AD NUMBER
AD848782
LIMITATION CHANGES
TO: **Approved for public release; distribution is unlimited.**
FROM: **Distribution authorized to U.S. Gov't. agencies and their contractors; Administrative/Operational Use; 29 JUL 1969. Other requests shall be referred to Naval Air Development Center, Johnsville, PA.**
AUTHORITY
NADC per DTIC form 55

U.S. NAVAL AIR DEVELOPMENT CENTER

JOHNSVILLE, PENNSYLVANIA

Aeronautical Electronic and Electrical Laboratory

REPORT NO. NADC-EL-6043 29 JUL 1960

PHASE REPORT
REGENERATIVE HEAT SINKS FOR COOLING AIRBORNE ELECTRONIC EQUIPMENT

BUREAU OF NAVAL WEAPONS
TED Project No. ADC AV-44016

Reported by: A. R. Saltzman
A. R. Saltzman
Development Support Division

Approved by: E. R. Mullen
E. R. Mullen, Superintendent
Development Support Division

A. L. Habermacher
A. L. Habermacher
Deputy Director

SUMMARY

INTRODUCTION

One of the objectives of TED Project No. ADC AV-44016, reference (a), was to provide cooling specifications for airborne electronic equipment. Report No. NADC-EL-59113, reference (b), delineated the technical requirements upon which the specification should be developed. It recommended an analytical specification, the requirements of which should be based upon a unit operations philosophy.

The proposed program for the development of an analytical cooling specification was accepted by the Bureau of Naval Weapons (BUWEPS), reference (c), as being satisfactory and recommended that it be carried out as submitted in reference (b).

The technical elements of the proposed analytical specification will call out the rules that require the contractor to analytically prove that his cooling package design will provide reliable component temperature levels within the electrical equipment and maintain cooling hardware penalties such as weight and fuel consumed in overcoming momentum drag of the aircraft to a minimum. Reference (b) pointed out that all participants involved either in the preparation or review of the reports would require analytical design tools for determining whether temperature and penalty values are being kept to a minimum. Analytical tools of heat transfer for determining the performance characteristics of various types of unit operations of cooling as well as weight penalty factors are to be covered in a series of reports.

The first report on analytical tools deals with the unit operation of cooling known as regenerative heat sinks for airborne electronic equipment.

Regenerative heat sinks are a means of removing heat from a heat sink in electronic equipment during a low speed phase of a flight mission. During the high speed phase of the mission, the electronic equipment can maintain component temperatures at reliable levels due to the initial low temperature and heat capacity characteristics of the regenerator.

REPORT NO. NADC-EL-6043

SUMMARY OF RESULTS

The effectiveness of an analytical specification will be dependent upon the availability and utilization of analytical tools of design covering various unit operations of cooling airborne electronic equipment and methods of calculating penalty factors to the over-all aircraft system.

The packed bed regenerator is considered to be an important unit operation of equipment cooling design. Analytical theory and equations have been provided which can be used for the design of packed bed regenerators. Additional analyses required for different and more complicated types of boundary conditions will be covered in subsequent reports.

CONCLUSIONS AND RECOMMENDATIONS

Methods of analysis have been provided which allow calculations to be made for the size and weight of packed bed regenerator required to maintain temperature limits within a required range. Future reports will cover analytical tools for converting weight of packed bed regenerators to over-all penalty factors for aircraft systems.

It is recommended that this report be made available to navy contractors for information purposes and to be used as an aid in their technical development programs.

REPORT NO. NADC-EL-6043

TABLE OF CONTENTS

REPORT NO. NADC-EL-6043

LIST OF TABLES AND FIGURES

Aeronautical Electronic and Electrical Laboratory

REPORT NO. NADC-EL-6043

DISCUSSION

GENERAL

Electronic engineers for some time have been depending on the principle of regeneration, known better as "thermal inertia," for maintaining components of equipment packages at reliable levels during supersonic flight. Generally for certain aircraft systems, it was recognized that no cooling or an inadequate amount of refrigerated cooling air would be supplied to the electronic equipment during supersonic flight. The heat capacity of the mass of the electronic equipment had to be depended upon to absorb heat from electronic components for a given period of time.

In general, no attempt was made to calculate, prior to hardware development, temperatures of the equipment or whether penalty factors would be maintained at a minimum. Further, since analytical tools of design did not exist, it was not possible to analytically determine whether a regenerative heat sink or another type of cooling should have been used for a particular operation of a flight mission.

The discussion presented below provides theory and mathematical equations for calculating temperatures for cooling and heating packed bed regenerators.

THEORY OF REGENERATION OF PACKED BEDS

The temperatures of the air and packed bed of a regenerative heat exchanger depend on space and time. Only certain analytical procedures are considered in this paper. Some of the main publications of analytical solutions are based on the studies of Heiligenstaedt, Hausen, Nusselt, and Schumann, references (d), (e), (f), and (g). One of the most important of the analytical solutions is attributed to Schumann who solved the complicated theoretical heat-transfer rate equations for the simple case of an incompressible fluid passing through a bed of solid particles. The following equation

$$-(\partial T/\partial Y) = T - \theta = (\partial \theta/\partial Z)$$

describes the heating or cooling of a packed bed with the initial conditions

$$Y = 0; \quad T = T_0$$
$$Z = 0; \quad \theta = 0$$

by the passage of a fluid entering at a temperature T_0. T and θ are the temperatures of the fluid and packed bed, respectively, Y units from the inlet plane at a time given by the group Z, where

Y = $(hX\rho/CG)$, dimensionless

h = heat-transfer coefficient, BTU/hr ft^3 °F

X = distance from entrance of bed, ft

ρ = fluid density, lb/ft^3

C = fluid heat capacity per unit volume at constant pressure, BTU/ft^3 °F

G = fluid flow rate based on a superficial cross-sectional area of packed bed, lb/hr ft^2

Z = $\left\{h\left[t - (X/v)\right]\right\} ,/C_b(1 - f)$, dimensionless

t = time after start of fluid flow, hr

v = fluid flow per superficial cross-sectional area, ft^3/ft^2 hr

C_b = packed-bed heat capacity per unit volume, BTU/ft^3 °F

f = void fraction, dimensionless

In his analysis, Schumann assumed:

1. The particles were so small or had such a high thermal diffusivity that any given particle may be considered to be at a uniform temperature.

2. The resistance to transfer of heat by conduction in the solid itself was negligible.

3. The rate of heat transfer from the fluid to a solid at any point was proportional to the average difference in temperature between fluid and solid at any given point.

4. The change in volume of the fluid and solid due to the change in temperature was negligible.

5. The thermal constants were independent of temperature. With the initial temperature of the bed equal to zero, he produced the following results:

$$(\theta/T_0) = 1 - e^{-y-z} \sum_{n=0}^{\infty} Y^n M_n(YZ) = e^{-y-z} \sum_{n=1}^{\infty} Z^n M_n(YZ)$$

$$(T/T_0) = 1 - e^{-y-z} \sum_{n=1}^{\infty} Y^n M_n(YZ) = e^{-y-z} \sum_{n=0}^{\infty} Z^n M_n(YZ)$$

where $M_n(YZ)$ is the nth derivative,

$$\left[d^n M_0(YZ)\right] / \left[d(YZ)^n\right]$$

with respect to YZ, and

$$M_0(YZ) = 1 + YZ + \left[(YZ)^2/2!\right] + \left[(YZ)^3/3!\right] + \ldots .$$

Schumann provided numerical solutions in the range Y = 0 - 10, Z = 0 - 9½ in the form of graphs.

DEVELOPMENT OF EQUATIONS FOR COOLING OF PACKED BEDS

There appears to be little published literature for using numerical solutions with a digital computer to determine the time required to cool the packed bed with power being dissipated as heat into the packed bed. In the development of the method of finite differences, for cooling the packed bed, the annular cylinder containing the bed material is divided into a number of equally spaced intervals each of length ΔS, as shown in figure 1.

An energy balance is made on an elemental volume. This total energy includes a term H_s for rate of external heat addition (by the electronic equipment) to the packed bed per unit volume. If no heat is being added to the packed bed, as might occur with the electronic equipment in a standby position, the heat term H_s is dropped from equation (1).

$$H_sAdS + \rho C_p(\partial T/\partial t)AdS = G_oC_p(\partial T/\partial S)AdS + (1 - f)\rho_b C_b(\partial \theta/\partial t)AdS \qquad (1)$$

where

C_p = specific heat of air, BTU/lb °F
A = cross sectional area of bed, ft^2
dS = incremental height of bed, ft
ρ_b = specific heat of packed bed, lb/ft^3
G_o = flow rate per unit cross sectional area, lb/hr ft^2

Since $\rho C_p(\partial T/\partial t)AdS$ is considered negligible compared to H_sAdS or $G_oC_p(\ T/\partial s)AdS$, equation (1) becomes

$$H_sAdS - G_oC_p(\partial T/\partial S)AdS = (1 - f)\rho_b C_b(\partial \theta/\partial t)AdS \qquad (2)$$

Now it is to be noted that

$$(1 - f)\rho_b C_b(\partial \theta/\partial t)AdS = ha(T - \theta)AdS$$

and equation (2) becomes

$$H_s - G_oC_p(\partial T/\partial S) = ha(T - \theta) \qquad (3)$$

To transpose equation (3) in terms of finite differences, let

$$\partial T/\partial S = (T_{m+1,\ n} - T_{m,\ n})/\Delta S;$$

$$\partial \theta/\partial t = (\theta_{m+1,\ n} - \theta_{m,\ n})/\Delta t;$$

$$T_{m+(1/2),\ n} = (T_{m+1,\ n} + T_{m,\ n})/2$$

Now substitution of the finite difference terms into equation (3) results in

$$H_s - G_oC_p\,(T_{m+1,\,n} - T_{m,\,n})/\Delta S = ha(T_{m+(1/2),\,n} - \theta_{m+(1/2),\,n}) \qquad (4)$$

rearranging terms,

$$\frac{H_s}{ha} - \frac{G_oC_pT_{m+1,\,n}}{ha\Delta S} + \frac{G_oC_pT_{m,\,n}}{ha\Delta S} = (T_{m+(1/2),\,n} - \theta_{m+(1/2),\,n}) \qquad (5)$$

By using a modulus $M = (G_oC_p/ha\Delta S)$ and substituting into equation (5), there results

$$(H_s/ha) - MT_{m+1,\,n} + MT_{m,\,n} = (T_{m+(1/2),\,n} - \theta_{m+(1/2),\,n}) \qquad (6)$$

again rearranging terms gives

$$T_{m+1,\,n} - T_{m,\,n} = (1/M)\left[\theta_{m+(1/2),\,n} - T_{m+(1/2),\,n} + (H_s/ha)\right] \qquad (7)$$

substituting $T_{m+(1/2),\,n}$ gives

$$T_{m+1,\,n} = \left[1/(1 + 2M)\right] X \left[2\theta_{m+(1/2),\,n} + T_{m,\,n}\,(2M - 1) + K\right] \qquad (8)$$

where $K = 2H_s/ha$. It is to be noted that for the case of no power being dissipated as heat into the packed bed, the term $K = 2H_s/ha$ is taken out of equation (8). A finite energy balance is now made on both the air and the bed material contained in an elemental volume.

$$G_o C_p T_{m,n} - G_o C_p T_{m+1,n} = (\Delta S/\Delta t)(1-f)\rho_b C_b \theta_{m+(1/2),n+1} - (\Delta S/\Delta t)(1-f)\rho_b C_b \theta_{m+(1/2),n} \qquad (9)$$

Further manipulation gives

$$G_o C_p (T_{m,n} - T_{m+1,n}) = (\Delta S/\Delta t)(1-f)\rho_b C_b (\theta_{m+(1/2),n+1} - \theta_{m+(1/2),n}) \qquad (10)$$

Now let a modulus $N = \left[\Delta S\,(1-f)\rho_b C_b\right]/(G_o C_p \Delta t)$, equation (10) becomes

$$\theta_{m+(1/2),n+1} - \theta_{m+(1/2),n} = (1/N)(T_{m,n} - T_{m+1,n}) \qquad (11)$$

Equations (8) and (11) are now combined. For the case of no heat added to the bed, this becomes

$$\theta_{m+(1/2),n+1} = (1/2)\left\{\left[1 - 2M + (2/N)\right]T_{m,n} + \left[1 + 2M - (2/N)\right]T_{m+1,n}\right\} \qquad (12)$$

Equations (8) and (12) become the working equations for determining the temperature of a packing material and air with respect to time and position within the packed bed.

It is necessary to select suitable values of M and N consistent with the physical configuration under investigation. Examination of equation (8) for $M \geq (1/2)$ discloses that a negative effect on $T_{m,n}$ is avoided if $M = 1$. Suitable N values may be estimated if T_{m+1} is eliminated between equations (8) and (12). The value selected for M fixes ΔS; the value selected for N then fixes Δt when ΔS is fixed.

For example, when M and N = 1, then equations (8) and (12) become

$$T_{m+1,n} = (1/3)(2\theta_{m+(1/2),n} + T_{m,n}) \qquad (13)$$

$$\theta_{m+(1/2),\, n+1} = (1/2)(T_{m,\, n} + T_{m+1,\, n}) \tag{14}$$

A similar approach is used to cover the condition of power dissipated as heat going into the packed bed. Equation (13) becomes

$$T_{m+1,\, n} = (1/3)(2\theta_{m+(1/2),\, n} + T_{m,\, n} + K) \tag{15}$$

DEVELOPMENT OF EQUATIONS FOR HEATING OF PACKED BED

A discussion now follows on the packed bed of the electronic equipment previously cooled to a predetermined low-temperature value and now acting as an absorber of heat from the electronic components (the entire equipment being sealed and insulated from a high-temperature compartment environment caused by high Mach Number flight of the aircraft).

The interstices between the packings are filled with air. The bed and air for the calculations performed are initially at a uniform temperature. Heat from the electronic components is conducted into the container through the walls as indicated in figure 1. The container wall temperature is specified in terms of a given temperature-time history. It is also assumed, for any time, that the packing and air temperatures at any axial coordinate are constant over the cross section and, further, that the effects of axial conduction and free convection of air are minor and may be neglected. A heat balance on the system is

$$h_{wg}A_w(\theta_w - T) = \rho C_p f(dT/dt) + \rho_b C_b(1 - f)(d\theta/dt) \tag{16}$$

where

h_{wg} = film coefficient between wall and air, BTU/hr ft^2 °F
A_w = surface area of wall, ft^2
θ_w = temperature of wall, °F

Using the same assumptions, a heat balance may be set up for the flow of heat from air to bed resulting in the following equation:

$$ha(T - \theta) = \rho_b C_b (1 - f)(d\theta/dt) \qquad (17)$$

Equation (16) then becomes

$$h_{wg} A_w (\theta_w - T) = \rho C_p f(dT/dt) + ha(T - \theta) \qquad (18)$$

Equations (17) and (18) may be conveniently solved by means of an analog computer.

Computer Program

With the equations derived above, the program was divided into three phases specified below:

1. In phase I, for a given airflow rate and low air temperature occurring during the low-speed phase of the flight profile, calculations were made to determine the time required to lower the temperature of the packed bed to the temperature of the incoming air, with the equipment not operating.

2. Phase II was essentially the same as phase I with the exception that various amounts of power dissipated by the electronic package would be transferred into the area of the packing.

3. In phase III, the entire electronic package was considered thermally insulated from the aircraft compartment with no air entering or leaving the equipment. All of the heat dissipated by the electronic compartments was being taken by the packings. The entire electronics package was viewed as being thermally insulated from the high-temperature aircraft environment by means of insulating barriers. The packing materials were all considered to be spherical with a diameter of 1/8 inch. The materials were selected to cover a range of properties as shown in table I.

For phase III, the interstices between the packings are filled with air. The bed and air for the calculations performed are initially at a uniform temperature. Heat is introduced into the container wall. This is specified in terms of a given temperature-time history.

The physical configurations of the various packed beds are shown in table II. (All dimensions are in inches.)

The cooling air conditions are as specified in table III. Two sets of calculations were made, one with an initial packed-bed temperature of 86° F, and the other 140° F.

Equations (13) and (14) were used for phase I. A typical set of calculations for M = N = 1 is shown in figure 2. Figure 3 shows curves for phase II with power dissipated as heat into the packed bed.

Figure 4 shows typical curves obtained for phase III. In examination of the data shown in figure 4, it should be realized that two film coefficients of heat transfer are involved in the problem. There is the film coefficient of heat transfer between the concentric cylinder and the air designated as $h_{wg}A_w$ and between the air and the bed material ha. During actual environmental flight conditions these may vary, depending upon variables such as the cooling design of the electronic equipment, the type of aircraft, or its mission.

Data were obtained to cover a broad range of film coefficients. Figure 4 gives only the extreme ranges of film coefficients.

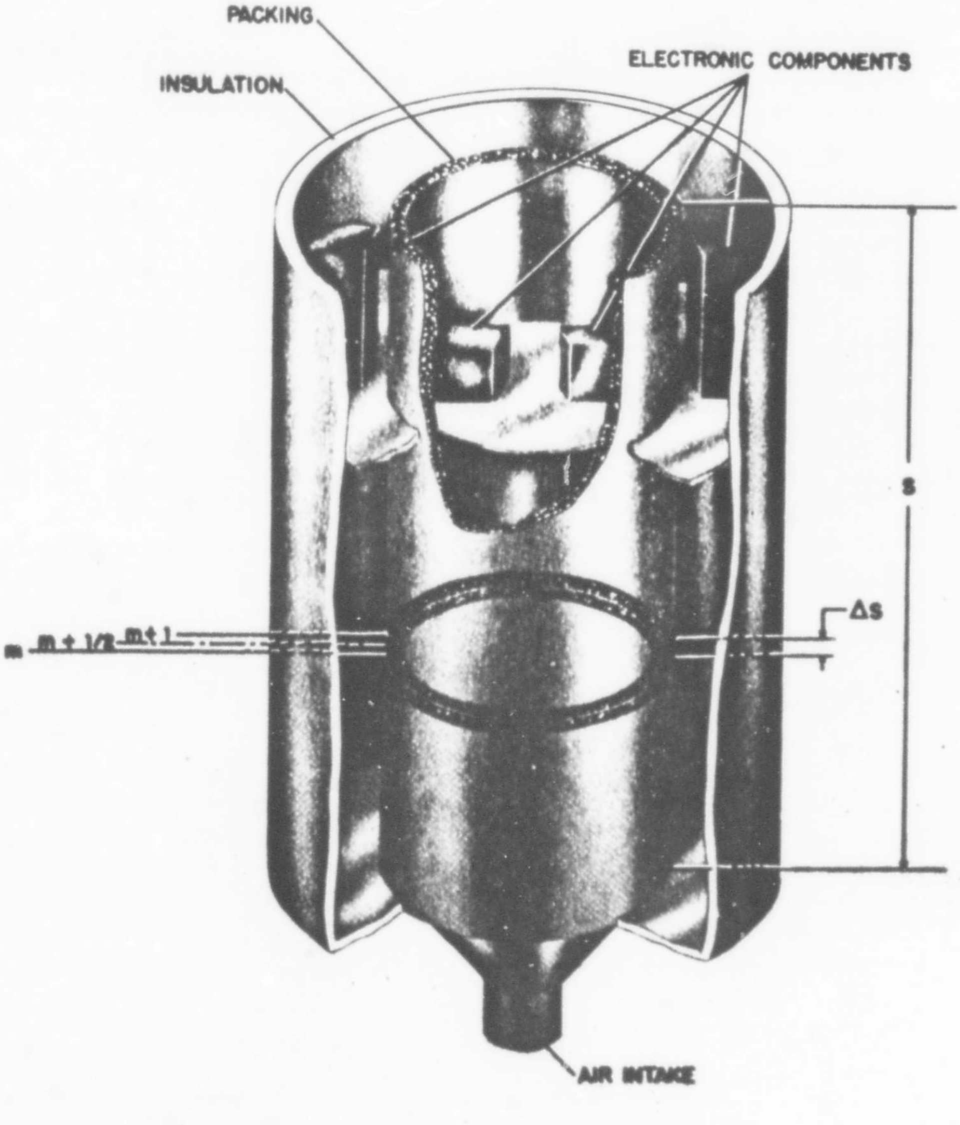

FIGURE 1 - Regenerative Packed Bed for Electronic Equipment Cooling

TABLE I

PACKING MATERIALS AND PROPERTIES

	Steel Balls	Glass Beads	Gravel	Foam
Density (lb/ft^3)	497	157	165	10
Specific heat (BTU/lb ° F)	0.106	0.20	0.65	0.206
Surface area/unit volume	334	334	334	334
Voids	0.420	0.420	0.420	0.420

TABLE II

PACKED BED DIMENSIONS
(IN)

Column Length	Inside Diameter of Concentric Cylinder	Outside Diameter of Concentric Cylinder
8		5
12		5
16		5
8		5.5
12		5.5
16		5.5
8		6
12		6
16		6

TABLE III

AIR SUPPLY CONDITIONS

Flow Rate (lb/hr)	Air Temperature (° F)
781	68
563	32
404	-4
266	-40
175	-63
108	-63

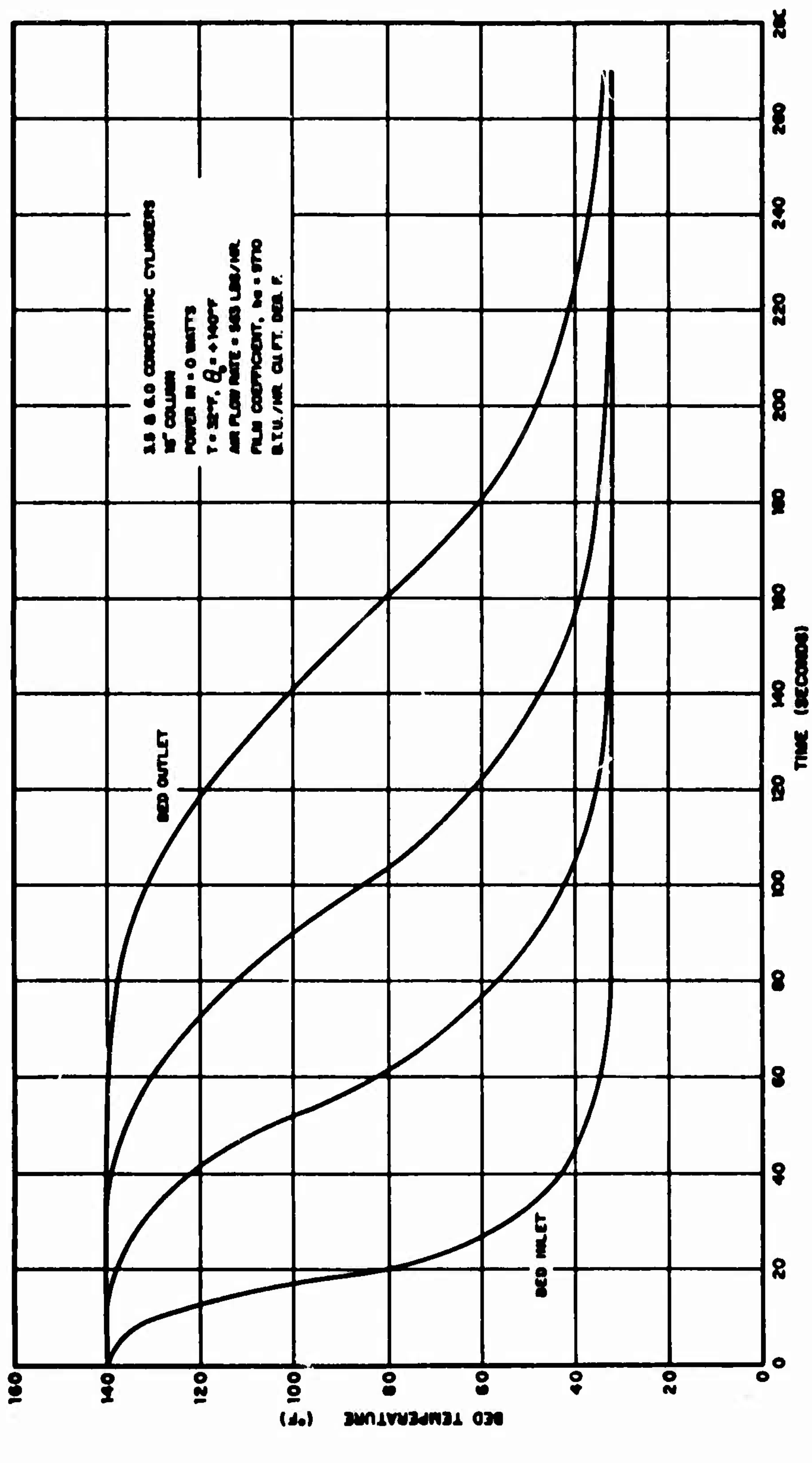

FIGURE 2 - Phase I Bed Temperature-Time History for Foam Material

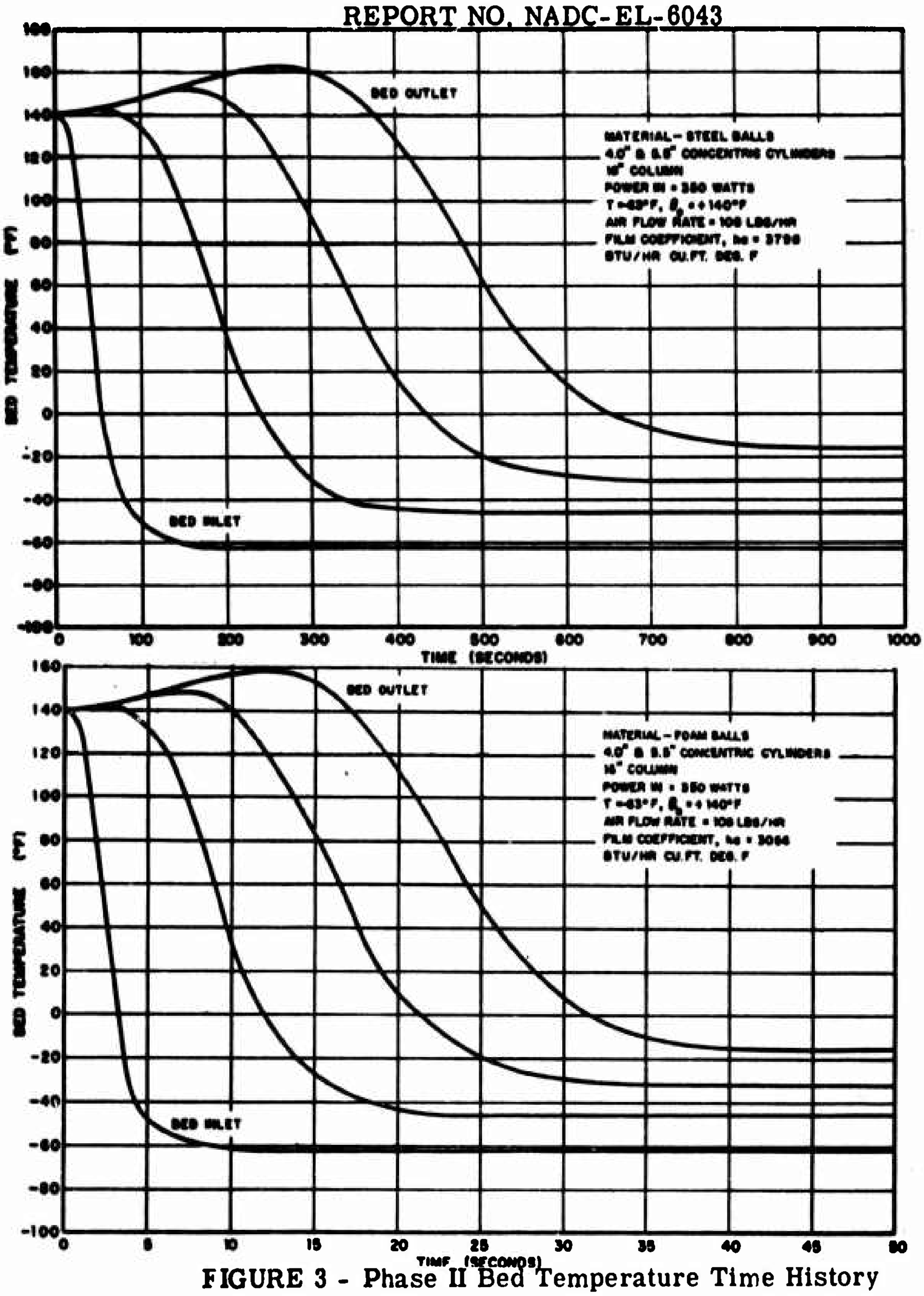

FIGURE 3 - Phase II Bed Temperature Time History

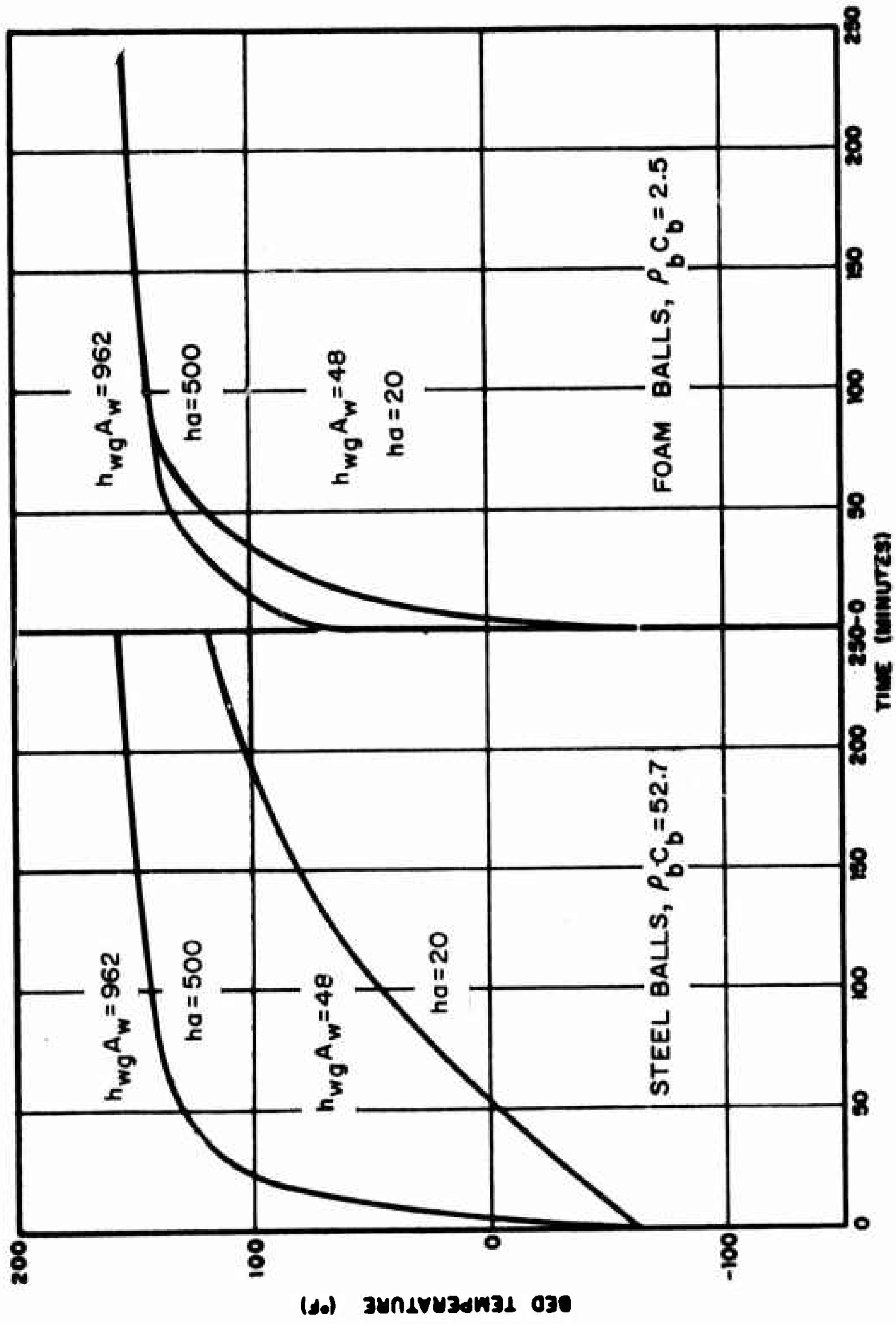

FIGURE 4 - Phase III Bed Temperature-Time History for Two Materials

REPORT NO. NADC-EL-6043

REFERENCES

(a) BUAER ltr Aer-AV-44272/12 of 25 Mar 1959
(b) Report No. NADC-EL-59113, Progress Report on Development of Cooling Specifications for Airborne Electronic Equipments
(c) BUWEPS ltr RAAV-4423/105 of 8 Apr 1960
(d) Heiligenstaedt, W., Arch. Eisenhuttenw.. Vol 2, p 217, 1928-1929
(e) Hausen, H., Z. Ver. Deut. Ing., Vol 73, p 431, 1929a
(f) Nusselt, W., Z. Ver. Deut. Ing., Vol 71, p 85, 1927
(g) Schumann, T. E., Journal, Franklin Institute, pp 208, 405