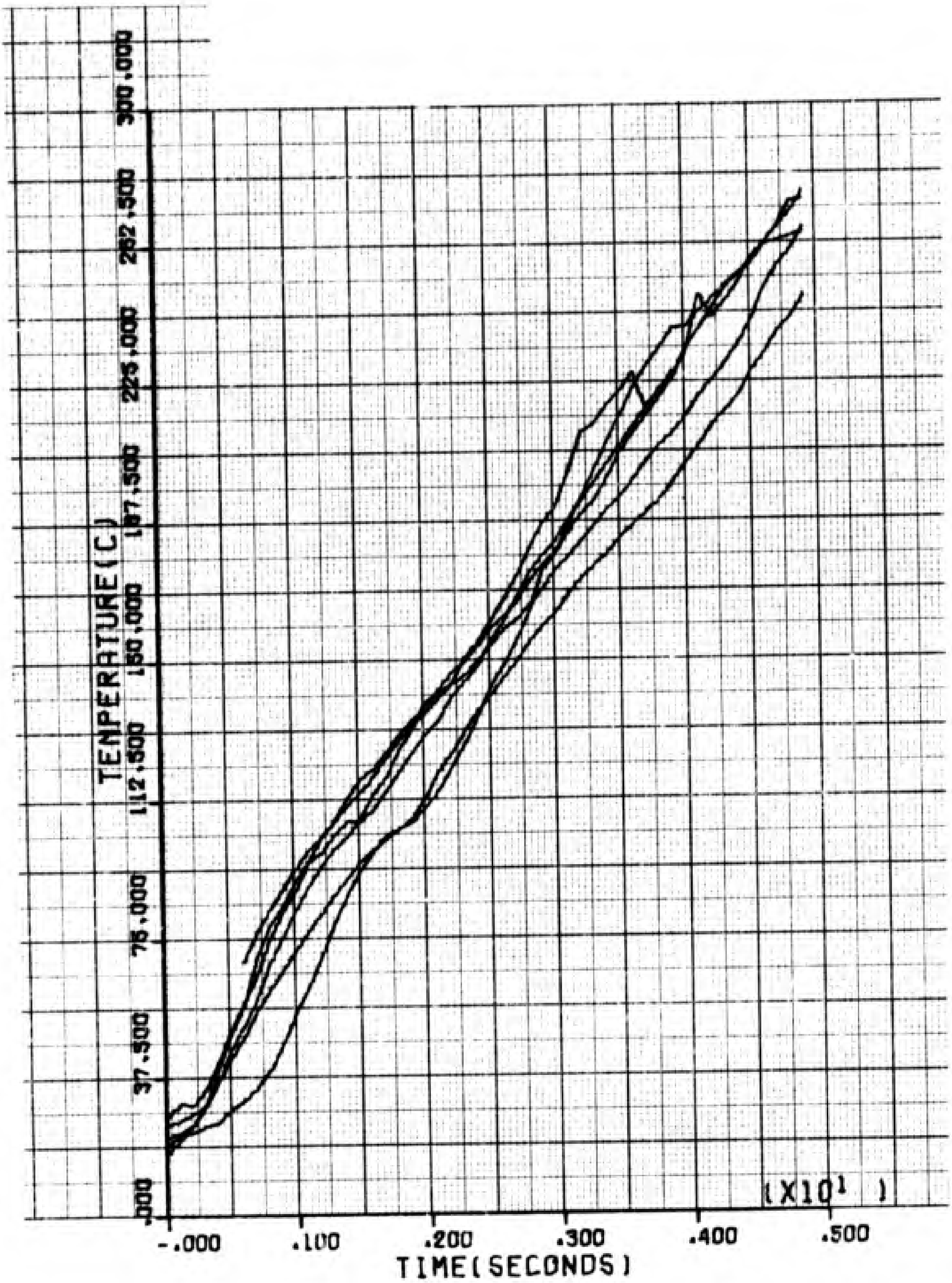

FIG. B-3 TEMPERATURE HISTORIES FOR STATION 3 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

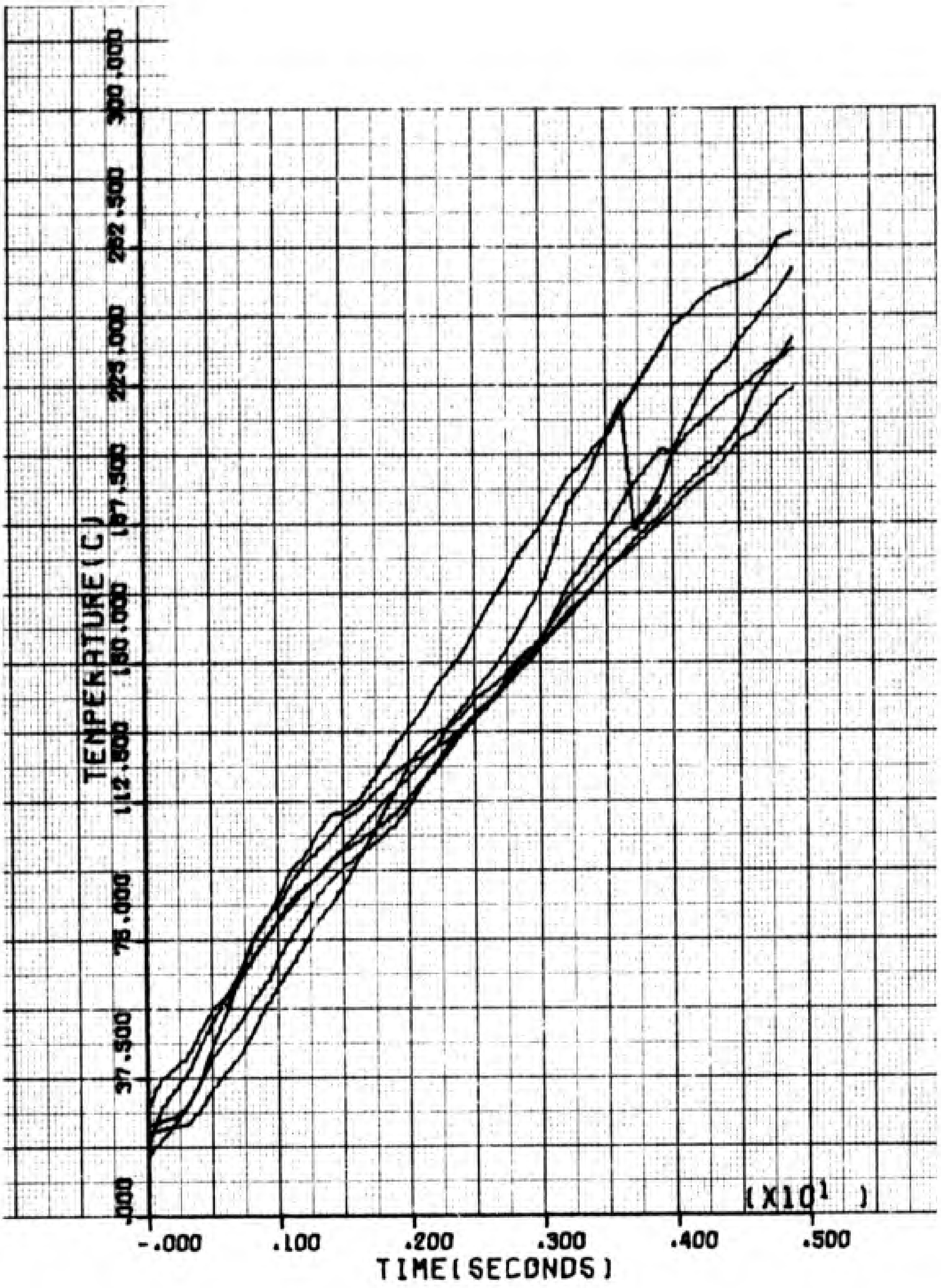

FIG. B-4 TEMPERATURE HISTORIES FOR STATION 4 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

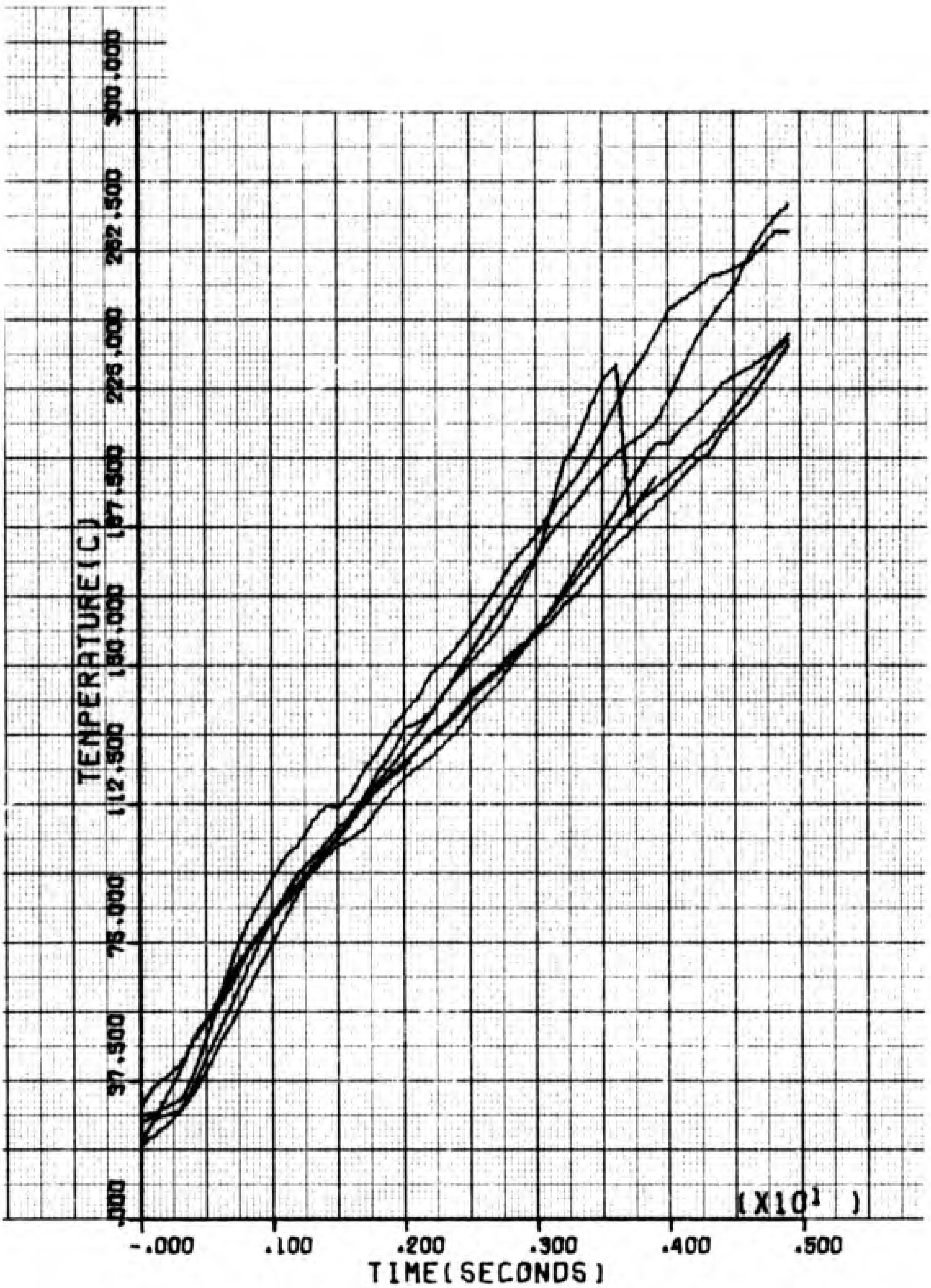

FIG. B-5 TEMPERATURE HISTORIES FOR STATION 5 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

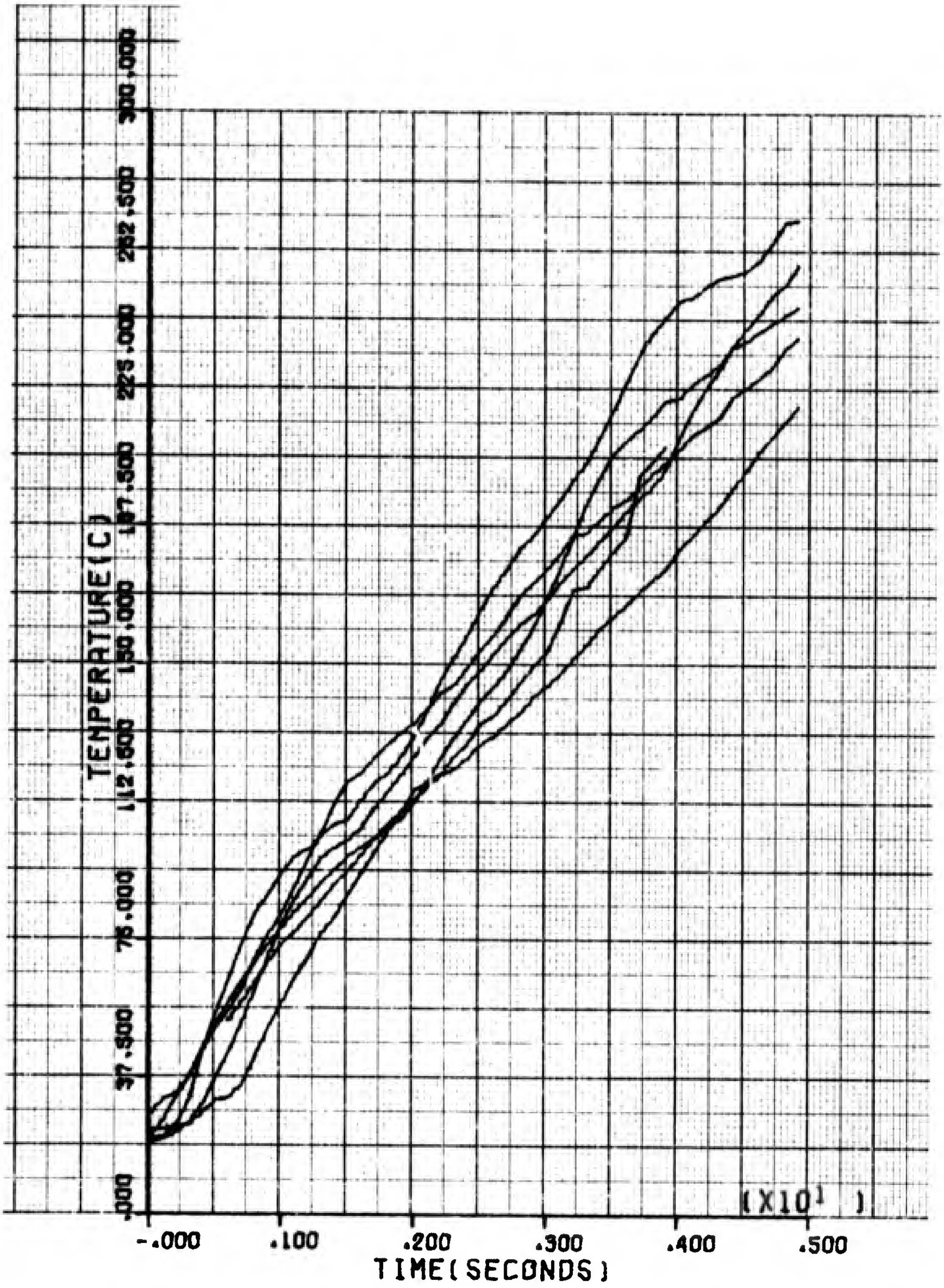

FIG. B-6 TEMPERATURE HISTORIES FOR STATION 6 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

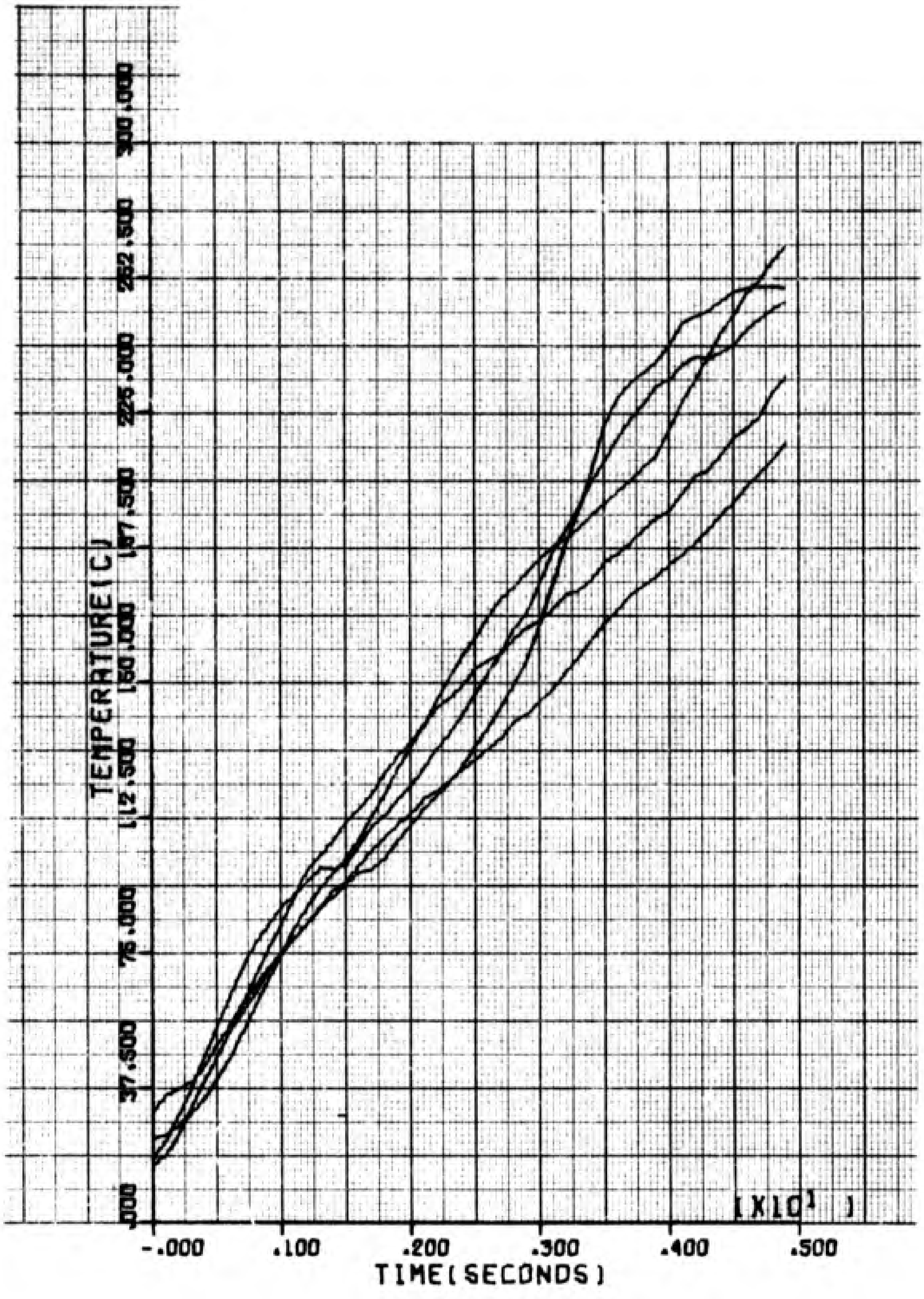

FIG. B-7 TEMPERATURE HISTORIES FOR STATION 7 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

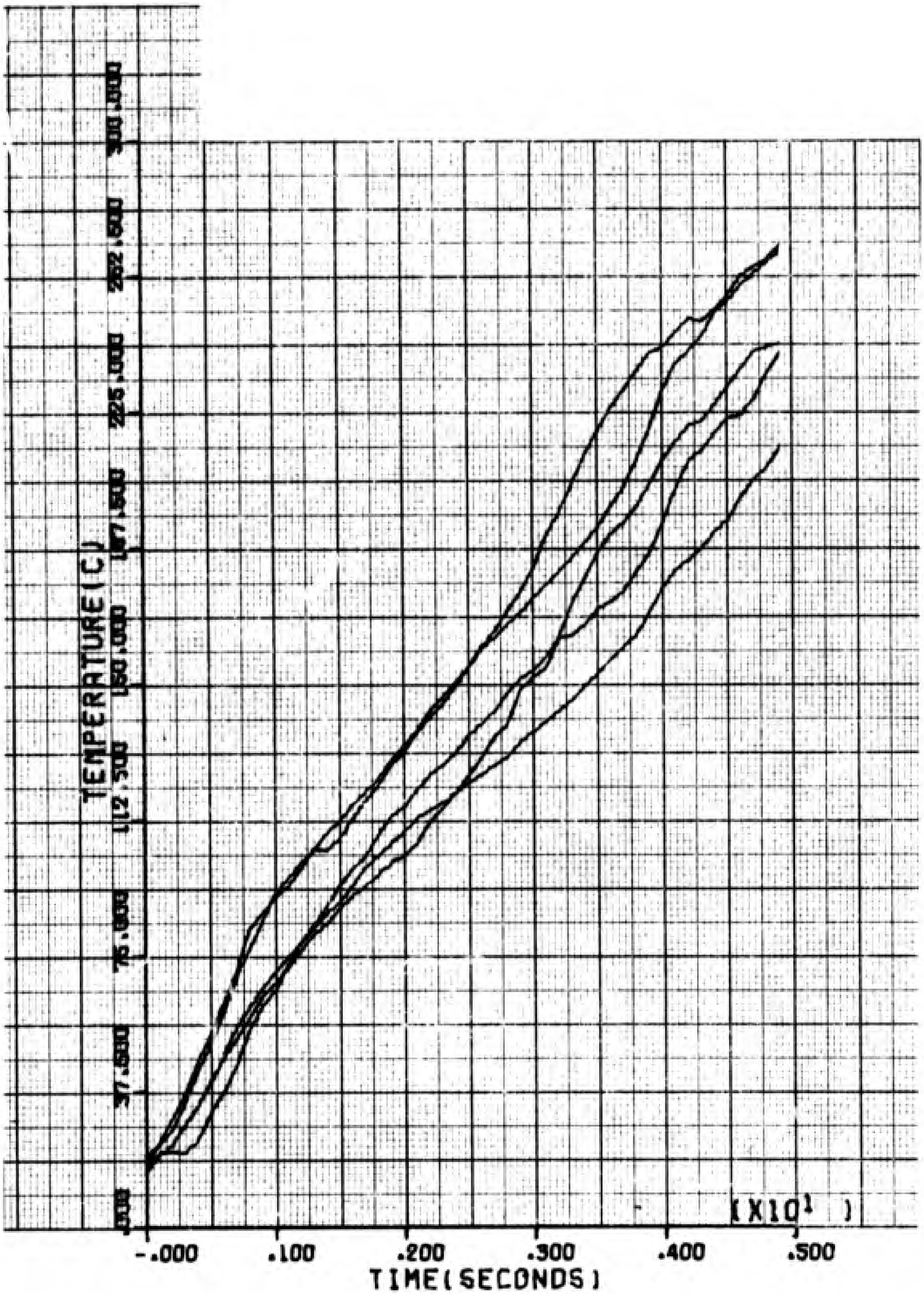

FIG. B-8 TEMPERATURE HISTORIES FOR STATION 8 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

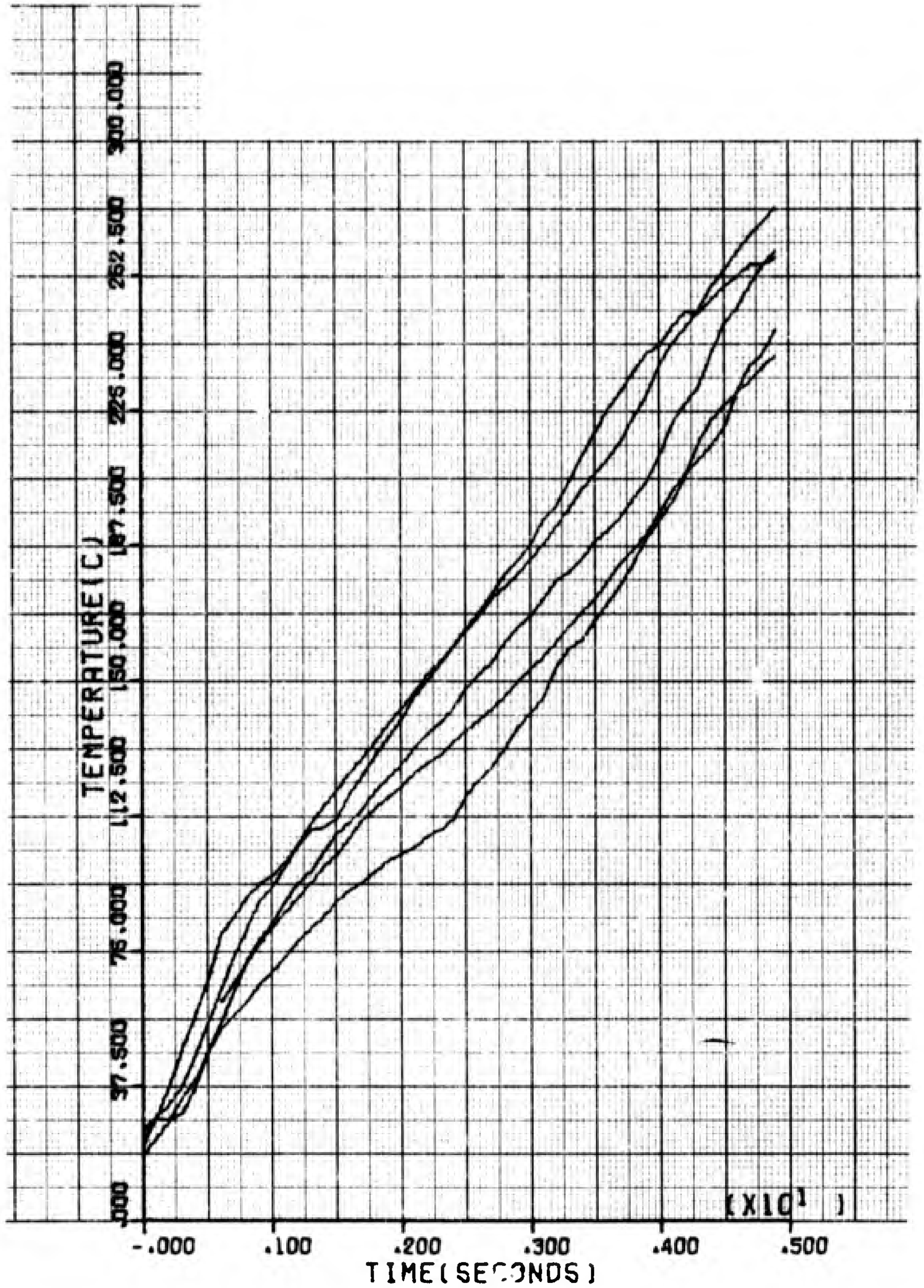

FIG. B-9 TEMPERATURE HISTORIES FOR STATION 10 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

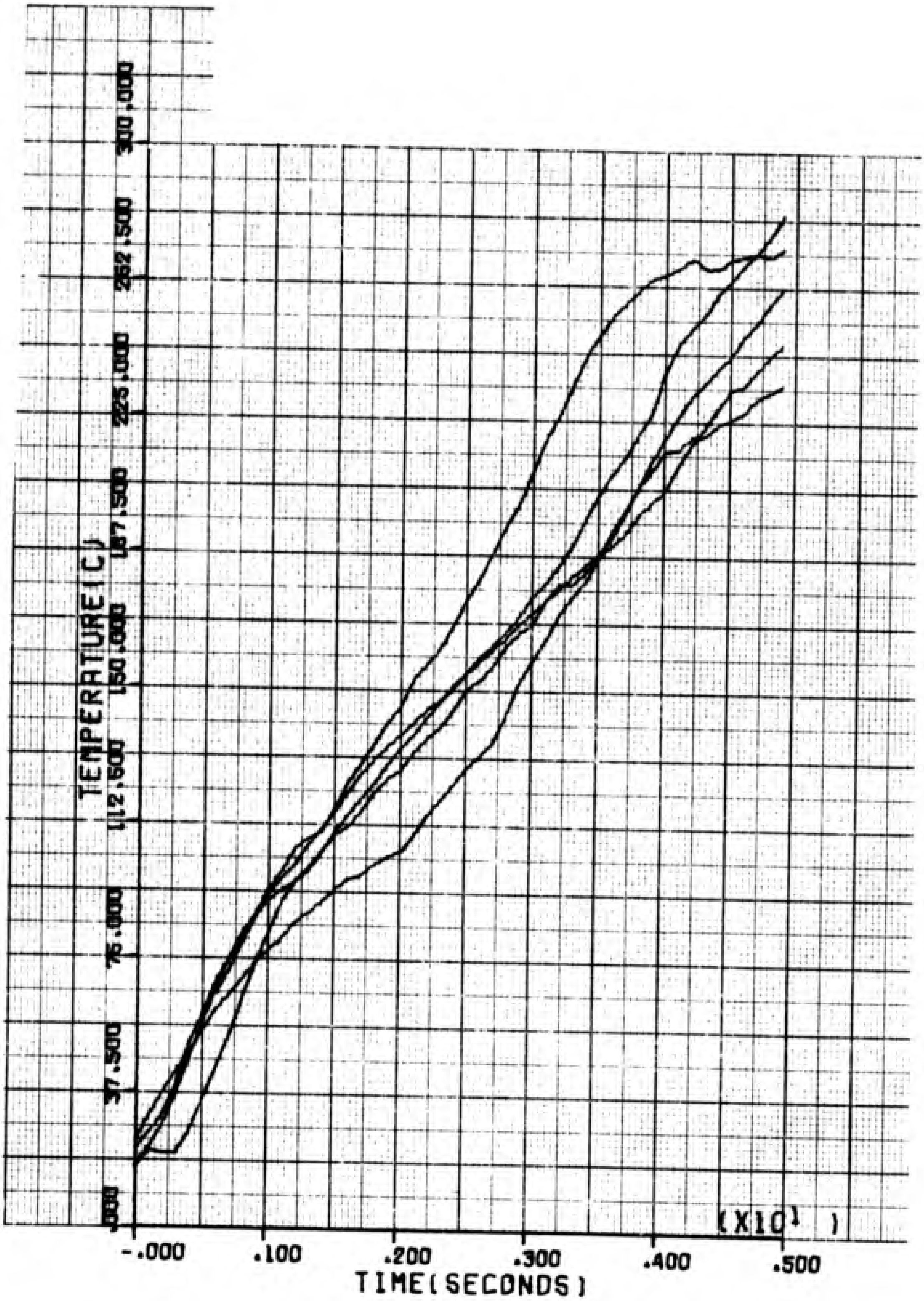

FIG. B-10 TEMPERATURE HISTORIES FOR STATION 11 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

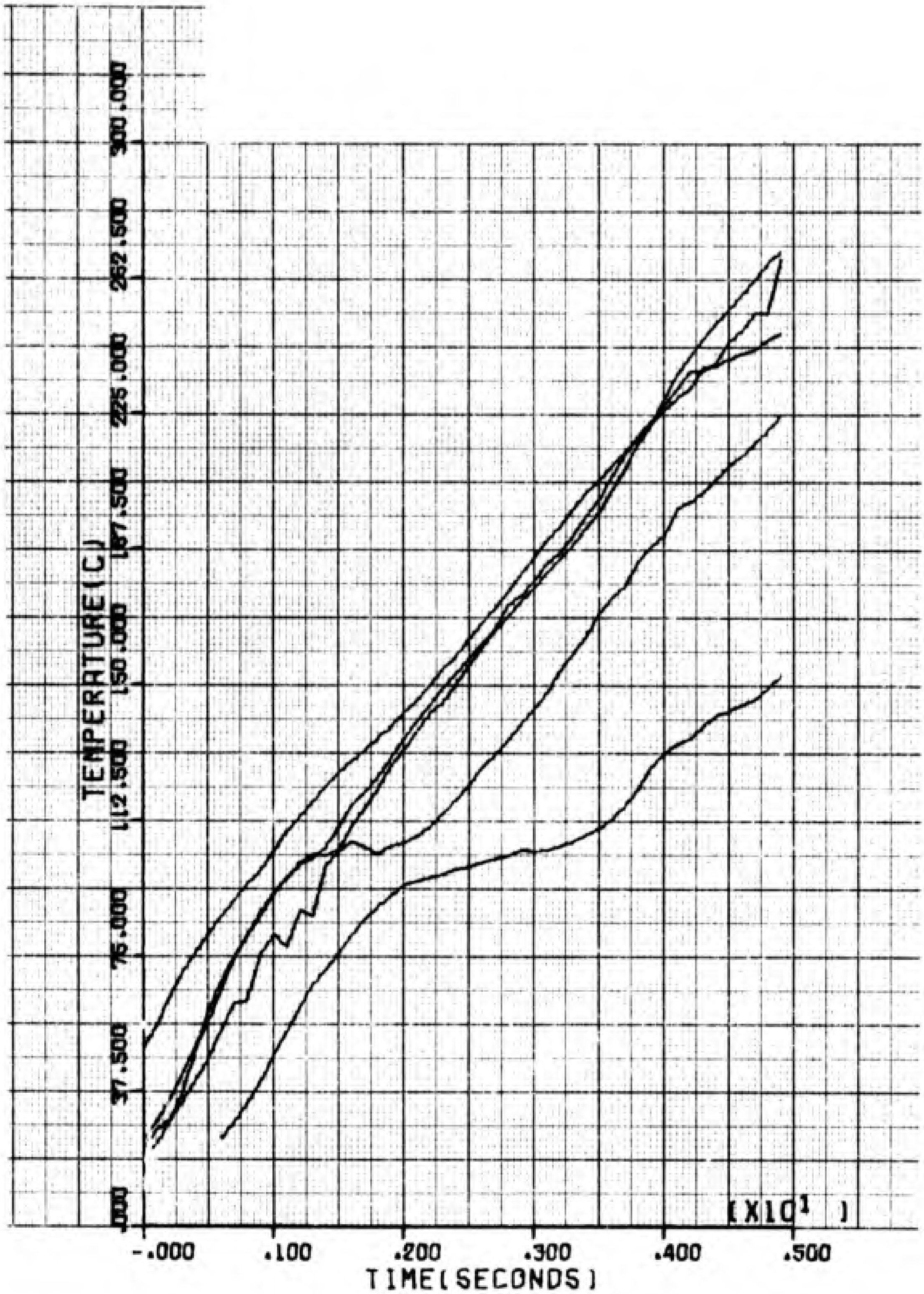

FIG. B-11 TEMPERATURE HISTORIES FOR STATION 12 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

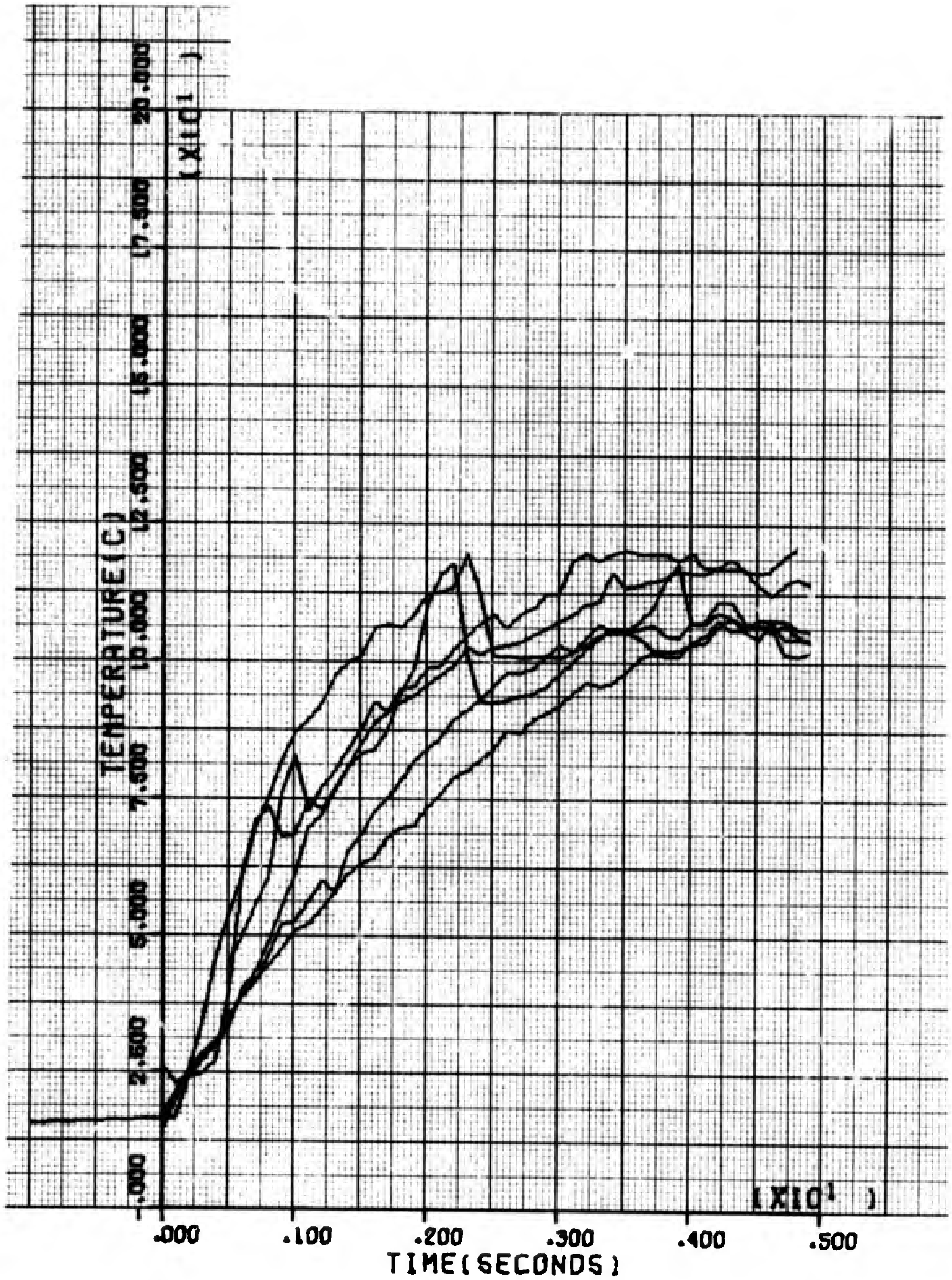

FIG. B-12 TEMPERATURE HISTORIES FOR STATION 1 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

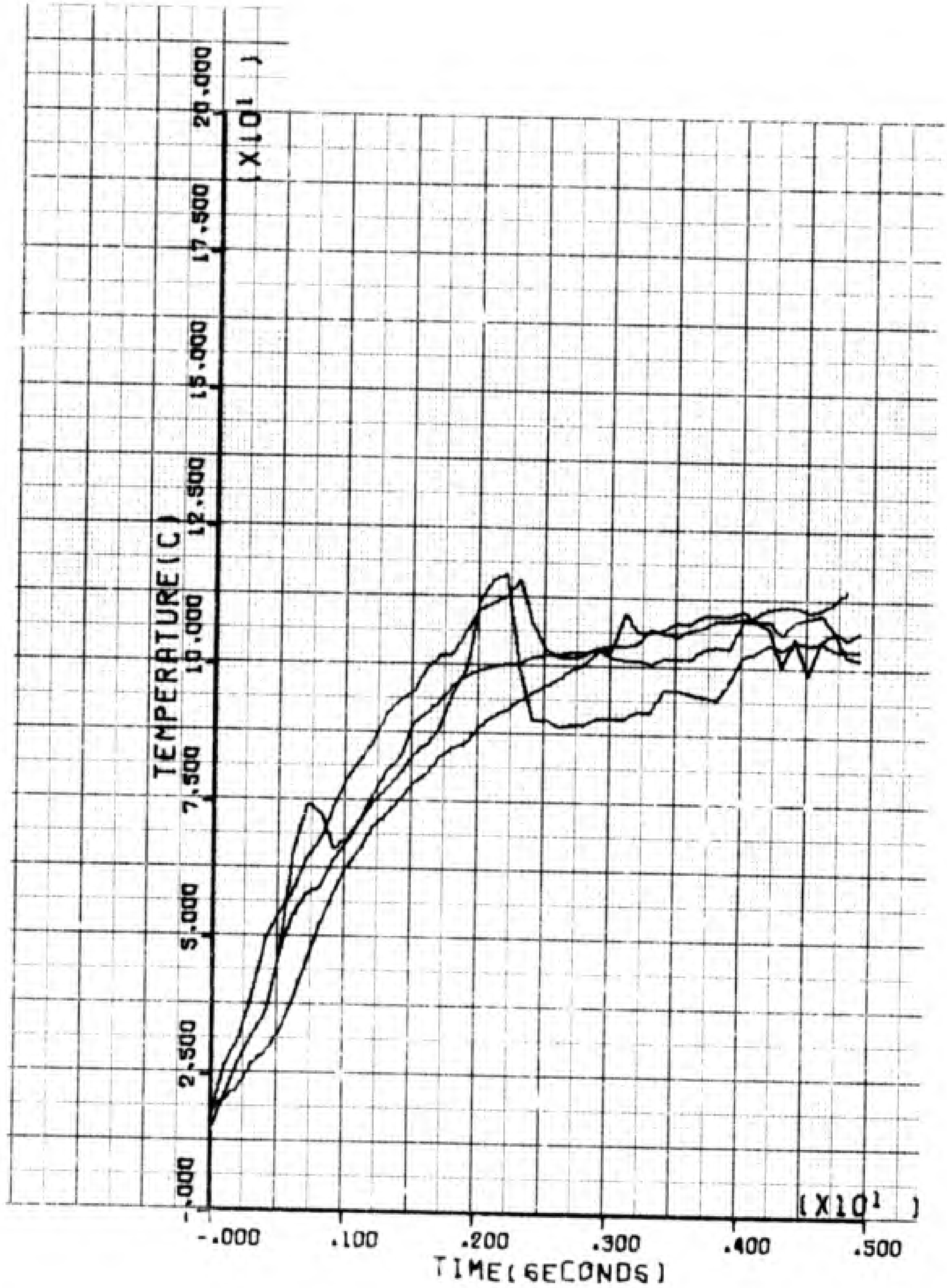

FIG. B-13 TEMPERATURE HISTORIES FOR STATION 2 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

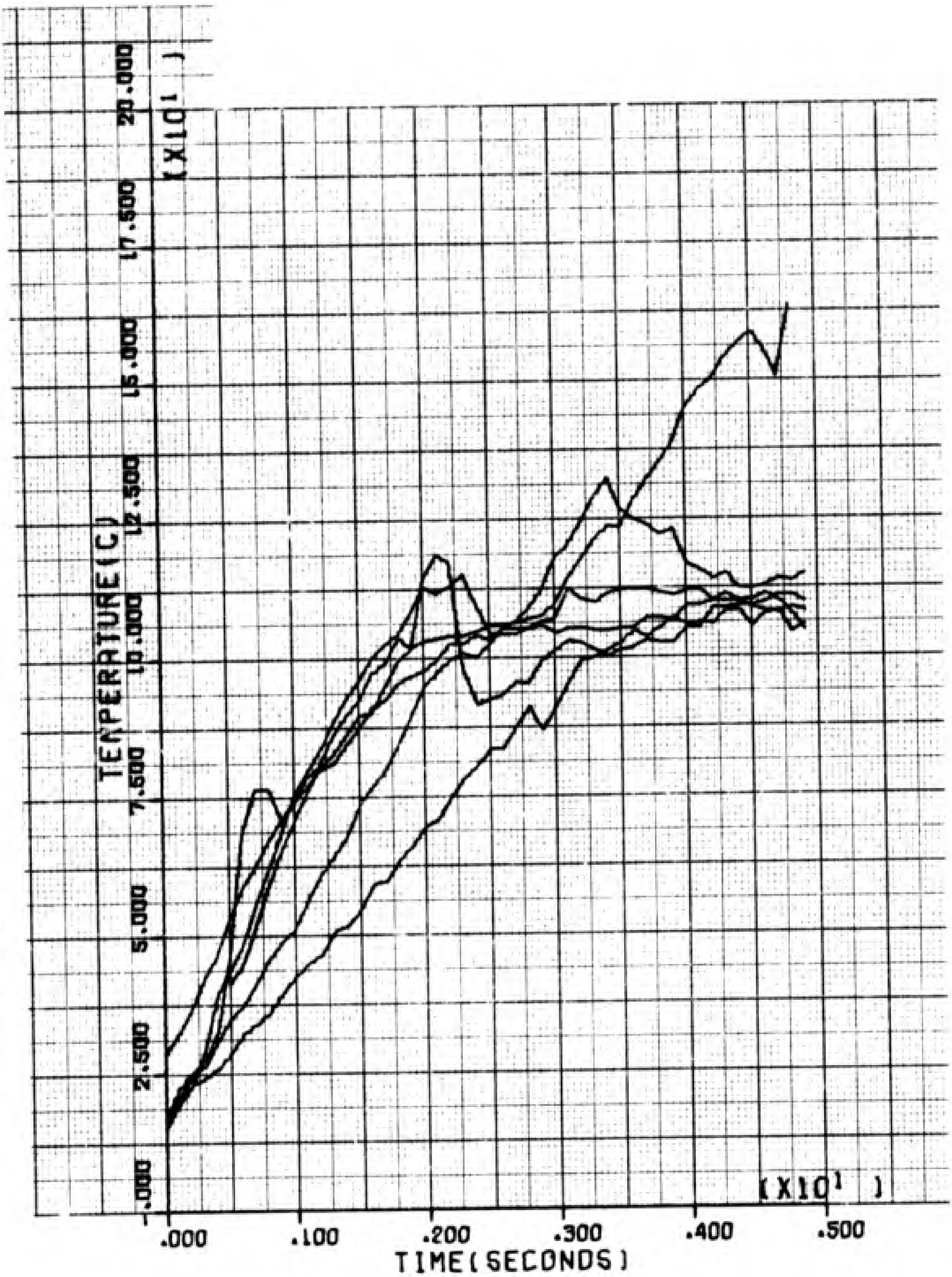

FIG. B-14 TEMPERATURE HISTORIES FOR STATION 3 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

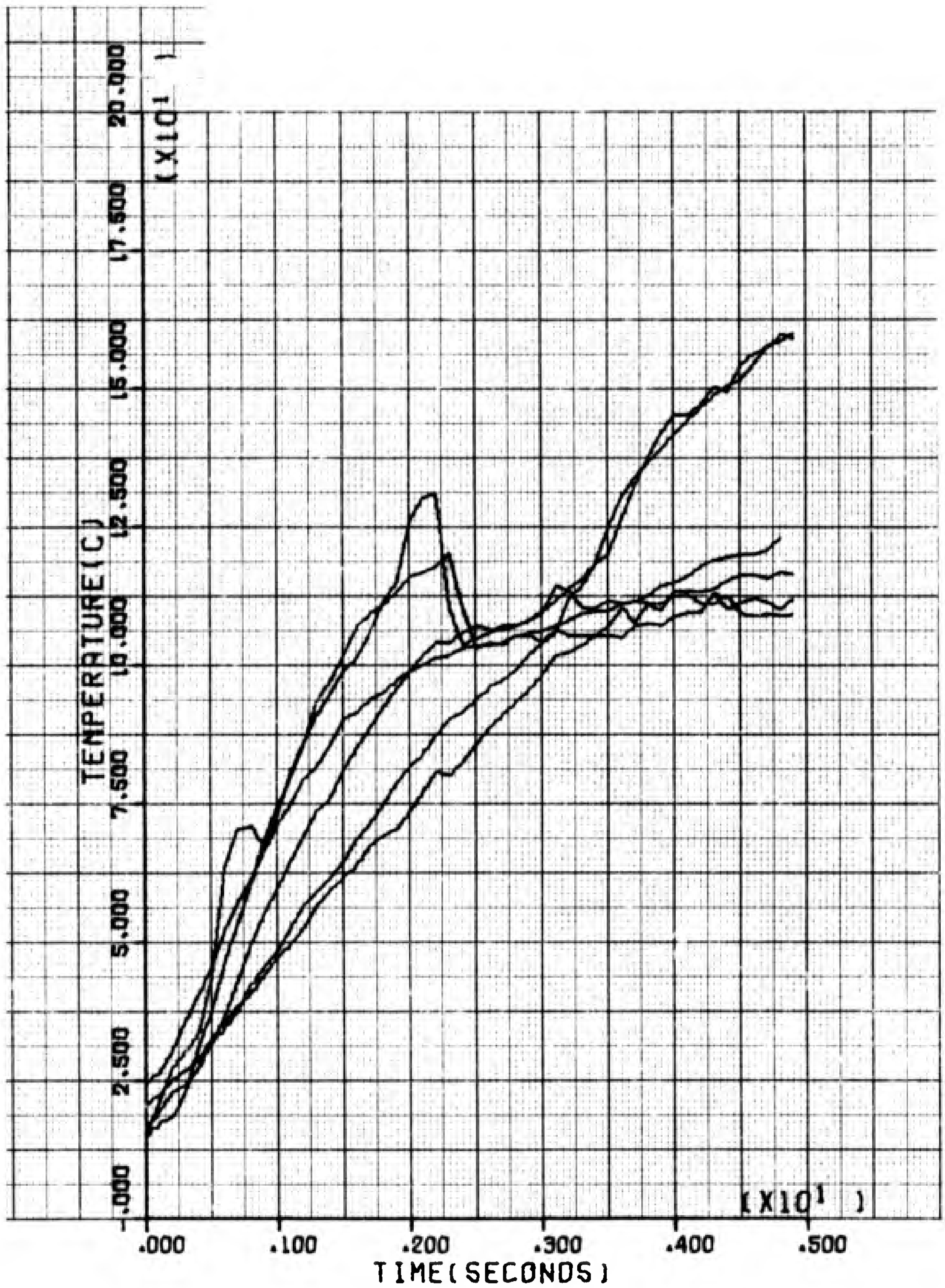

FIG. B-15 TEMPERATURE HISTORIES FOR STATION 4 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

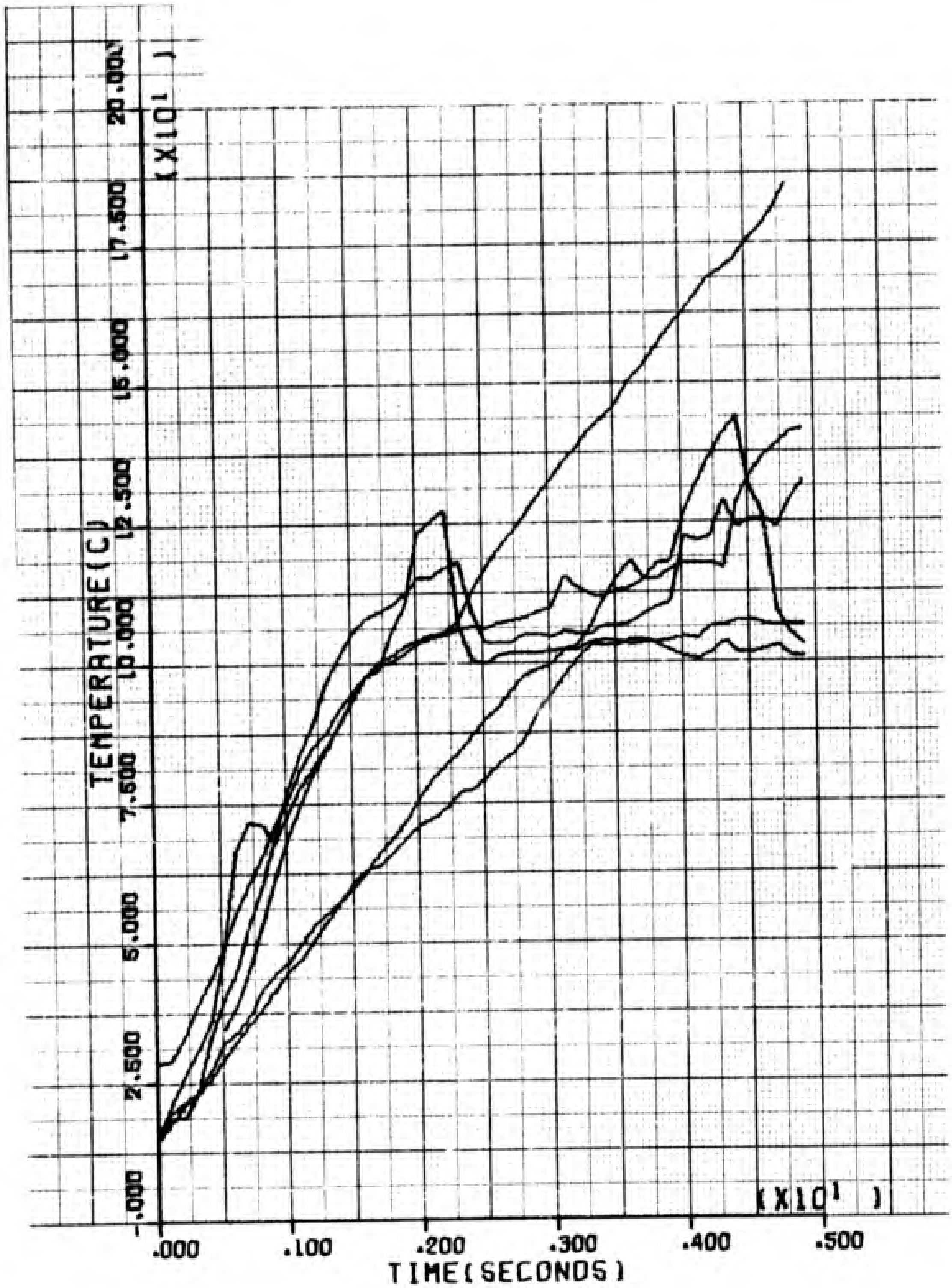

FIG. B-16 TEMPERATURE HISTORIES FOR STATION 5 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

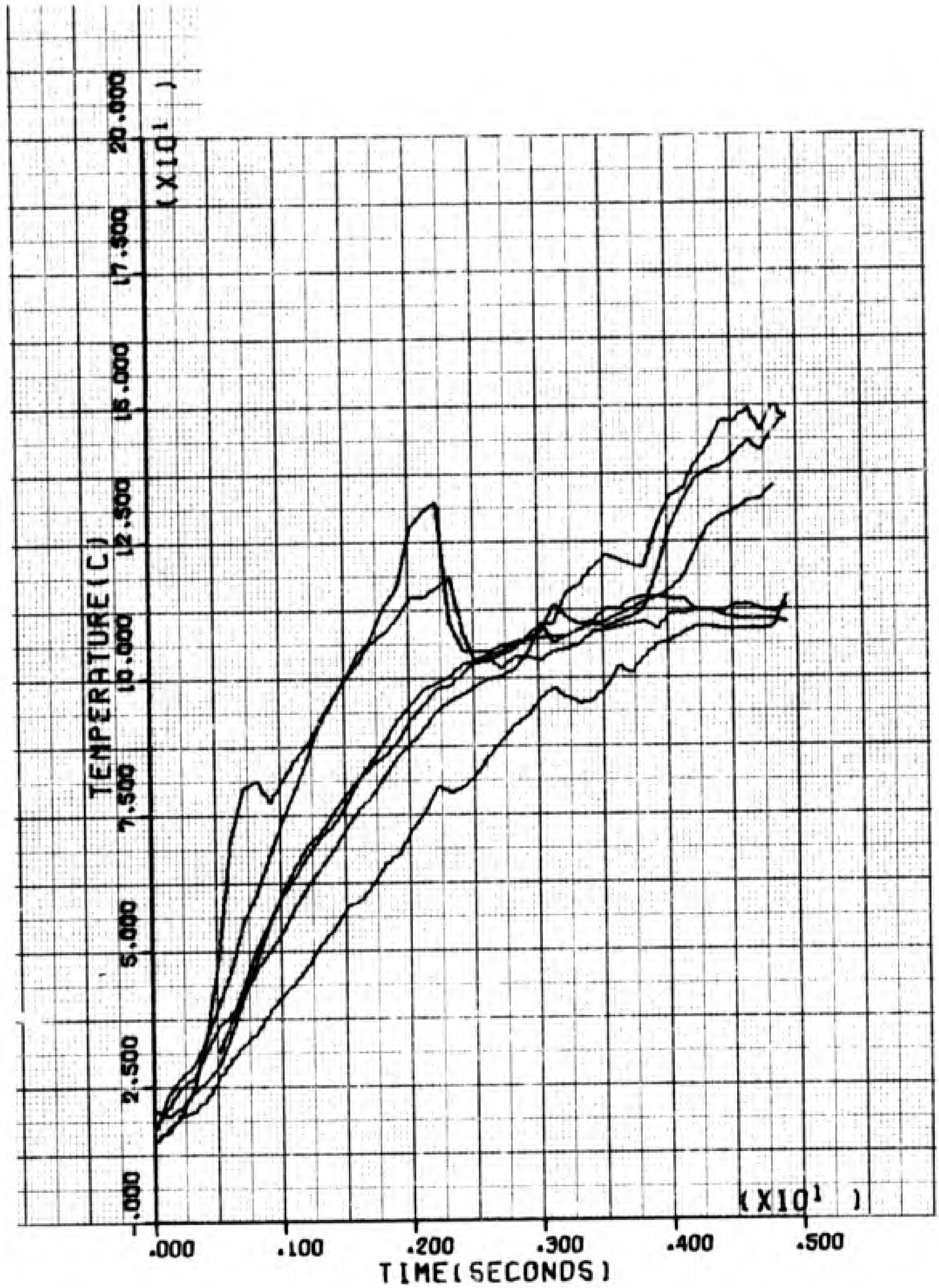

FIG. B-17 TEMPERATURE HISTORIES FOR STATION 6 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

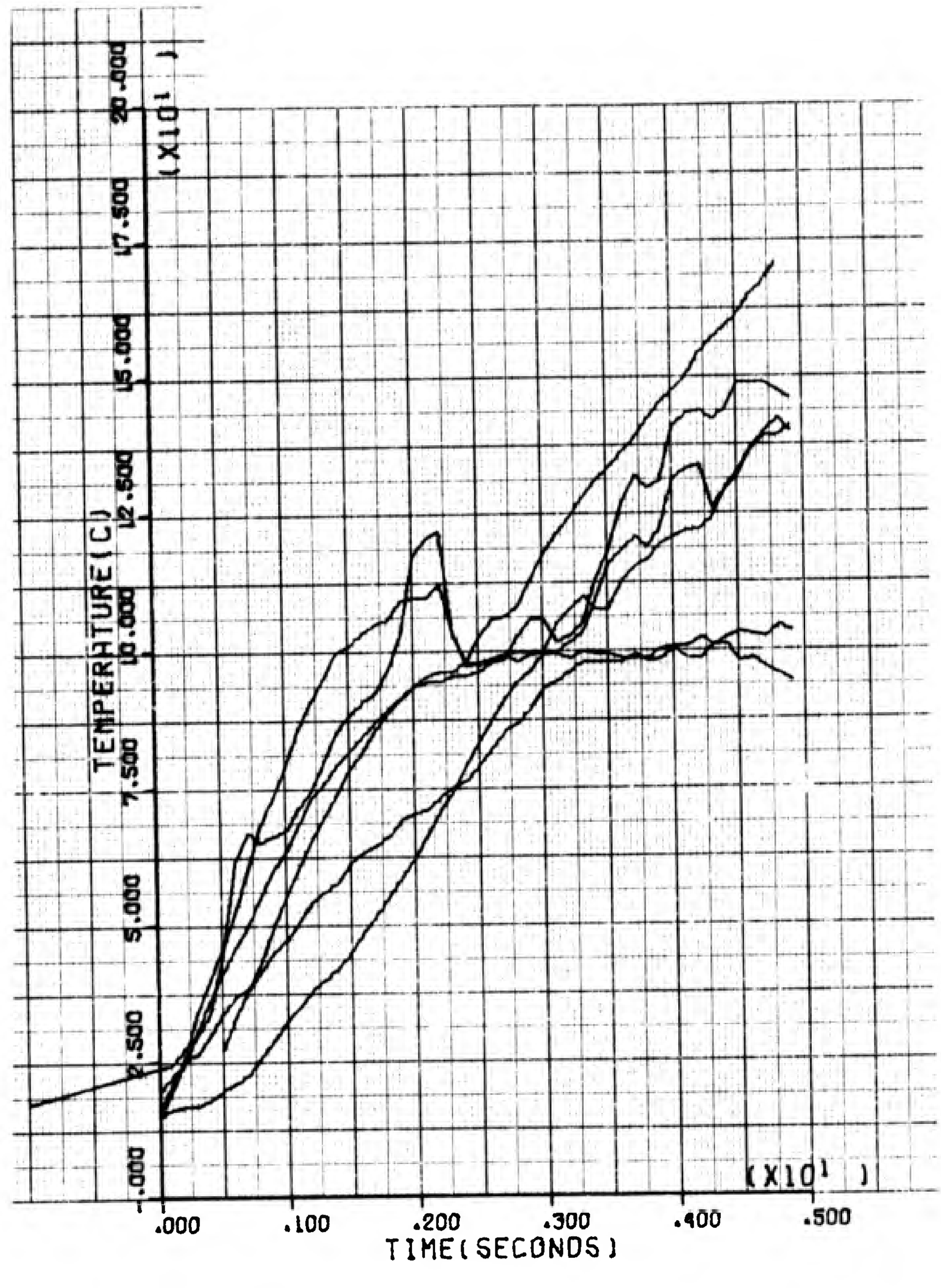

FIG. B-18 TEMPERATURE HISTORIES FOR STATION 7 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

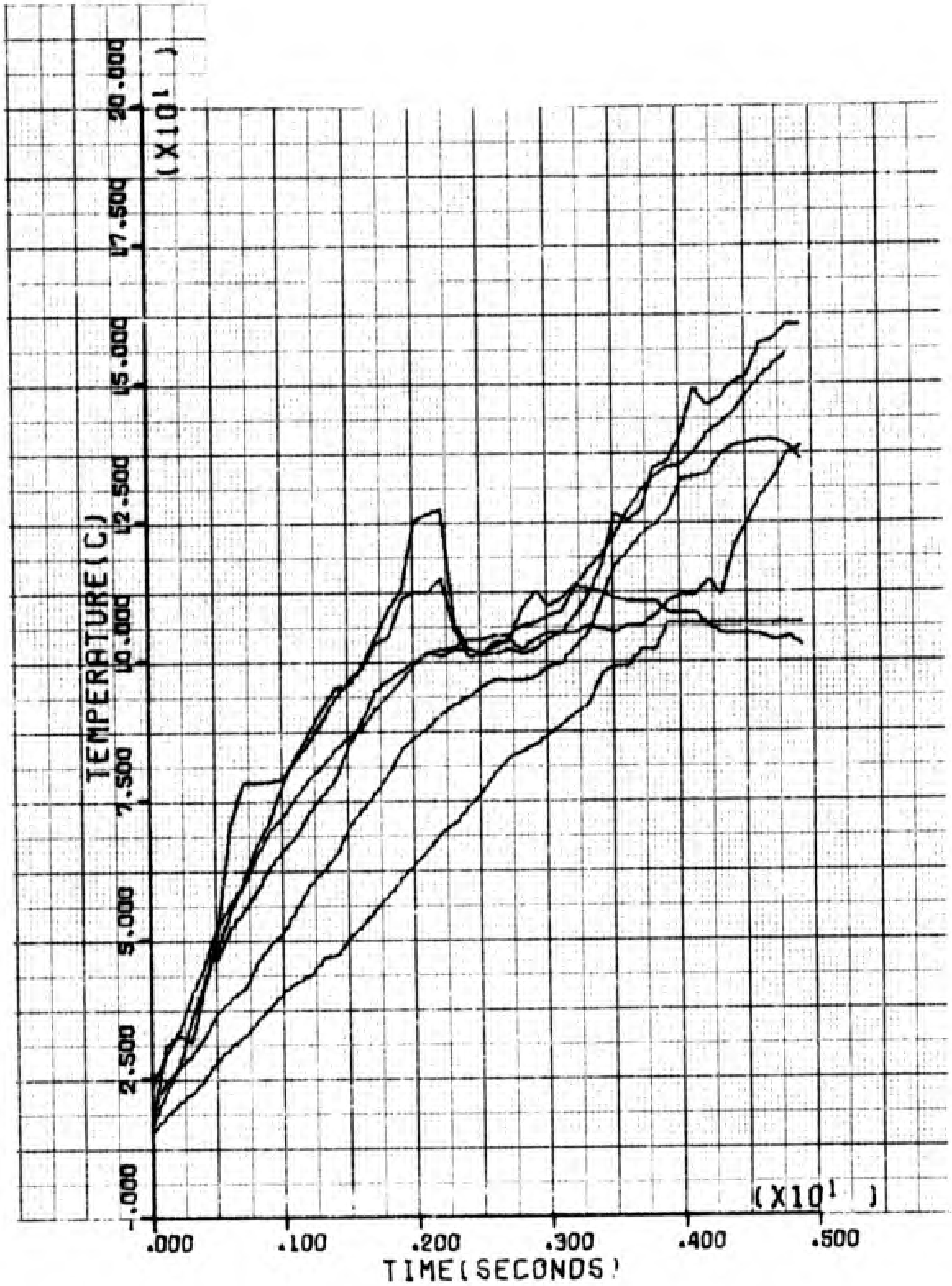

FIG. B-19 TEMPERATURE HISTORIES FOR STATION 8 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

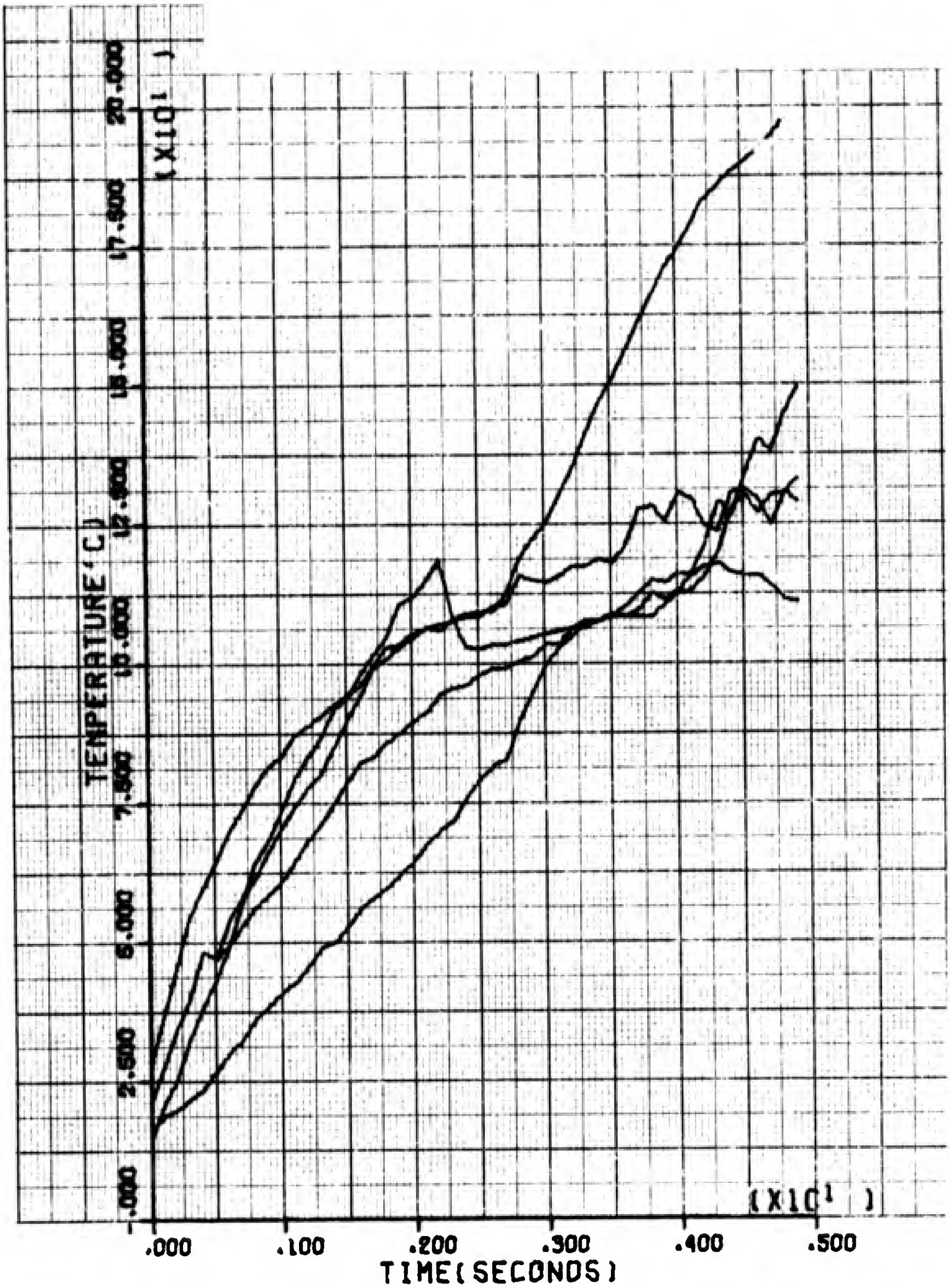

FIG. B-20 TEMPERATURE HISTORIES FOR STATION 10 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

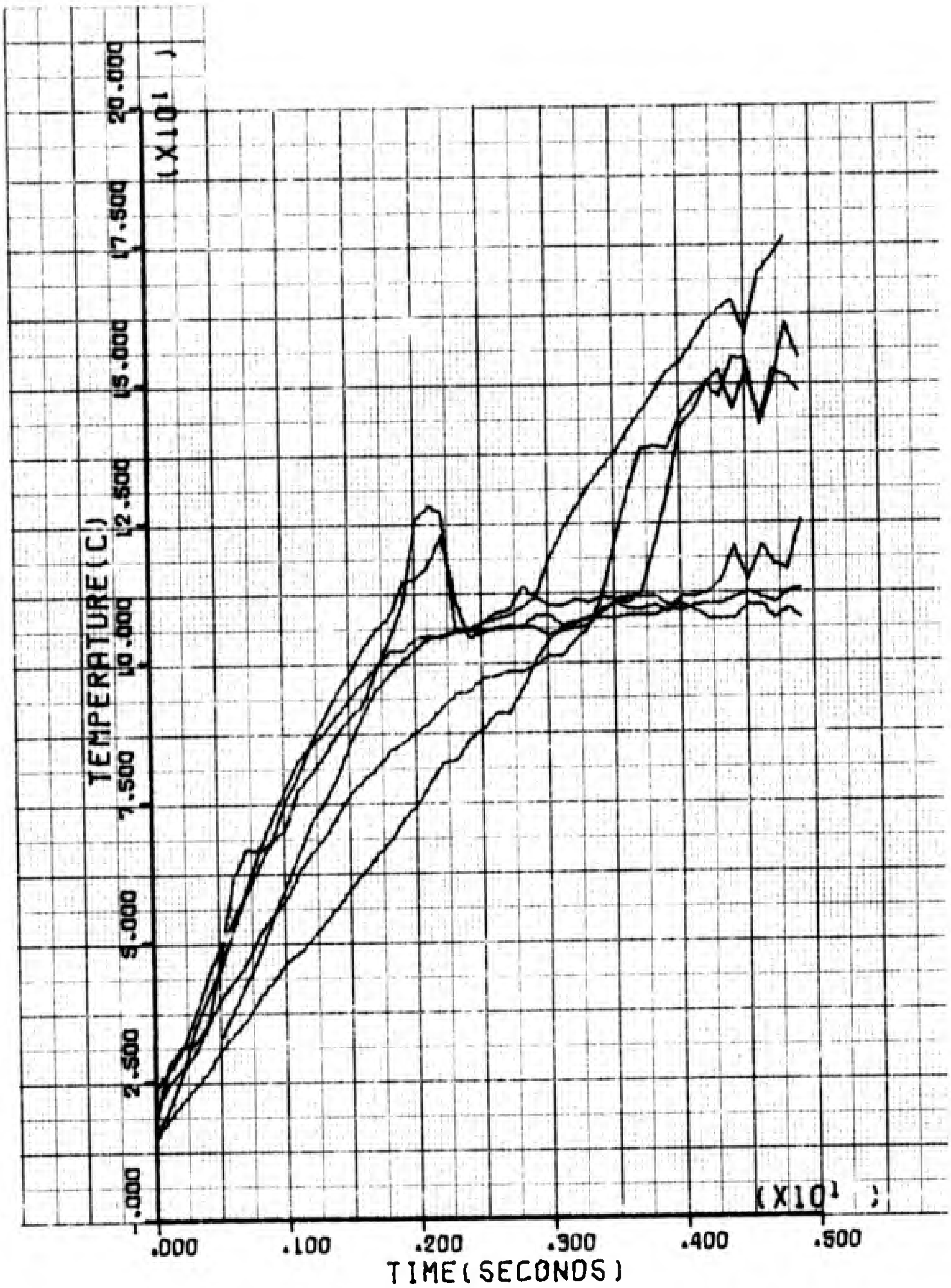

FIG. B-21 TEMPERATURE HISTORIES FOR STATION 11 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

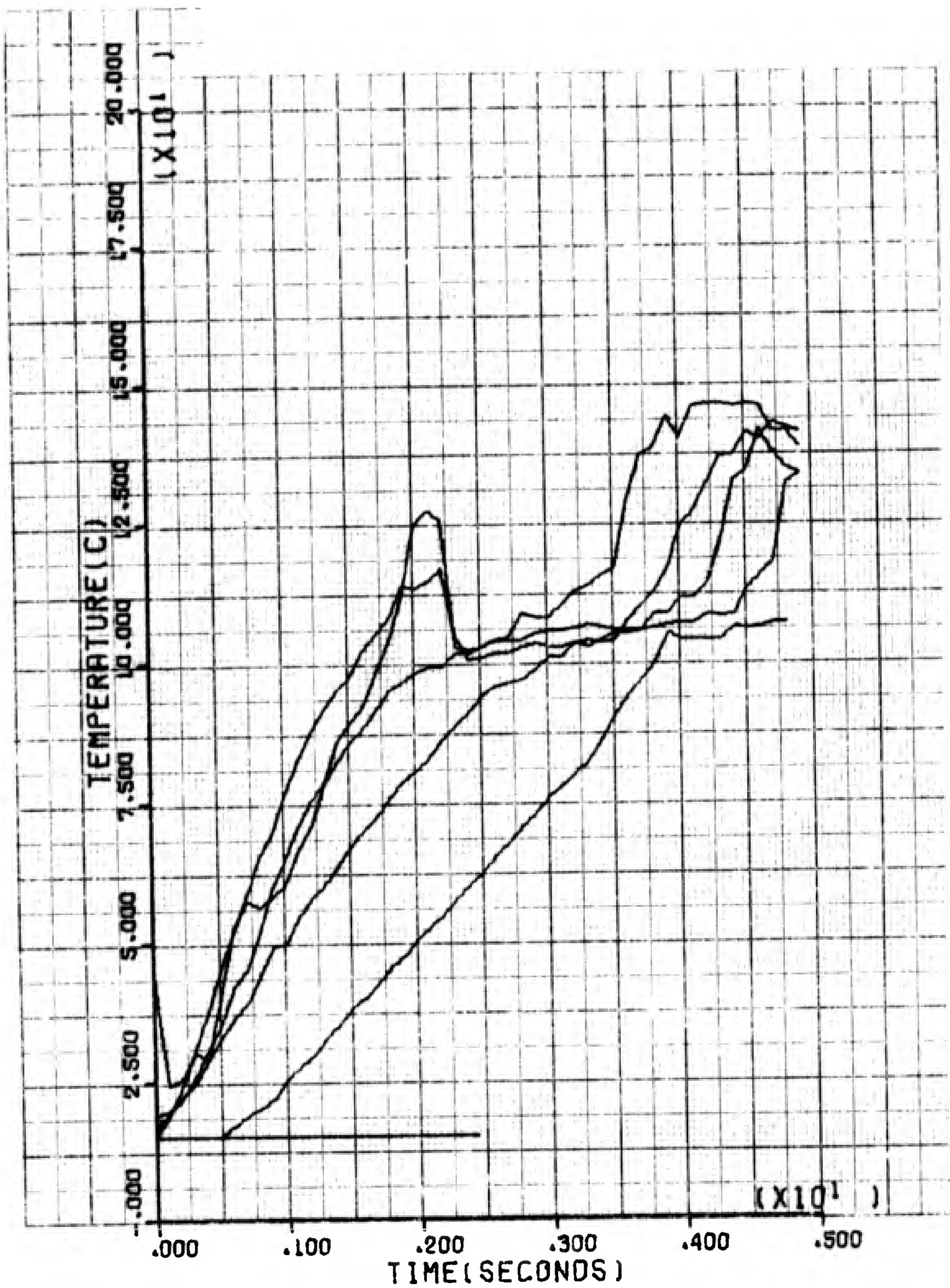

FIG. B-22 TEMPERATURE HISTORIES FOR STATION 12 (WATER-OFF WASHDOWN HEAT TRANSFER TESTS)

APPENDIX C

COMPUTER PROGRAMS TO CALCULATE PLATE TEMPERATURE HISTORIES FOR WASHDOWN AND SPRAY COOLING SYSTEMS

The heat transfer laws developed in this report for washdown cooling systems (equations (13) through (18)) and for spray cooling systems (equations (19) through (31)) were written into computer programs which calculate the temperature histories of plates being simultaneously heated and cooled. Heating can either be given arbitrarily, such as was done in specifying the heating rates supplied by burning a sheet of propellant, or the nuclear weapons thermal radiation pulse can be specified. The nuclear weapons thermal radiation pulse is given by incorporating the normalized dimensionless thermal pulse (Figure 1) into the programs. A thermal pulse for a particular problem is determined by providing values for Hmax and Tmax, the maximum thermal irradiance and the time at which the maximum thermal irradiance is reached. In each of these programs it was assumed that 80 percent of the incident thermal energy is absorbed by the plate surface. Numerical methods will be used to compute the plate temperatures in a stepwise manner.

Washdown System Program

The computer program for washdown systems simultaneously computes temperature rises in the plate being heated and in the water flowing over it. This program requires that the plate dimensions (length and thickness) and plate thermal properties be specified along with the water flow rate per unit width and the water average velocity. A time increment for successive calculation and the total time of heating are the only additional parameters required. Plate temperatures are computed by a system of finite difference equations which account for the heat absorption and conduction within the plate. For this system, the basic equation is:

$$\rho \; Cp \; V \frac{\partial T}{\partial T} = KA \frac{\partial T}{\partial x} + KA \frac{\partial T}{\partial y} + qin \; \Delta y \; \Delta T \qquad \text{(C-1)}$$

In equation (C-1), the heat transfer rate, qin, is the difference between the input heating and the heat removed by the water. The input heating is given by either the nuclear weapon thermal radiation pulse or is read indirectly for the case of propellant heating. The heat removed by the water is given by equations (13) through (18). The plate was divided into three nodal layers in the Y (thickness) direction and an arbitrary number of nodal points in the X (length direction). Explicit algebraic equations are written for each nodal point by expressing the partial derivatives of equation (C-1) in finite difference form. Water temperature rises are computed directly from equation (10) where the heat added, Δq, is given by the experimental heat transfer laws (equations (13) through (18)). The calculation of plate and water temperature rise must be matched by choosing plate and water elemental volumes of identical lengths. That is, the incremental length in the X direction of the plate is chosen to be the length of the water layer on the plate in the given time increment, ΔT. This is simply,

$$\Delta x = U\Delta T \qquad \text{(C-2)}$$

where U is the average water velocity (see Appendix A). One additional feature of the washdown computer program is the option of specifying an initial temperature gradient in the Y direction. It was previously assumed that 0.5 seconds would be required to activate a water cooling system upon sensing a nuclear weapon explosion. Hence, in calculating the temperature response of heating due to a nuclear weapons thermal radiation pulse, the washdown cooling calculation would begin with a temperature gradient caused by 0.5 seconds of unprotected heating. These temperature gradients were computed independently and read into the washdown computer program. A listing of the washdown program is given in Figure (C-1). The Fortran deck and instructions as to its use may be obtained from the author.

<u>Spray System Program</u>

The computer program for spray cooling systems computes the temperature rise only in the plate being heated. Experience gained in using the washdown computer program showed that heat conduction plays a very small role in determining temperature histories in relatively thin aluminum plates (i.e., plate thickness $\leq$ 0.25 inches). Hence, a simpler program neglecting heat conduction was written for spray cooling problems. Here, the plate temperature rise is computed directly from the basic heat transfer equation. This is,

$$\rho \cdot Cp \cdot V \cdot \frac{\partial T}{\partial T} = qin \ \Delta y \ \Delta T \qquad \text{(C-2)}$$

Equation (C-2) is written in finite difference form and solved stepwise by computing incremental temperature rises due to incremental additions of heat, qin, in a time interval, ΔT. Here the heating rate, qin, is also the difference between the input heating (i.e., the nuclear weapon thermal pulse) and the heat removed by the water (i.e., equations (19) through (31)). The spray cooling calculation begins at a specified initial plate temperature which is the average plate temperature after 0.5 seconds of unprotected heating. The additional input variables required to run this program are the plate thermal properties (except conductivity) and thickness and the time increment, ΔT. A listing of the spray system program is given in Figure C-2. The Fortran deck and instructions as to its use may be obtained from the authors. The normalized dimensionless thermal pulse is specified for either the washdown or spray cooling programs through a subroutine named PULSE. A listing of this subroutine is given in Figure C-3 and a Fortran deck may also be obtained from the authors.

```
      PROGRAM PLNWC(INPUT,OUTPUT)
      DIMENSION TEMP(200),TSAV(200),T(3,200),V(3,200),C(200),QIN(200)
     1,QOUT(200),X(200),TIME(1500)      ,HZ(110),TH(110),GB(1500)
     2,WEV(200),WOV(200)
    1 READ 101,W,E,XIN,TOTL,DTIM,TIMM,HCONV,IPRINT
  101 FORMAT(7F10.3,I10)
      IF(W) 100,100,2
    2 READ 102,RHO,CP,XK,HMAX,TMAX,TI,TJ,TK
  102 FORMAT(8F10.3)
      JL=0
      AA=-1.0
      BB=-1.0
      E=30.48*E
      DY=XIN*2.54/2.0
      DX=E*DTIM
      TOTL=2.54*TOTL
      L=(TOTL+0.001)/DX
      LA=L
      M=(TIMM+0.0001)/DTIM
      ALPHA=XK/(RHO*CP)
      A=ALPHA*DTIM/(DY*DY)
      B=(DY/DX)**2
      D=DY/XK
      JPRINT=IPRINT
      WGPM=W/0.139
      E=E/30.48
      PRINT 200,WGPM,E,XIN
  200 FORMAT(1H1,19H   WATER FLOW RATE=,F6.3,6HGPM/FT,5X,16H WATER VELOC
     1ITY=,F5.2,6HFT/SEC,10X,17H PLATE THICKNESS=,F6.4,6HINCHES)
      PRINT 210,RHO,CP,XK,DTIM
  210 FORMAT(1H0,11H   DENSITY=,F6.1,7HLBS/FT3,12H  SPEC.HEAT=,F5.2,8HBT
     1U/LB D,15H  CONDUCTIVITY=,F7.5,13HBTU/SEC-FT2-D,5X,11H TIME STEP=,
     2F6.4,7HSECONDS)
      PRINT 220,HCONV,HFILM
  220 FORMAT(1H0,26H   CONVECTIVE COEFFICIENT=,F4.3,13HCAL/SEC-CM2-C,
     120X,18H FILM COEFFICIENT=,F4.3,13HCAL/SEC-CM2-C)
      DO 5 J=1,L
      QB(J)=0.0
      WEV(J)=0.0
    5 WOV(J)=0.0
      CALL PULSE(HZ,TH,MH)
      MM=MH+1
      DO 69 I=2,MM
      K=MM+2-I
      HZ(K)=HZ(K-1)*HMAX
   69 TH(K)=TH(K-1)*TMAX
      TH(1)=0.0
      HZ(1)=0.0
```

FIG. C-1 COMPUTER PROGRAM FOR WASHDOWN COOLING

```
      DO 10 K=1,L
      T(1,K)=TI
      T(2 K)=TJ
   10 T(3,K)=TK
      TEMP(1)=20.0
      X(1)=0.0
      L1=L-1
      DO 12 J=1,L1
   12 X(J+1)=X(J)+DX/2.54
      TIME(1)=0.50
      DO 15 K=1,L
   15 TSAV(K)=20.0
      IK=1
      DO 50 I=1,M
      TAU=TIME(I)
      DO 80 K=1,MH
      IF(TAU.GE.TH(K).AND.TAU.LE.TH(K+1))73,80
   73 H=HZ(K)+(HZ(K+1)-HZ(K))/(TH(K+1)-TH(K))*(TAU-TH(K))
      GO TO 70
   80 CONTINUE
      H=HZ(MM)
   70 CONTINUE
      DO 20 K=1,L
   20 C(K)=D*H
      DO 30 K=1,5
      N=LA
      N1=N-1
      V(1,1)=T(1,1)+A*(8.*T(2,1)/3.+B*T(1,2)-T(1,1)*(8./3.+B)+2.*C(1))
      DO 125 J=2,N1
  125 V(1,J)=T(1,J)+2.*A*(4.*T(2,J)/3.+B*(T(1,J+1)+T(1,J-1))/2.-T(1,J)*
     1(4./3.+B)+C(J))
      V(1,N)=T(1,N)+A*(8.*T(2,N)/3.+B*T(1,N1)-T(1,N)*(8./3.+B)+2.*C(N))
      V(2,1)=T(2,1)+A*(4.*(T(3,1)+T(1,1))/3.+B*T(2,2)-T(2,1)*(8./3.+B))
      DO130 J=2,N1
  130 V(2,J)=T(2,J)+A*(4.*(T(1,J)+T(3,J))/3.+B*(T(2,J-1)+T(2,J+1))-T(2,J
     1)*(8./3.+2.*B))
      V(2,N)=T(2,N)+A*(4.*(T(3,N)+T(1,N))/3.+B*T(2,N1)-T(2,N)*(8./3.+B))
      V(3,1)=T(3,1)+A*(8.*T(2,1)/3.+B*T(3,2)-T(3,1)*(8./3.+B))
      DO 140 J=2,N1
  140 V(3,J)=T(3,J)+2.*A*(4.*T(2,J)/3.+B*(T(3,J-1)+T(3,J+1))/2.-T(3,J)
     1*(4./3.+B))
      V(3,N)=T(3,N)+A*(8.*T(2,N)/3.+B*T(3,N1)-T(3,N)*(8./3.+B))
      DO 25 J=1,L
      QOUT(J)=HCONV*(V(1,J)-TSAV(J))
      IF(AA.GT.0.0) GO TO 211
      IF(V(1,J)-100.) 24,21,21
   21 AA=1.0
      TT=TIME(I)
```

FIG. C-1 CONTINUED

```
211 IF(TEMP(J).GT.100.0) GO TO 23
    IF(V(1,J).LT.100.) GO TO 24
    Z=TIME(I)-TT
    IF(Z-0.3)22,22,71
 22 QOUT(J)=18.3*(Z+0.1)**0.375
    GO TO 24
 71 IF(Z-0.7)75,75,76
 75 QOUT(J)=6.85/(Z+0.1)**0.7
    GO TO 24
 76 QOUT(J)=8.0
    GO TO 24
 23 BB=1
    Z=TIME(I)-TT
    IF(Z-0.3) 60,60,61
 60 QOUT(J)=12.7*(Z+0.1)**0.375
    GO TO 24
 61 IF(Z-0.5) 65,65,66
 65 QOUT(J)=4.12/(Z+0.1)**0.85
    GO TO 24
 66 QOUT(J)=5.5
 24 QIN(J)=H-QOUT(J)
 25 C(J)=D*QIN(J)
 30 CONTINUE
    IF(BB.LT.1.0)GO TO 38
    DO 35 J=1,L
    IF(TEMP(J).LT.100.)GO TO 35
       QB(J)= (TEMP(J)-100.)*W*453.6*DTIM/540.
    TSAV(J)=100.0
 35 CONTINUE
    DO 36 J=2,L
 36 WEV(J)=WOV(J-1)+QB(J)
    DO 37 J=1,L
 37 WOV(J)=WEV(J)
 38 DO 40 J=1,3
    DO 40 K=1,L
 40 T(J,K)=V(J,K)
    DO 46 J=2,L
    TEMP(J)=TSAV(J-1)+QOUT(J-1)*DX*30.48/(1.0*W*453.6)
 46 CONTINUE
    IF(AA.GT.0.) GO TO 47
    XTIM=TIME(I)+DTIM
 47 TIME(I+1)=TIME(I)+DTIM
    IF(I.LT.IPRINT) GO TO 49
    IPRINT=IPRINT+JPRINT
    PRINT 201,TIME(I),H
201 FORMAT(1H0,6H TIME=,F6.2,7HSECONDS, 50X,15HWEAPON HEATING=,F7.2,11
   1HCAL/SEC-CM2)
    PRINT 202,(X(N),N=1,L,5)
```

FIG. C-1 CONTINUED

```
202 FORMAT(1H ,15H  DISTANCE(IN)=,15F7.2)
    PRINT 203,(TEMP(N),N=1,L,5)
203 FORMAT(1H ,15H WATER TEMP(C)=,15F7.2)
    PRINT 204,(T(1,N),N=1,L,5)
    PRINT 204,(T(2,N),N=1,L,5)
    PRINT 204,(T(3,N),N=1,L,5)
204 FORMAT(1H ,15H  WALL TEMP(C)=,15F7.2)
 49 DO 48 K=1,L
 48 TSAV(K)=TEMP(K)
 50 CONTINUE
    PRINT 400,XTIM
400 FORMAT(1H0,33H   GRAMS WATER BOILED STARTING AT,F5.3,7HSECONDS)
    PRINT 401,(WEV(KL),KL=1,L)
401 FORMAT(1H ,5X,16F8.3)
    GO TO 1
100 CONTINUE
    STOP
    END
```

FIG. C-1 CONTINUED

```
      PROGRAM SPCOOL(INPUT,OUTPUT)
      DIMENSION T(2000),HZ(110),TH(110)
    1 READ 101,TI,RCPX,DTAU,HMAX,TMAX
  101 FORMAT(5E10.3)
      IF(TI.LE.0.0)GO TO 199
      TAU=0.5
      QTT=2.6*HMAX*TMAX
      T(1)=TI
      CALL PULSE(HZ,TH,MH)
      MM=MH+1
      DO 5 I=2,MM
      K=MM+2-I
      HZ(K)=HZ(K-1)*HMAX
    5 TH(K)=TH(K-1)*TMAX
      TH(1)=0.0
      HZ(1)=0.0
      J=0
      PRINT 250,RCPX,QTT,HMAX,TMAX
  250 FORMAT(1H1,7H  RCPX=,F7.4,11H  Q(TOTAL)=,F7.2,7H  HMAX=,F7.2,
     17H  TMAX=,F4.2)
      PRINT 200
  200 FORMAT(1H0,28H  TIME(SEC)   TEMP(C)   QOUT,12X,1HH)
      DO 100 I=1,1000
      TAU=TAU+DTAU
      DO 8 K=1,MH
      IF(TAU.GE.TH(K).AND.TAU.LE.TH(K+1))7,8
    7 H=HZ(K)+(HZ(K+1)-HZ(K))/(TH(K+1)-TH(K))*(TAU-TH(K))
      GO TO 9
    8 CONTINUE
      H=HZ(MM)
    9 J=J+1
      IF(T(I).GT.105)GO TO 10
      QOUT=0.223*(T(I)-24.)
      T(I+1)=T(I)+(H-QOUT)*DTAU/RCPX
      GO TO 50
   10 IF(T(I).GT.168.)GO TO 20
      QOUT=8.36*(T(I)-100.)**0.5
      T(I+1)=T(I)+(H-QOUT)*DTAU/RCPX
      GO TO 50
   20 IF(T(I).GT.230.)GO TO 30
      QOUT=275.8-1.231*T(I)
      T(I+1)=T(I)+(H-QOUT)*DTAU/RCPX
      GO TO 50
   30 QOUT=.0180*(T(I)-24.)
      T(I+1)=T(I)+(H-QOUT)*DTAU/RCPX
   50 IF(J.LT.10)GO TO 100
      J=0
      PRINT 201,TAU,T(I),QOUT,H
  201 FORMAT(1H0,4F10.3)
  100 CONTINUE
      GO TO 1
  199 CONTINUE
      STOP
      END
```

FIG. C-2 COMPUTER PROGRAM FOR SPRAY COOLING

```
SUBROUTINE PULSE(H,T,I)
DIMENSION T(109),H(109)
I=109
T(001)=  .1
T(  2)=  .15
T(  3)=  .2
T(  4)=  .25
T(  5)=  .3
T(  6)=  .35
T(  7)=  .4
T(  8)=  .45
T(  9)=  .5
T(010)=  .55
T( 11)=  .6
T( 12)=  .65
T( 13)=  .7
T( 14)=  .75
T( 15)=  .8
T( 16)=  .85
T( 17)=  .9
T( 18)=  .95
T( 19)=01.
T( 20)= 1.05
T( 21)= 1.1
T( 22)= 1.15
T( 23)= 1.2
T( 24)= 1.25
T( 25)= 1.3
T( 26)= 1.35
T( 27)= 1.4
T( 28)= 1.45
T( 29)= 1.5
T( 30)= 1.55
T( 31)= 1.6
T( 32)= 1.65
T( 33)= 1.7
T( 34)= 1.75
T( 35)= 1.8
T( 36)= 1.85
T( 37)= 1.9
T( 38)= 1.95
T( 39)= 2.
T( 40)= 2.05
T( 41)= 2.1
T( 42)= 2.15
T( 43)= 2.2
T( 44)= 2.25
T( 45)= 2.3
T( 46)= 2.35
```

FIG. C-3 SUBROUTINE PULSE-NORMALIZED DIMENSIONLESS THERMAL RADIATION PULSE

```
T( 47)= 2.4
T( 48)= 2.45
T( 49)= 2.5
T( 50)= 2.55
T( 51)= 2.6
T( 52)= 2.65
T( 53)= 2.7
T( 54)= 2.75
T( 55)= 2.8
T( 56)= 2.9
T( 57)= 3.
T( 58)= 3.1
T( 59)= 3.2
T( 60)= 3.3
T( 61)= 3.4
T( 62)= 3.5
T( 63)= 3.6
T( 64)= 3.7
T( 65)= 3.8
T( 66)= 3.9
T( 67)= 4.
T( 68)= 4.1
T( 69)= 4.2
T( 70)= 4.3
T( 71)= 4.4
T( 72)= 4.5
T(73 )= 4.6
T( 74)= 4.7
T( 75)= 4.8
T( 76)= 4.9
T( 77)= 5.
T( 78)= 5.125
T( 79)= 5.25
T( 80)= 5.375
T( 81)= 5.5
T( 82)= 5.625
T( 83)= 5.75
T( 84)= 5.875
T( 85)= 6.
T( 86)= 6.125
T( 87)= 6.25
T( 88)= 6.375
T( 89)= 6.5
T( 90)= 6.625
T( 91)= 6.75
T( 92)= 6.875
T( 93)= 7.
T( 94)= 7.125
T( 95)= 7.25
```

FIG. C-3 CONTINUED

```
T( 96)= 7.375
T( 97)= 7.5
T( 98)= 7.625
T( 99)= 7.75
T(100)= 7.875
T(101)= 8.
T(102)= 9.
T(103)=10.
T(104)=15.
T(105)=20.
T(106)=30.
T(107)=40.
T(108)=50.
T(109)=90.
H(001)=  .026
H(002)=  .064
H(003)=  .105
H(  4)=  .155
H(  5)=  .209
H(  6)=  .27
H(  7)=  .357
H(  8)=  .444
H(  9)=  .54
H( 10)=  .633
H( 11)=  .717
H( 12)=  .784
H( 13)=  .84
H( 14)=  .89
H( 15)=  .926
H( 16)=  .96
H( 17)=  .988
H( 18)=  .997
H( 19)=01.
H( 20)=  .997
H( 21)=  .988
H( 22)=  .967
H( 23)=  .946
H( 24)=  .91
H( 25)=  .88
H( 26)=  .84
H( 27)=  .79
H( 28)=  .73
H( 29)=  .7
H( 30)=  .624
H( 31)=  .59
H( 32)=  .554
H( 33)=  .53
H( 34)=  .503
H( 35)=  .483
```

FIG. C-3 CONTINUED

```
H( 36)=  .463
H( 37)=  .445
H( 38)=  .43
H( 39)=  .415
H( 40)=  .4
H( 41)=  .387
H( 42)=  .37
H( 43)=  .36
H( 44)=  .35
H( 45)=  .34
H( 46)=  .33
H( 47)=  .32
H( 48)=  .31
H( 49)=  .3
H( 50)=  .292
H( 51)=  .284
H( 52)=  .276
H( 53)=  .27
H( 54)=  .263
H( 55)=  .256
H( 56)=  .244
H( 57)=  .23
H( 58)=  .219
H( 59)=  .209
H( 60)=  .199
H( 61)=  .19
H( 62)=  .182
H( 63)=  .174
H( 64)=  .165
H( 65)=  .157
H( 66)=  .15
H( 67)=  .143
H( 68)=  .138
H( 69)=  .133
H( 70)=  .127
H( 71)=  .122
H( 72)=  .118
H( 73)=  .114
H( 74)=  .11
H( 75)=  .107
H( 76)=  .103
H( 77)=  .1
H( 78)=  .097
H( 79)=  .092
H( 80)=  .089
H( 81)=  .085
H( 82)=  .082
H( 83)=  .08
H( 84)=  .077
```

FIG. C-3 CONTINUED

```
H( 85)=  .074
H( 86)=  .071
H( 87)=  .069
H( 88)=  .067
H( 89)=  .065
H( 90)=  .062
H( 91)=  .061
H( 92)=  .06
H( 93)=  .058
H( 94)=  .056
H( 95)=  .054
H( 96)=  .053
H( 97)=  .051
H( 98)=  .05
H( 99)=  .049
H(100)=  .048
H(101)=  .047
H(102)=  .039
H(103)=  .034
H(104)=  .0175
H(105)=  .011
H(106)=  .0057
H(107)=  .0036
H(108)=  .0025
H(109)=  .001
 RETURN
 END
```

FIG. C-3 CONTINUED