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CREW-SPACE HEAT TRANSFER FOR A MEDIUM COMBAT TANK

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ABSTRACT

The literature covering physiological effects due to changes in the quality and quantity of air supplied to enclosed living spaces is extensive. There is some agreement concerning the desirability for control of some or all the common environmental variables: temperature, pressure, humidity, velocity, distribution, and impurities. It is known that there are upper and lower limits of environmental conditions under which men are able to perform certain work, or to perform efficiently. A combat tank crew is subject to these limitations.

This report is concerned with the temperature and velocity parameters for the crew space of a medium combat tank subjected to extreme atmospheric temperatures. Effects due to design changes involving crew-space heat addition, heat subtraction, and air delivery rate are considered.

A minimum heat addition of 60,000 Btu per hour is required for crew-space heating. The use of crew-space insulation, reduction of air leakage, and superior warm-air distribution throughout the crew space adds to heating effectiveness. Temperature differentials across the uninsulated armor are usually less than 2°F. For no crew-space air leakage a minimum of 5-1/2 tons of refrigeration is required to maintain the crew-space ambient at 80°F for most adverse atmosphere conditions. One-quarter-inch thickness of insulation, thermal conductivity of 0.04 Btu per (ft)(hr)(°F), applied to the crew-space surfaces reduces the requirement to 2-1/2 tons.

An experimental method for finding the total leakage area of a buttoned-down combat tank is described. For the tank tested this total leakage area is 27 square inches. Air velocity in the crew space due to heater operation, except in the region 1 to 2 feet from the heater outlet, is less than 15 fpm. Air velocity in the crew space due to turret-ventilation-blower operation ranges from 35 to 450 fpm in the turret-basket region and from 13 to 45 fpm in the driver region. A description is given of electric analogs representing heat transfers for the crew space of medium combat tanks.

OBJECTIVE

The objective of this project was to determine the feasibility of ventilating and heating, or ventilating and cooling, the crew compartment of a medium combat tank in environments characterized by extreme temperatures. The upper limit of environmental temperature anticipated by the project is 120°F and the lower limit is -30°F. The possibility of controlling temperature in the main engine compartment was also to be studied, and an estimate was to be made of the practicability of attaining all the above objectives through the design of a universal package system which could be fitted to all tanks without the necessity for redesign.

NOMENCLATURE

A	area, square feet
$\overline{A}_2$	area of leakage openings in armor surrounding tank crew, square feet
Btu	British thermal unit
c	specific heat at constant pressure, Btu/(lb)(°F)
C_1	air leakage coefficient, $C_1 = 60\ A\sqrt{2g}$
C	armor leakage coefficient, $C = 294\ \overline{A}_2$
cfm	cubic feet per minute
e	vapor pressure, feet of water
F	gasoline consumption rate, U.S. gallons per hour
F'	emissivity factor
fpm	feet per minute
g	acceleration due to gravity, 32.2 ft/(sec^2) or 4.18×10^8 ft/(hr^2)
G	solar radiation, Btu/(ft^2)(hr)
h	film coefficient for heat transfer, Btu/(hr)(ft^2)(°F)
H	head, feet
H'	thermal heating value, Btu/lb
k	thermal conductivity, Btu/(hr)(ft^2)(°F/ft)
ℓ, L, X	a thickness dimension, feet
m'	flowrate, pounds per minute
m"	flowrate, pounds per hour
M	a dimensionless modulus
mph	miles per hour
n	some number, dimensionless
P	pressure, pounds per square foot
psf	pounds per squre foot
q	heat flowrate, Btu/hr
q'	heat flowrate, Btu/(hr)(ft^2)
Q'	rate of temperature change per time increment due to heat addition, °F/$\Delta\theta$
r	ratio of surface area to volume of a material, 1/ft
R	gas constant, ft-lb/(lb)(°R)
$\overline{R}$	thermal resistance, $\overline{R} = \ell/kA$
rpm	revolutions per minute
t	temperature, degrees Fahrenheit
t'	heater air inlet temperature, °F
t"	heater-delivery air temperature, °F
T	temperature, degrees Rankine
U	over-all heat-transfer coefficient, Btu/(hr)(ft^2)(°F)

NOMENCLATURE (continued

V	velocity, feet per second
w	specific weight, pounds per U.S. gallon
W	flowrate, cubic feet per minute
α	thermal diffusivity, A = $k/\rho c$, ft^2/hr
β	thermal coefficient of expansion , $1/°F$
$\Delta P'$	pressure difference, inches CH_3OH
ΔP	pressure difference, pounds per square foot
Δt	temperature difference, degrees Fahrenheit
ΔT	temperature difference, degrees Rankine
ΔX	thickness increment, feet
θ	time, hours
$\Delta\theta$	time increment, hours
ϵ	emissivity, dimensionless ratio
η	thermal efficiency, dimensionless
μ	absolute viscosity, lb/(hr)(ft)
ρ	density, pounds per cubic foot
σ	Stefan - Boltzmann constant, $\sigma = 0.173 \times 10^{-8}$ Btu/(hr)(ft^2)($°R^4$)

Subscript Notation

1, 2	reference points
a	ambient
A	atmosphere
b	human body
c	convection
cc	combined coefficient
cl	clothing
e	engine compartment
h	heater
i	interior
is	insulation surface
o	exterior
p	armor plate
r, R	radiation
t	crew compartment
v	evaporation
x	skin

Script notation in Appendix, used to emphasize fluid as opposed to solid conditions.

DEFINITIONS

Buttoned-down tank	The combat tank with all hatches closed
Crew space	Volume within the combat tank provided for crew
Equation of state	$P = \rho RT$
Exterior	Remote location from viewpoint of combat tank crew member in crew space
Heater thermal efficiency	Ratio of net heat delivered to warm air stream and heating value of gasoline used
Insulation	A material having low thermal conductivity
Interior	Near location from viewpoint of combat tank crew member in crew space
Radiation absorptivity	Fraction of impinging radiation which is absorbed directly
Radiation emissivity	Ratio of total radiating power of a nonblack surface to that of a black surface at same temperature
Radiation reflectivity	Ratio of reflected to incident radiation

INTRODUCTION

Provisions for operation of a combat tank in atmospheric temperatures ranging from 120° to -65°F should enable the tank personnel to act efficiently throughout this climatic temperature range.

This investigation deals with temperature and heat-flow parameters of concern to the tank personnel for the given range of atmospheric temperatures. An M-47 medium combat tank, Designation USA 30169766 (DE-29), was used for experimental study.

PERSONNEL HEATER CHARACTERISTICS

The M-47 combat tank is equipped with one personnel air heater of the combustion type which uses gasoline-air fuel. Air to be heated enters the heater from the crew space, is heated, and discharges from heater ducts in the region of the feet of the driver and the feet of the machine gunner. A fan is an integral part of the heater. Combustion air enters the heater from the crew space and exhaust products are dumped overboard.

The manufacturer's specification for the personnel heater unit installed in the vehicle tested is tabulated in Table I. The heating effects produced by this heater were investigated and the test data are given in Table IIa, b, and c.

For each test, crew-space temperatures were permitted to reach steady values by letting the unbuttoned vehicle stand. The vehicle then was buttoned down and crew-space air temperatures and velocities were recorded. The personnel heater then was started and operated until steady crew-space temperature readings were again obtained and recorded. Crew-space air velocities then were measured and recorded. Iron-constantan thermocouples and a recording potentiometer were used to measure air temperatures and a hot-wire anemometer was used to measure air velocities. The instruments are listed in Table XIV. All temperature and velocity measurements were made in the region of each crew member approximately at face level. A temperature differential not greater than 7°F and averaging less than 5°F was observed for the heated air discharged from the two outlet ducts and an average is reported.

TABLE I

SPECIFICATION OF MODEL 978 PERSONNEL HEATER
INSTALLED IN TEST VEHICLE

Fuel consumption (gal/hr approx.)	high heat	0.25
	low heat	0.125
Heat output (Btu/hr fresh air)	high heat	20,000
	low heat	10,000
Heat input (total Btu/hr)	high heat	30,000
	low heat	11,340
Air temperature rise, °F	high heat	220
Fresh heated air delivery (cfm)		85

Electrical consumption (watts)

	24v	12v	6v
Start	264	144	75
Run	60	60	60

Physical size, overall

Weight, pounds	23
Height, inches	9-3/4
Length, inches	18-1/2
Width, inches	8-1/4
Installation space, cubic feet	0.859
Starting time (normal voltage)	10 seconds
Starting time at -65°F	30-45 seconds

TABLE IIa

PERFORMANCE OF MODEL 978 HEATER IN TEST VEHICLE
(All heater operation on "high fire.")

	Test Number		
	A	B	C
Date	1-27-55	1-31-55	2-11-55
Duration, hours	1	1	3
Vehicle storage prior to test	In Bldg. 7	In Bldg. 7	Outdoors 6 hr prior to test
Atmosphere temperature, °F	38-53	65	16-19
Tank and gun position, facing	East	East	South
Hatch position for test	Buttoned	Buttoned	Buttoned
Auxiliary engine operation	Operating	Operating	Operating
Main engine operation	No	No	No
Avg. turret temp, °F, start	63	65	40
stop	79	84	57
rise	16	19	17
Avg. driver, machine gunner			
Temperature, °F, start	61	65	30
stop	110	103	76
rise	49	38	46
Avg. heater-duct outlet temp., °F	272	269	*
Avg. temp. rise, heater and duct, °F	166	167	*
Heat addition, Btu/hr	14,200	14,500	
U, Btu/(hr)(ft^2)(°F)	1.97	1.7	

*No reading due to thermocouple failure.

TABLE IIb

MEASURED AIR VELOCITIES IN CREW SPACE
(Tests A, B and C)

Location	Air Velocity, fpm	
	Prior to Operation of Heater	With Heater Operating
Driver	0 - 21 (8)	10 - 31 (17)*
Machine gunner	0 - 15 (7)	12 - 15 (13)
Loader	0 - 12 (3)	0 - 20 (12)
Commander	0 - 10 (3)	0 - 12 (5)
Gunner	0 - 14 (7)	14 - 40 (19)
Heater outlet duct, driver side		-1000

*Brackets enclose average of velocity readings.

TABLE IIc

MEASURED AIR VELOCITIES IN CREW SPACE DUE TO PERSONNEL HEATER OPERATION

Test Conditions:

Buttoned-down tank indoors with gun pointed forward between driver and machine-gunner cockpits; observer in crew space; observations made December 7, 1954, using hot-wire anemometer; all heater operation on "high fire."

	Location of Hot-Wire Anemometer Probe	Air Velocity, fpm (No Heater Operation)	Air Velocity, fpm (Heater Operation)
1	Air into heater, 12 inches upstream		180
2	Air from heater, 12 inches from outlet duct	15	360
3	6 inches from wall, 16 inches above loader's seat	12	14
4	Diametrically opposed to (3) above	13	15
5	16 inches above top edge of rebound pad, rear of turret basket	12	14
6	12 inches above shell rack forward of loader's seat	12	17
7	Symmetrical to (6), referred to longitudinal plane	15	16
8	6 inches above driver's seat	12	23
9	6 inches above machine gunner's seat	12	18

TABLE IId

MEASURED AIR VELOCITIES IN CREW SPACE DUE TO
TURRET-VENTILATION AIR-BLOWER OPERATION

Test Conditions:

Buttoned-down tank indoors with gun pointed forward between driver and machine-gunner cockpits; observer in crew space; observations made December 7, 1954, using hot-wire anemometer.

Location of Hot-Wire Anemometer Probe below Upper Interior surface, in.	Air Velocity, fpm			
	Beneath Loader's Hatch	Rear of Gun Breech	Beneath Commander's Hatch	Central Driver Cockpit
6	50-90	-75	-130	30-45
12	50-70	-200	150-200	-13
18	40-100	100-120	70-100	-25
24	-35	150-240	-150	-20
30	35-100	150-400	-150	-20
36	50-90	-450	50-60	
40		-300		
44		50-100		

Heater- and duct-installed performance is calculated from temperature data of Table IIa based on the assumption that the heated air delivery equals the manufacturer's rated 85 cfm. For Test A, Table IIa, the heat addition to the crew space due to heater operation is

$$q_h = 60\rho\, Wc\, \Delta t_h \qquad (1)$$

$$= 60(0.07)(85)(0.241)(272 - 110)$$

$$= 13{,}900 \text{ Btu/hr.}$$

With the assumption that the total surface area surrounding the crew space is 450 square feet (see pages 16-18 for discussion of this area) and noting that a steady-state temperature condition exists, the over-all heat-transfer coefficient U is found based on the average temperature of the turret-basket region:

$$q_t = U\, A_t(t_t - t_A); \qquad (2)$$

$$13{,}900 = U(450)(79 - 53) \text{ and}$$

$$U = 1.2 \text{ Btu/(hr)(ft}^2\text{)(°F).}$$

For Test A, Equation (2) may be rewritten

$$q_t = 1.2(450)(t_t - t_A) = 540(t_t - t_A). \qquad (2a)$$

Using data of Test B, Table IIa, similar calculations yield

$$q_h = 14{,}600 \text{ Btu/hr;}$$

$$U = 1.7 \text{ Btu/(hr)(ft}^2\text{)(°F), and}$$

$$q_t = 765(t_t - t_A).$$

A reasonable assumption is that for the turret-basket region of the uninsulated combat tank crew space, the heater-duct installation, and practical heat addition to the crew space, the values for U and UA fall in the range

$$1 < U < 2, \text{ and} \qquad (3)$$

$$450 < UA < 900,$$

and the average for Tests A and B, Table IIa, is approximately at the midpoint of the extremes.

If the average steady-state heated-air temperatures of the driver and machine-gunner region (listed in Table IIa) are used instead of the average turret-basket region heated-air temperatures, and the same assumptions are made, values of U approximately one half of those calculated are obtained and the corresponding temperature rises for the driver and machine-gunner region are approximately double those calculated.

The method is available for estimation of the crew-space temperature rise due to a given known heat addition. As an example, suppose consideration is given to installation of a 60,000-Btu-per-hour heater to replace the rated 20,000-Btu-per-hour heater tested in the vehicle. With the expression

$$q_t = 1.5(450)(t_t - t_A) = 675(t_t - t_A),$$

the predicted temperature rise of the turret-basket region due to operation of the 60,000-Btu-per-hour heater is

$$t_t - t_A = \frac{60,000}{675} = 89°F.$$

For the above assumption, if the heater installed in the test vehicle actually added the rated 20,000 Btu per hr to the heated-air stream, the predicted temperature rise in the turret-basket region of the crew space approaches 30°F.

Crew-space temperature rises due to a fixed heat addition, with the test combat tank located in a fixed atmosphere, vary with heater location, the number, location, and direction of warm-air discharge ducts, air discharge velocity from the duct, and the location and area of air-leakage openings in the armor surrounding the crew space.

The method of warm-air distribution in the crew space is an important heating factor. The difference between the average air temperature of the turret-basket region and the average air temperature of the driver and machine-gunner region (Table IIa) indicates that the heater-duct installation tested does not produce uniform warm-air distribution throughout the crew space. This is not surprising when it is noted (Table IIb) that at face level the average warm-air motion when the test heater is in operation is always less than 20 fpm and usually less than 15 fpm. By inference from the quotation which follows, which refers to air distribution for an occupied zone, an average warm-air movement less than 15 fpm is not satisfactory:

> "With reference to permissible room air motion it is not possible to establish a specific standard covering the entire complex problem of air distribution. Velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm will disturb loose paper sheets on desks and may result in a sensation of draft. Air velocities of 25 to 35 fpm in the occupied zone are most satisfactory, but air motion of 20 to 50 fpm will usually be acceptable, particularly when the lower part of this range of velocity is used in cooling applications, and the higher values on heating jobs. In any case, it is certain that the effect of room air motion on comfort or discomfort depends on air temperature and direction as well as on velocity."[1]

Although unit-heater warm-air discharge velocities usually vary from about 400 to 2,500 fpm, depending on a number of factors including the distance (blow) the discharged air is to be projected, the presence of obstacles to air motion in the crew space of a combat tank leads to the assumption that an increase in heater warm-air discharge velocity will not solve the air-distribution problem. Provision of additional well-located warm-air outlets is the solution.

Equation (1) is useful in the comparison of heater parameters when it is written

$$t'' = t' + \frac{q_h}{60\rho Wc} . \tag{1a}$$

Equation (1a) permits calculation of heater-air discharge temperature for fixed heater-air entry temperature, fixed heat addition to the fresh-air stream through the heater, and known fresh-air flowrate. Data from calculations for a range of heat additions, air flowrates and for fixed heater ambient air-entry temperatures of 40°, 0°, -40°, and -70°F are listed in Table III and shown in Figures 1, 2, 3, and 4. These data illustrate the sensitive relation between heater-air delivery temperature and heater-air flow for low-air flows.

Figures 1 through 4 are useful only if the air-discharge temperature for a heater of given capacity is known. The heater manufacturer usually lists the air-temperature rise for a given heat addition to the fresh-air stream corresponding to fixed fresh-air-stream flowrate. Usually, the specification does not include either the heater entrance or discharge temperatures of the fresh-air stream. These air temperatures may be found by use of Equation (1) and the state equation where ρ is evaluated for the heater entrance condition

$$\frac{1}{\rho} = 60c\, W\left(\frac{\Delta t_h}{q_h}\right), \tag{1}$$

and with the state equation

$$t' = \left(\frac{60cP}{R}\right) W\left(\frac{\Delta t_h}{q_h}\right) - 460.$$

By approximation, the air pressure at the heater entrance is taken as standard atmosphere pressure (2116 psf) and for air the previous expression reduces to

$$t' = 574\, W\left(\frac{\Delta t_h}{q_h}\right) - 460. \tag{4}$$

also

$$t'' = \Delta t_h + t'. \tag{5}$$

Two objections are anticipated to the selection of a combat tank heater installation having higher performance than that of the heater installation tested. The objections are the physical size of the heater and its fuel consumption rate. The first objection is a design consideration. Fuel consumption rate, however, for a combustion-type air heater may be estimated by

$$F = \frac{q_h}{\eta\, w\, H'} \tag{6}$$

TABLE III

CALCULATED PERSONNEL HEATER-DELIVERY AIR TEMPERATURES
VERSUS HEATER-AIR FLOWRATE AND THERMAL CAPACITY

Heater-Air Entry Temp., °F	Fresh-Air Flow through Heater cfm	Fresh-Air Flow through Heater No./hr	Heater-Air Discharge Temperatures, °F, for Fixed Heat Addition, Btu per Hour.								
			20,000	30,000	40,000	50,000	60,000	70,000	80,000	90,000	100,000
+40	100	477	214	301	388	475					
	200	954	127	170	214	258	302	344	388	431	
	400	1908	83	105	127	149	170	192	213	236	247
	600	2862	69	83	98	112	127	141	156	171	185
0	100	516	160	241	321	401	482				
	200	1032	80	120	161	201	241	281	321	361	401
	400	2064	40	60	80	100	120	140	160	180	200
	600	3096	27	40	53	67	80	94	107	121	134
-40	100	567	106	179	252	325	396				
	200	1134	33	70	106	143	180	216	253	289	325
	400	2268	-4	15	33	51	70	88	106	124	142
	600	3402	-16	-3	9	21	33	45	58	70	82
-70	100	611	66	134	202	270	338	395			
	200	1222	-2	32	66	100	134	168	202	236	270
	400	2443	-36	-19	-2	15	32	49	66	83	100
	600	3665	-47	-36	-25	-13	-2	9	20	32	43

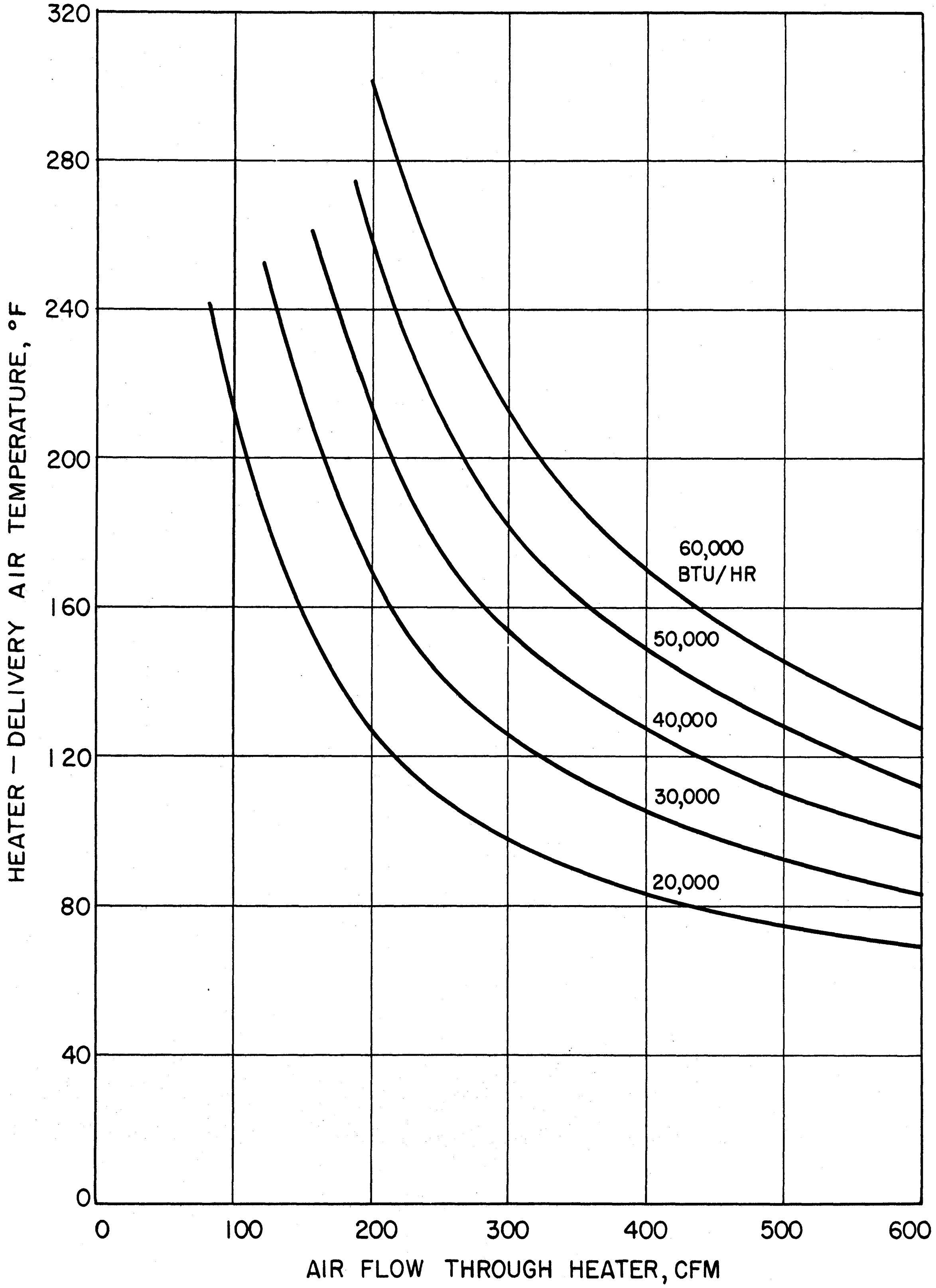


Figure 1. Calculated personnel heater-delivery air temperature vs heater air capacity and heat-release rate; heater ambient air temperature, +40°F.

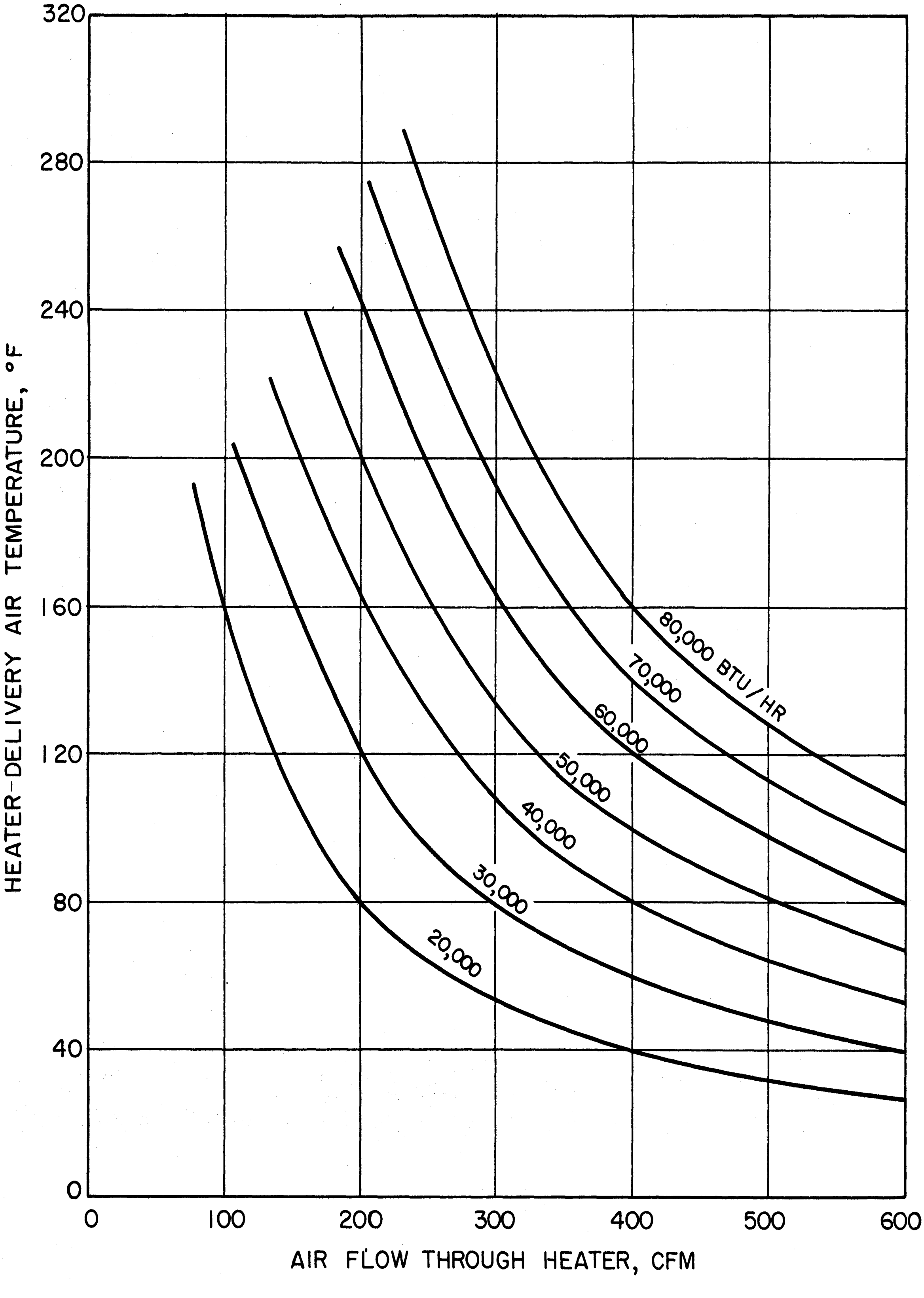


Figure 2. Calculated personnel heater-delivery air temperature vs heater air capacity and heat-release rate; heater ambient air temperature, 0°F.

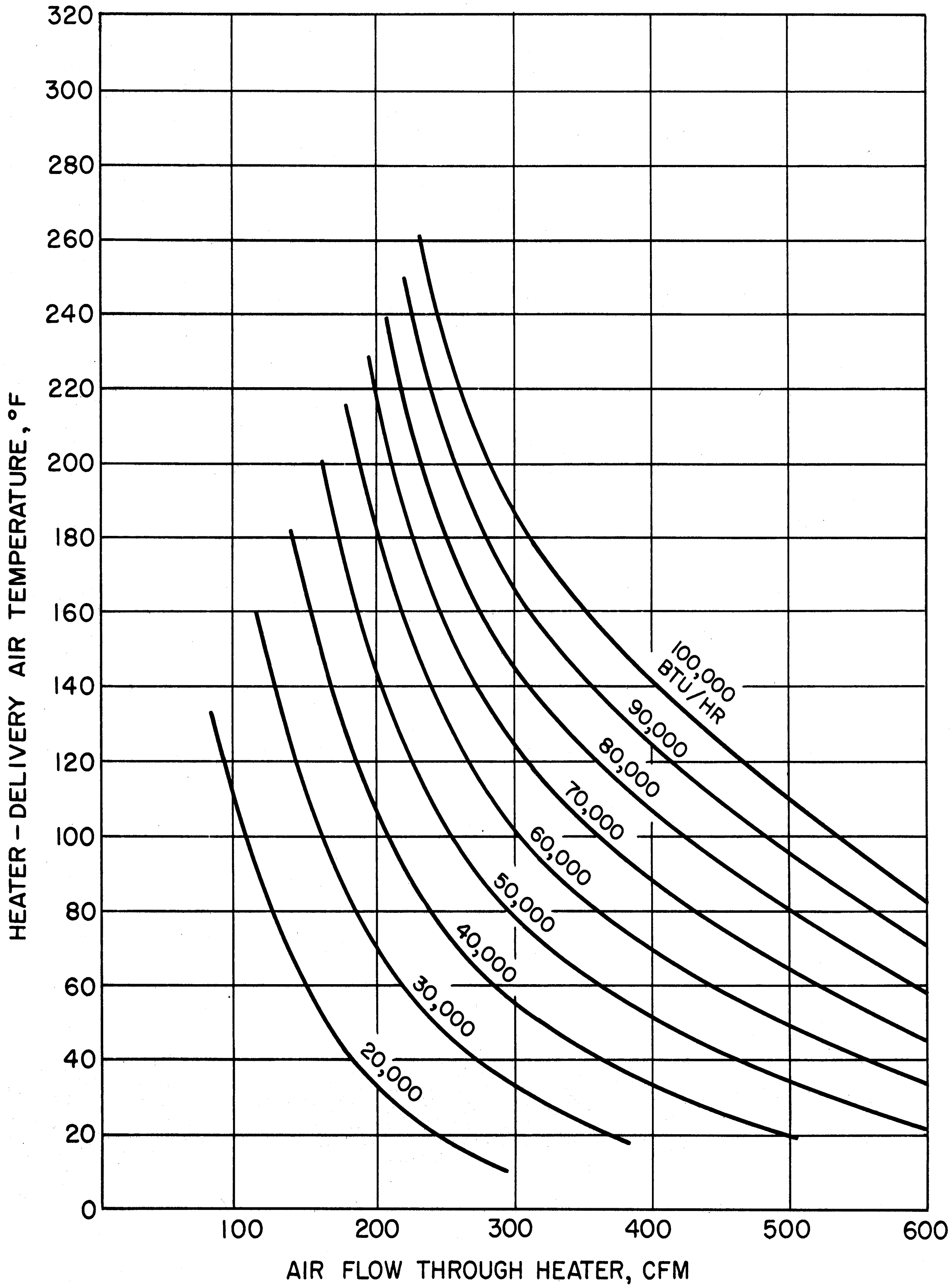


Figure 3. Calculated personnel heater-delivery air temperature vs heater air capacity and heat-release rate; heater ambient air temperature, -40°F.

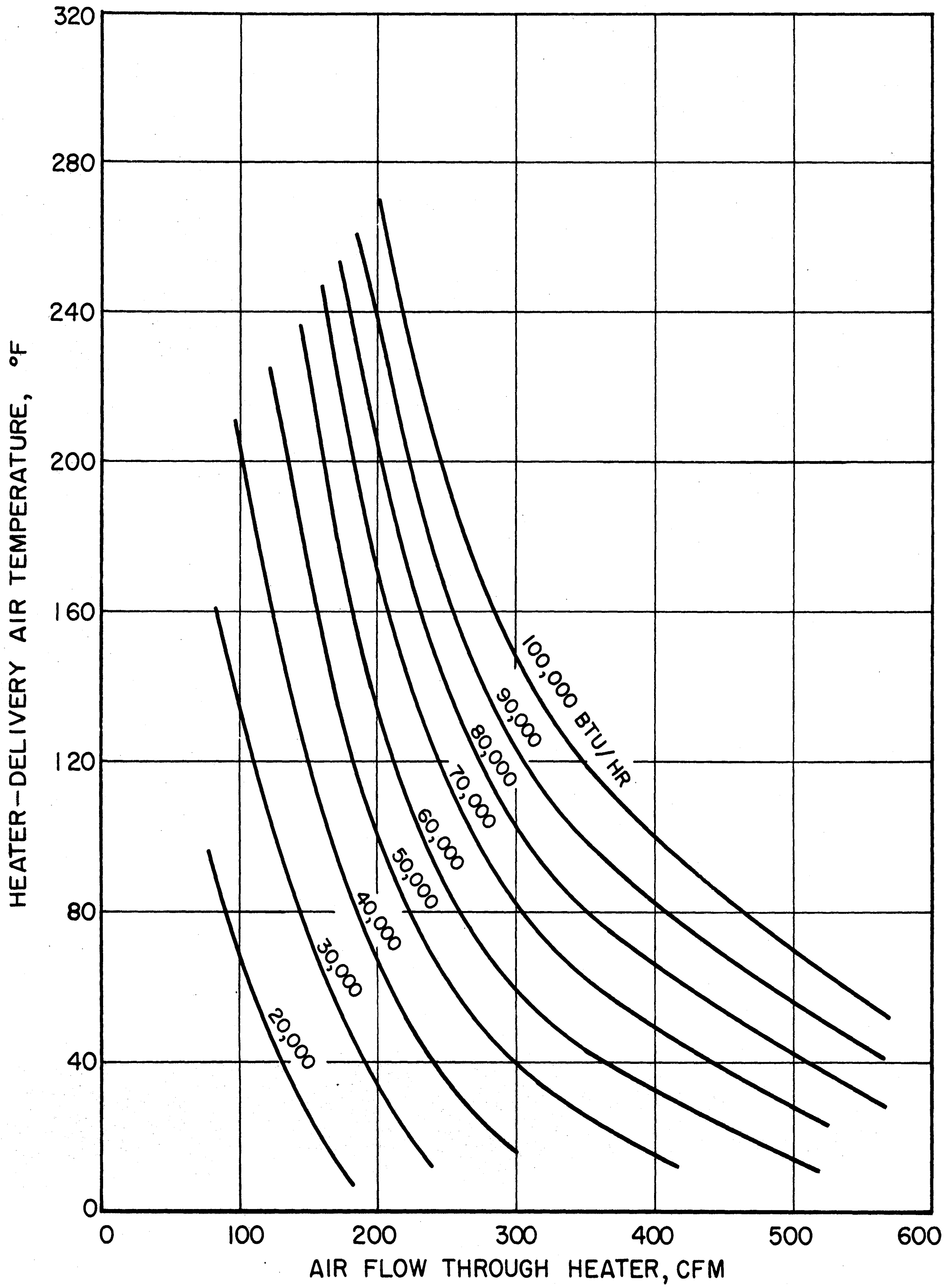


Figure 4. Calculated personnel heater-delivery air temperature vs heater air capacity and heat-release rate; heater ambient air temperature, -70°F.

From Table I, $\eta = 0.66$ for the heater installed in the test combat tank, based on the high heat-release rate and the higher heating value for gasoline. If ($\eta = 0.66$) is assumed, for gasoline, Equation (6) becomes

$$F = 0.125(10^{-4})q_h. \tag{6a}$$

Equation (6a) is plotted in Figure 5. If a gasoline consumption rate up to one gallon per hour is admissible, any heater having up to 80,000 Btu per hour heat release may be considered.

THE CYLINDER ANALOGY

A problem in the estimation of the heat flow for the crew space of a combat tank is the identification of the surface area concerned. The surfaces surrounding the crew space vary in surface finish, roughness, and in thickness which varies from 1 to 7 inches. The armored surfaces may be plane or curved, with orientation and curvature varying irregularly with respect to the horizontal and vertical planes. Interior and exterior surface environments may vary from point to point due to the physical dimensions of the combat tank.

Suppose then that for heat-transfer purposes the complex surface design surrounding the crew space is considered as a simple shape, the thermal characteristics of which can be evaluated. In the absence of the combat tank bustle and forward control space, the crew space approaches the shape of a vertical cylinder. For heat-transfer purposes, therefore, it is assumed that the combat tank crew space is analogous to a vertical cylinder, the height of which is equivalent to the combat tank dimension from the hull floor to the turret deck, and the diameter of which corresponds to the maximum turret dimension seen in plan view, and including the bustle projection if any.

The dimensions of the analogous vertical cylinder are taken as equivalent to the flat surface area required to enclose all the hull and the turret-basket space utilized for crew space. For the M-48 combat tank, the pertinent dimensions are[2]

height, hull floor to turret deck, 90 inches
turret, maximum plan-view dimension, 132 inches

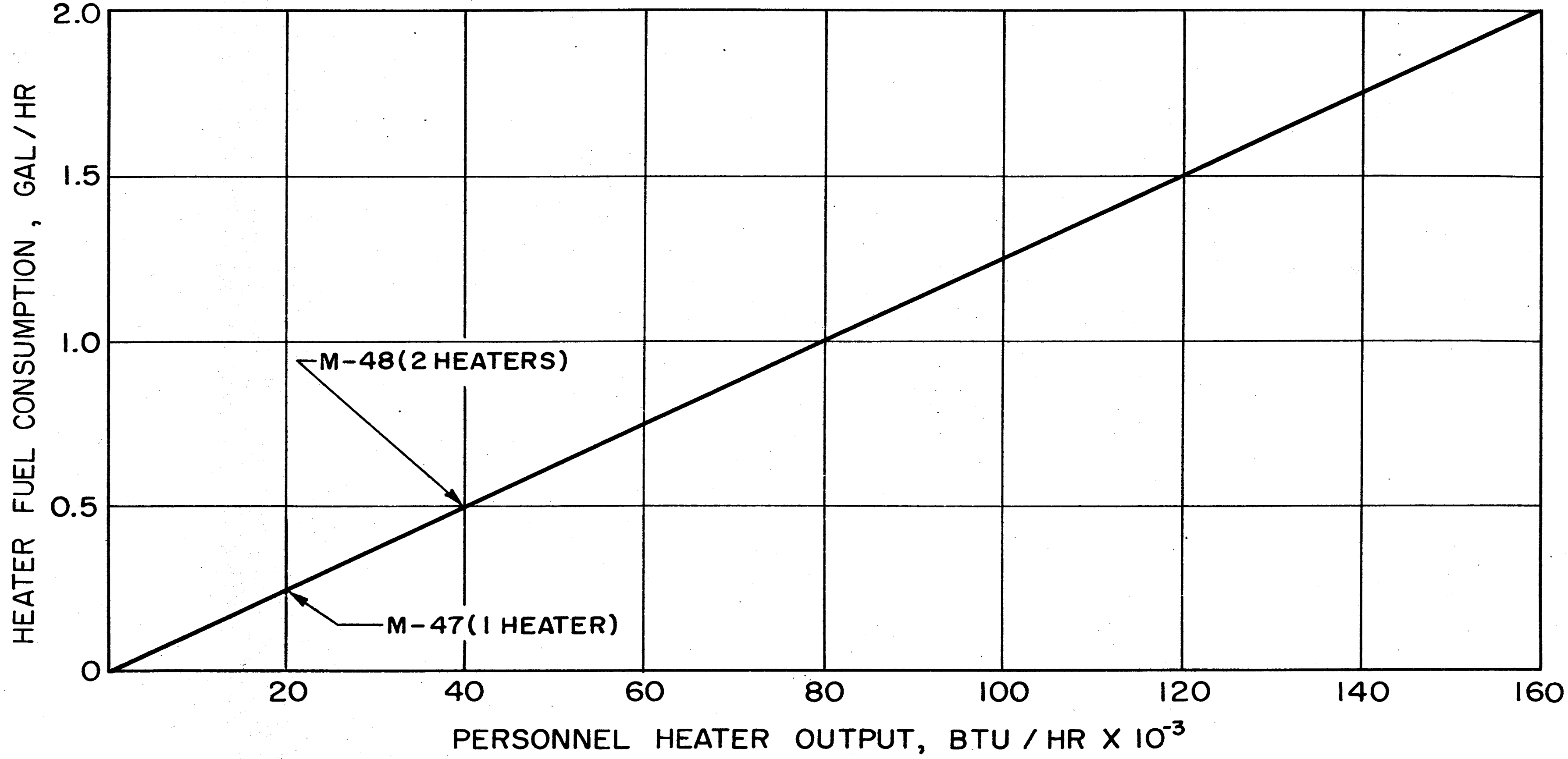


Figure 5. Calculated personnel heater fuel consumption vs heater output, Btu per hour.

The dimensions of the analogous cylinder are

height, 7-1/2 feet
diameter, 11 feet.

Total cylinder surface area is 450 square feet, the vertical walls are 260 square feet in area, and each of the end plates have an area of 95 square feet.

CREW-SPACE SURFACE INSULATION

For given heat addition and other conditions fixed, the application of insulation to the armor surrounding the crew space causes an increased crew-space ambient air temperature and a rise in armor temperature. Due to the exchange of heat by radiation between the body of the crew member and the surrounding armor surfaces, the temperature of these surfaces is of considerable importance to comfort. Interior crew-space surface-temperature evaluation for fixed heat addition of both the noninsulated and the insulated case is required for any study of the usefulness of crew-space surface insulation.

The effect of insulation on crew-space armor temperature is investigated by means of the analogous cylinder idealization, where the cylinder is airtight and constructed of 4-inch-thick armor plate. A heat source is located within the cylinder so that the heat density is applied uniformly over the inner surface. Atmospheric conditions external to the cylinder are fixed and no wind effects are considered. For such assumptions the heat flow is one-dimensional, with a heat source at the inner surface and a constant temperature at the outer surface.

A method of numerical heat flow analysis outlined by Dusinberre[3] is followed. The development of numerical heat-transfer procedures is lengthy and will not be treated in detail as the reader may consult the reference text. The general approach is that if the tmperature at each surface of a homogeneous body remains uniform over the entire surface and changes with time, and if at the initial time temperatures within the body are known, at any subsequent time the body temperatures can be approximated by:

(a) Dividing the body into η imaginary laminae of equal thickness ΔX. Any value may be chosen for η but the greater the value the greater the accuracy of this method.

(b) Temperatures t, t_o, t_1 ..., t_η at the boundaries of these imaginary laminae are determined at equal successive time intervals $\Delta\theta$, where the value of $\Delta\theta$ is

$$\Delta\theta = \frac{c\rho}{k}\left(\frac{\overline{\Delta x}^2}{M}\right). \tag{7}$$

(c) Known initial body temperatures are inserted in a body-temperature—time-increment table for $\Delta\theta = 0$ hour. Successive body temperatures are determined for every time interval $\Delta\theta$ from an average of preceding adjacent body temperatures where the averaging method is governed by the selection of the modulus M.

In the general case, the time dimension attached to the first few numerical cycles is not a valid approximation.

Suppose that consideration is given to the fixed heat addition to the analogous cylinder of 60,000 Btu per hour for the case where insulation is present and for the case where no crew-space surface insulation is used. Any insulation may be selected for the comparison but to accentuate differences an insulation with excellent thermal qualities is chosen. Cork is such an insulation and is chosen for the comparison. The cork insulation is not proposed for actual application to the combat tank, since a number of factors in addition to the thermal characteristics of the insulation require examination.

Thermal characteristics for the steel armor and the cork insulation are listed in Table IV.

TABLE IV

THERMAL CHARACTERISTICS FOR STEEL AND CORK

Nomenclature	Symbol	Steel	Cork	Dimensions
Thermal conductivity	k	25	0.0208	Btu/(hr)(ft^2)(°F/ft)
Specific heat	c	0.13	0.485	Btu/(lb)(°F)
Density	ρ	490	10	lb/ft^3
Thermal diffusivity	α	0.392	0.0043	ft^2/hr
	$c\rho$	63.6	4.85	Btu/(ft^3)(°F)

CASE 1. The analogous cylinder, 4-inch-thick steel walls, and no insulation, a heat source of 60,000 Btu per hour is located within the cylinder.

For this case,

$$q' = \frac{60{,}000}{450} = 133 \text{ Btu/(hr)(ft}^2).$$

Choosing

$$M = 3 \text{ and}$$
$$\Delta X = 4 \text{ inches} = 1/3 \text{ foot,}$$

then

$$\Delta\theta = \left(\frac{c\rho}{k}\right)\frac{\overline{\Delta X}^2}{M} = \frac{\overline{\Delta X}^2}{\alpha M} = \frac{1}{9(0.392)3} = 0.0942 \text{ hour,}$$

$$r \cong \frac{2(450)}{1/3(450)} = 6 \text{ ft}^2/\text{ft}^3, \text{ and}$$

$$Q' = \frac{q'r}{c\rho}\left(\frac{hr}{\Delta\theta}\right) = \frac{133(6)(0.0942)}{63.6} = 1.18°\text{F}/\Delta\theta.$$

Since the outer surface of the cylinder is constant, it is necessary to calculate only the inner surface temperature t_i. With the atmospheric temperature as datum, the numerical computation is given in Table V.

TABLE V

NUMERICAL COMPUTATION OF INTERIOR SURFACE TEMPERATURES OF 4-INCH-THICK STEEL-CYLINDER ANALOGY TO CREW SPACE FOR A HEAT ADDITION OF 60,000 BTU PER HOUR

No. of Time Cycles	Time Increment, θ hr	Interior Surface Temp, t_i °F	No. of Time Cycles	Time Increment, θ hr	Interior Surface Temp, t_i °F
0	0	0	4	0.377	1.75
		1.18			0.58
					1.18

TABLE V (continued)

No. of Time Cycles	Time Increment, θ hr	Interior Surface Temp, t_i °F	No. of Time Cycles	Time Increment, θ hr	Interior Surface Temp, t_i °F
1	0.094	1.18 0.37 1.18	5	0.471	1.76 0.59 1.18
2	0.188	1.55 0.52 1.18	6	0.565	1.77 0.59 1.18
3	0.283	1.70 0.57 1.18	7	0.659	1.77

The numerical computation of Table V demonstrates that with a heat source of 60,000 Btu per hour located in the interior of the 4-inch wall thickness, uninsulated steel cylinder, the temperature of the inside surface of the cylinder rises only 1.77°F above the atmospheric datum in a time period of 0.565 hour or 34 minutes. This computation for the analogy to the uninsulated armor surrounding the crew space indicates that with a 60,000-Btu-per-hour heat addition to the crew space, uninsulated cold armor surrounding the crew space remains cold.

Since the armor is not effective as a heat-flow barrier, the thermal effect of the cork insulation on the crew-space armor surfaces is approximated by consideration of the analogous cylinder composed of the cork insulation only.

CASE 2. The analogous cylinder, 1/4-inch-thick cork walls, and no steel armor; heat source of 60,000 Btu per hour is located within the cylinder.

For this case,

$$q' = 133 \text{ Btu/(hr)(ft}^2\text{)}.$$

Choosing

$$M = 3 \text{ and}$$

$$\Delta X = 1/4 \text{ inch} = 1/48 \text{ foot,}$$

then

$$\Delta\Theta = 0.0337 \text{ hour}$$

$$r = 96 \text{ ft}^2/\text{ft}^3, \text{ and}$$

$$Q' = 88.8°\text{F}/\Delta\Theta.$$

With atmospheric temperature as datum, the numerical computation follows in Table VI.

TABLE VI

NUMERICAL COMPUTATION OF INTERIOR SURFACE TEMPERATURES OF 1/4-INCH-THICK CORK-CYLINDER ANALOGY TO AN INSULATED CREW SPACE FOR A HEAT ADDITION OF 60,000 BTU PER HOUR

No. of Time Cycles	Time Increment, Θ hr	Interior Surface Temp, t_i °F	No. of Time Cycles	Time Increment, Θ hr	Interior Surface Temp, t_i °F
0	0	0 88.8	5	0.169	132.3 44.1 88.8
1	0.034	88.8 29.6 88.8	6	0.202	132.9 44.3 88.8
2	0.067	118.4 39.5 88.8	7	0.236	133.1 44.4 88.8
3	0.101	128.3 42.8 88.8	8	0.270	133.2 44.4 88.8

TABLE VI (continued)

Time Cycles	Increment, θ hr	Surface Temp, t_i °F	Time Cycles	Increment, θ hr	Surface Temp, t_i °F
4	0.135	131.6	9	0.303	133.2
		43.5			
		88.8			

The numerical computation of Table VI indicates that with a heat source of 60,000 Btu per hour located in the interior space of the analogous cork cylinder having 1/4-inch-thick cork walls, the temperature of the inside surface of the cork cylinder rises 133°F above the atmospheric datum in 0.27 hour or 16 minutes. With an atmospheric ambient temperature of -65°F, the interior surface of the cork cylinder rises to a temperature of +68°F. This computation for the analogy to the case of the armor surrounding the crew space, where the armor is insulated with a 1/4-inch thickness of cork, is in contrast to Case 1 (Table V) and removes any doubt concerning the desirability of insulated crew-space armor.

Data from additional numerical computations are given in Table VII for several thicknesses of cork insulation and a heat addition of 60,000 Btu per hour.

TABLE VII

INTERIOR SURFACE TEMPERATURES FOR SEVERAL THICKNESSES OF CORK INSULATION ON THE CREW-SPACE ARMOR FOR A HEAT ADDITION OF 60,000 BTU PER HOUR

Cork Wall Thickness, in.	Interior Wall-Surface Temp Rise, °F	Interior Wall-Surface Temp for a -65°F Atmosphere
1/32	16.7	-48.3
1/16	33.3	-31.7
1/8	66.7	+ 2.7
1/4	133.2	+68.2

The numerical computation method does not consider any exchange of air between the atmosphere and the interior of the analogous cylinder and so it is not possible to obtain all the predicted temperature rises in the combat tank where an air exchange exists between the crew space and the atmosphere. To obtain every advantage from insulation on the armor surrounding the crew space, unplanned air exchange between the combat tank crew space and the atmosphere should be minimized.

Insulation may be attached either to one or both surfaces of the armor surrounding the crew space. For insulation on the crew space surface side (on the interior surface of the vehicle) the temperature of the armor mass approaches the atmospheric temperature. Neglecting radiation effects and where the absorptivity of the insulation surface equals the absorptivity of the armor surface, for insulation on the atmospheric side of the armor, the temperature of the armor approaches the crew-space air temperature. These observations are based on the readily demonstrable fact that only a small temperature differential may occur across a steel wall.

A principal thermal effect due to attachment of insulation on the armor surface facing the atmosphere is that a time delay is introduced in the achievement of the near steady-state condition due to the heat flow to or from the armor mass. A greater time is required to heat the crew space to a given temperature level, and a longer time is required to cool the crew space to a given level, as compared to placement of insulation on the interior or crew-space armor surface. The time parameters of the numerical computations, Tables V and VI, are indicative that the time delay is not great and for a crew-space heat addition of 60,000 Btu per hour this lag probably is less than 1 hour.

From the thermal viewpoint, where radiation effects are neglected, as both the steel armor surface and the insulation surface are subject to similar surface treatment, placement of given insulation on the interior armor surface, the exterior armor surface, or on both surfaces, produces almost equivalent results.

While the observation is beyond the scope of this report, it has been suggested that placement of insulation on the atmospheric surface of the armor, with consequent armor heating for any crew-space heat addition, conceivable could result in reduced armor embrittlement at low atmospheric temperatures. The observation may have significance.

COOLED AIR INTRODUCTION INTO THE CREW SPACE OF THE TEST TANK

To obtain some heat-flow data for the crew space of the test tank, refrigerated air was introduced into the crew space. Air introduction time, atmospheric temperature and pressure, armor surface and crew-space air temperatures, solar radiation, cooling-air flow, air-cycle turbine air temperature, and pressure drop all were measured using instrumentation indicated in Table XIV.

The refrigerator unit was mounted in a steel box which was set on top of the test-vehicle turret bustle as shown in the test mock-up in Photograph 1. The air-cycle refrigeration unit mounted in the box, with the box cover plate removed, appears in Photograph 2. The test procedure began with "buttoning" the test vehicle and measurement of initial conditions. Cooled air then was introduced into the crew space and at time intervals the test parameters were measured and recorded.

Because of the availability of both an air-cycle expansion turbine and a source of compressed air, the air-cycle cooling unit was used to supply cooled air to the crew space of the combat tank. The air inlet of the air-cycle turbine was connected to the laboratory high-pressure air supply so that inlet air pressure to the turbine could be controlled by means of a pressure reducing valve. Discharge air from the air-cycle turbine was ducted into the combat tank crew space through an opening in a steel plate covering the opening in the tank bustle remaining after the turret ventilation blower was removed. The cooled-air delivery hose was of fiber, 4 inches in diameter, and was about 1 foot in length so as to limit thermal losses.

The aircraft-type air-cycle refrigeration unit was manufactured by AiResearch Manufacturing Company of Los Angeles, and is identified as follows:

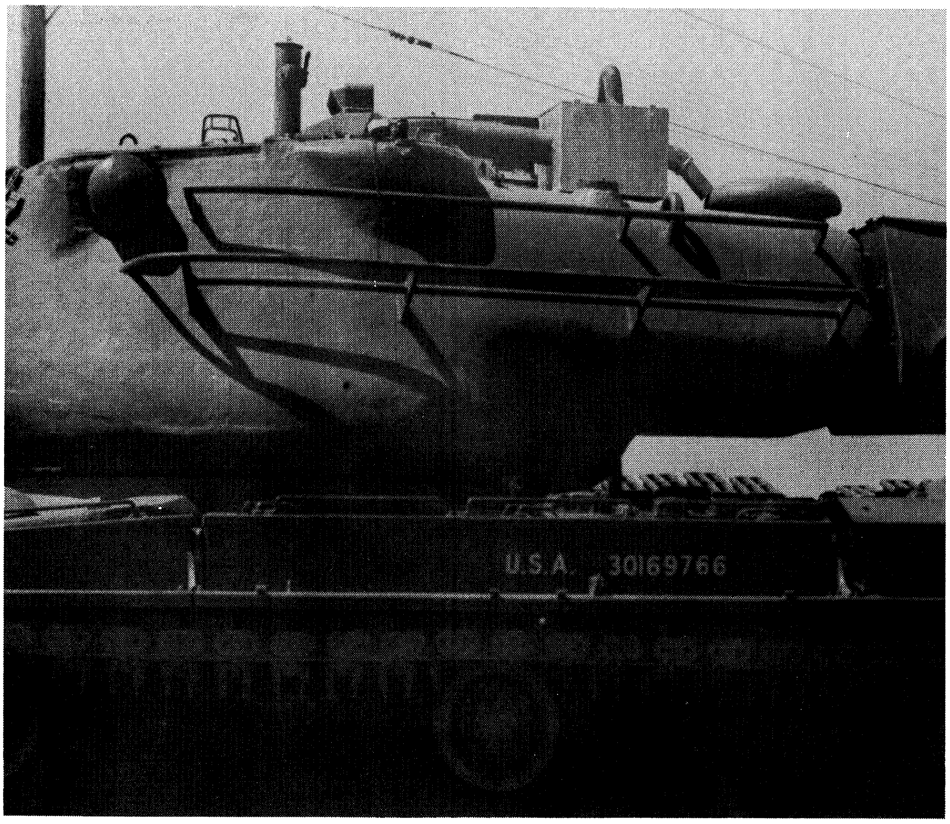
Drawing No.	-	52021-150
Type	-	Turbine
Assembly No.	-	52022-150
Order Serial	-	9-104
Patent No.	-	2,398,655

The above designation was submitted to the manufacturer to obtain performance data, but since the unit is an old model, no data were obtained.

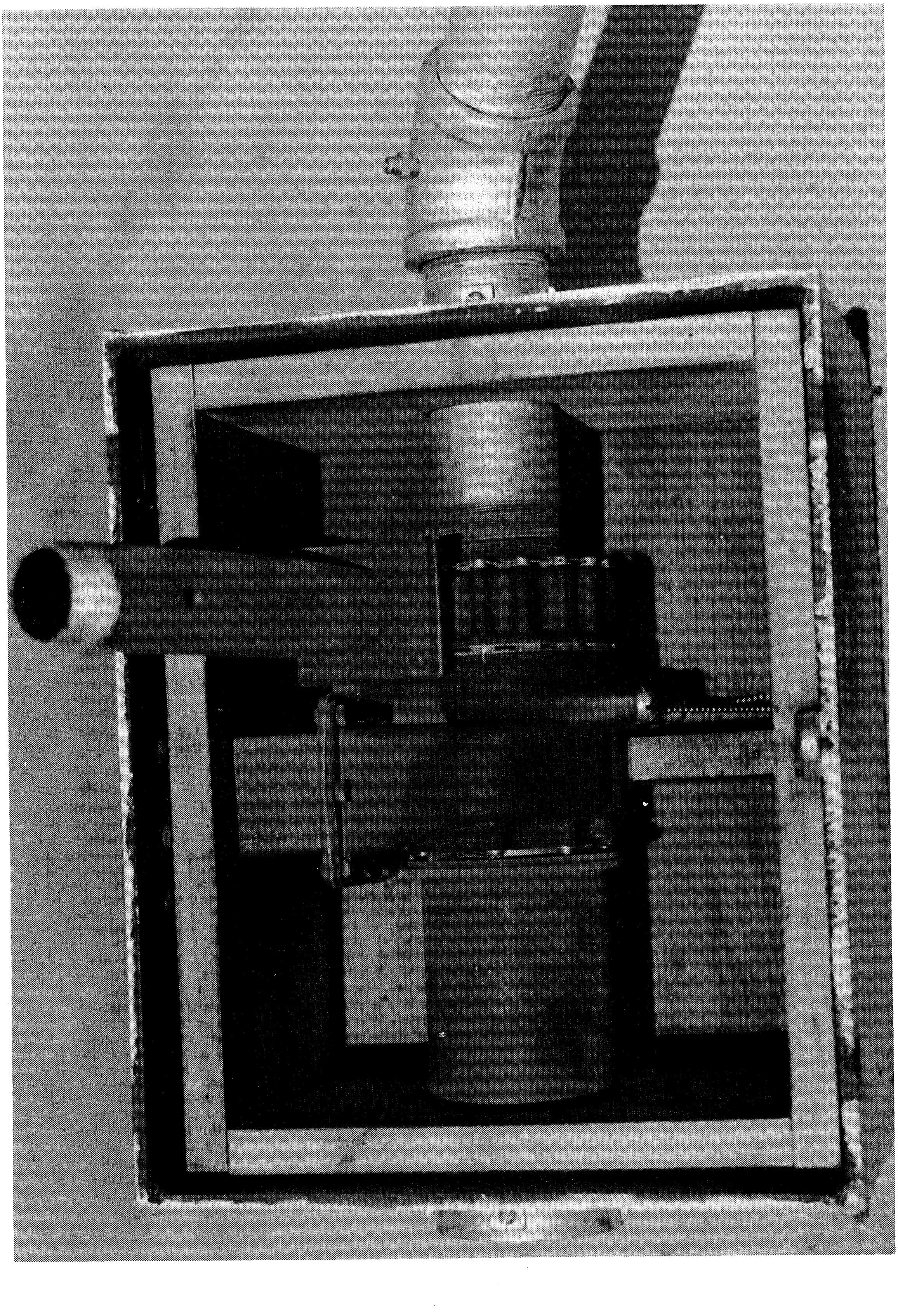
Measured turbine and fan diameters are 2.125 and 2.75 inches, respectively. The overall length of the air-cycle unit approximates 8 inches, and the width about 7 inches. Total unit weight is 6 pounds. No heat exchanger was obtained with this unit.

Air-cycle refrigeration uses a Brayton cycle except that the directions of the processes are reversed. The necessary processes are:

(a) an air compression,
(b) reduction of the heat of compression, usually by means of a heat exchanger, and



Photograph 1. Mock-up of air-cycle refrigeration unit mounted on test vehicle.



Photograph 2. Air-cycle refrigeration unit mounted in steel box.

(c) an expansion of the air by means of a turbine or other expansion device.

All three processes are requirements. Figure 6 is a schematic drawing of a conventional air-cycle cooling system where listed values are typical. Figure 7 illustrates turbine-fan operation and typical data. Figure 8 is a schematic drawing of the system used to supply cooled air to the crew space of the test vehicle, together with some typical performance numbers. Note that the heat exchanger requirement is supplied by storage bottle surface. Figure 9 shows the thermodynamic process for the air-cycle system which is shown in Figure 8 and used in the cooling tests.

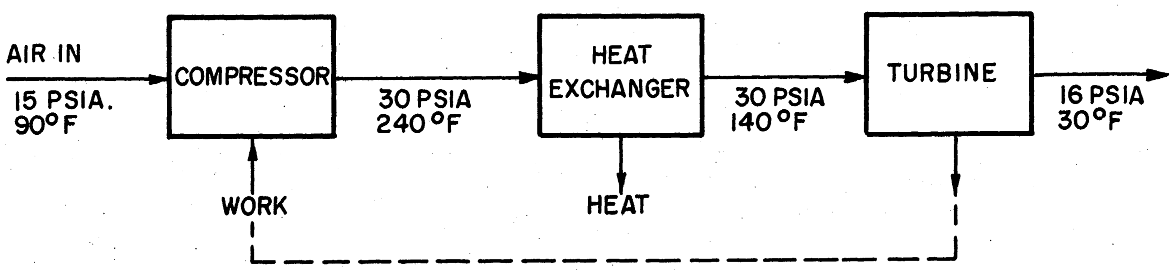


Figure 6. Conventional air-cycle expansion system.

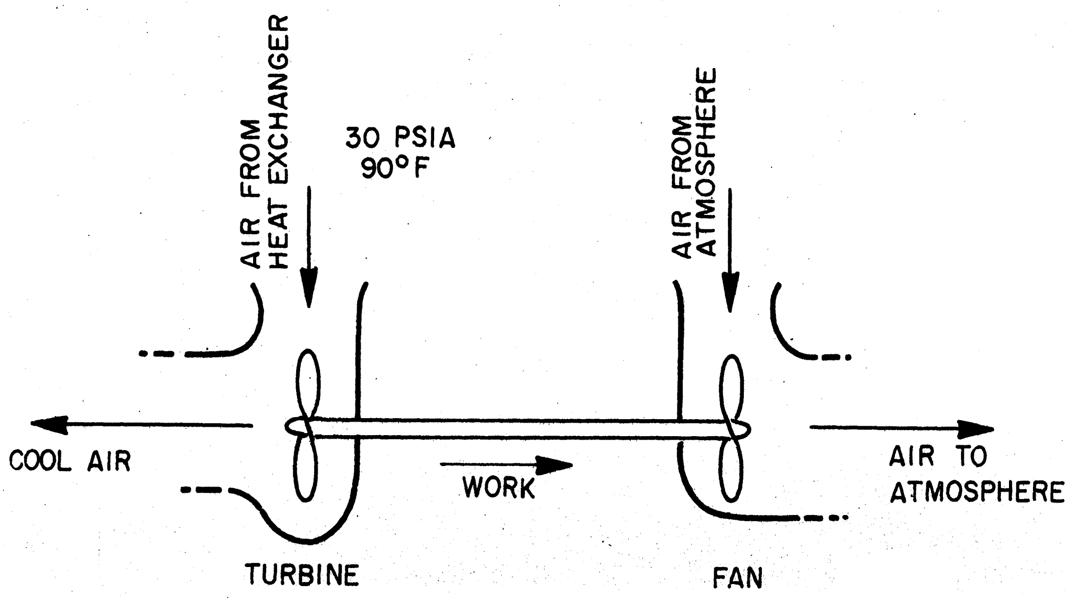


Figure 7. Turbine-fan operation for air-cycle cooling system.

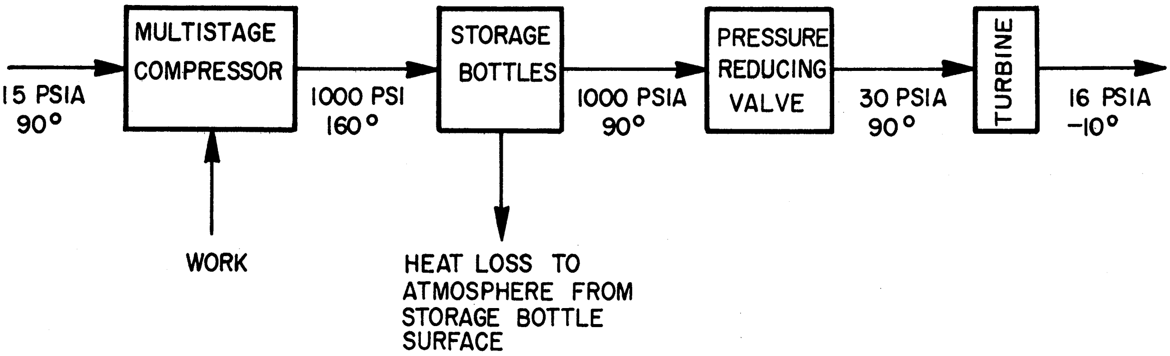


Figure 8. Air-cycle cooling system used to cool combat tank crew space.

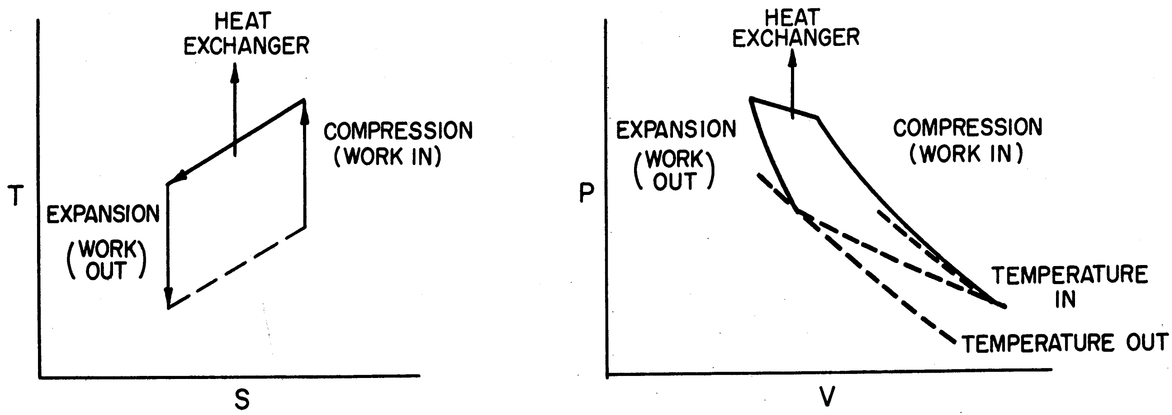


Figure 9. Thermodynamic process for air-cooling system used to cool combat tank crew space.

The noise intensity of the air-cycle unit used in the cooling tests was high and probably would not be tolerated in the field. No noise-intensity measurements were made since lower speed and less noisy air-cycle units are available.

In the cooling tests, the refrigeration effect was varied from about 1/2 to 1-1/2 tons with free air flows ranging from 90 to 145 cfm. Atmospheric temperatures varied from 68° to 84°F and solar inputs varied as listed in Table IX. Mixing of the cooled air made it practical to reduce test data to average values which are tabulated in Tables VIII, IX, and X. The largest refrigeration effect produced by the cooling system (1-1/2 tons) did not comfortably cool the uninsulated crew space of the test vehicle. The largest crew-space average ambient temperature drop was 18°F and there was a small

TABLE VIII

AVERAGE TEST CONDITIONS, AIR-CYCLE REFRIGERATION UNIT DISCHARGING COOLED AIR INTO CREW SPACE OF TEST COMBAT TANK

Test Number	1	2	3	4	5	6	7	8	9
Date	7-23-54	8-6-54	8-9-54	8-11-54	8-12-54	8-13-54	8-18-54	8-20-54	8-26-54
Time, Start	15:40	13:08	12:50	13:55	14:00	13:18	14:05	14:40	15:37
Stop	15:52	16:30	16:00	16:30	16:30	15:10	14:48	15:30	16:10
Duration, hr and min.	0-12	3-03	3-10	2-35	2-30	1-52	0-43	0-50	0-13
Refrigeration, tons	0.54	0.685	0.956	1.16	1.34	1.43	1.47	1.57	1.5
Refrigeration, Btu/hr	6480	8220	11,472	13,920	16,080	17,160	17,600	18,800	17,940
Air flow, lb/min	6.69	6.91	8.19	9.0	10.01	10.05	10.67	10.81	10.32
Free air flow, ft^3/min	90	92	109	120	133	134	142	144	138
Exterior armor temp., °F, start	86	93	88	80	87	89	72	90	90
stop	85	89	85	78	83	91	73	90	84
Interior armor temp., °F, start	86	92	86	79	86	87	71	91	89
stop	83	88	82	76	81	89	71	89	78
Crew-space air temp., °F, start	79	83	78	74	76	78	68	86	79
stop	74	71	64	63	58	66	53	72	65
Crew-space air temp. drop, °F	5	12	14	11	18	12	15	14	14
Atmospheric temp., °F, start	--	84	81	68	77	77	68	86	74
stop	--	80	79	72	77	79	71	83	73

TABLE IX

AVERAGE TEST CONDITIONS, AIR-CYCLE REFRIGERATION UNIT DISCHARGING COOLED AIR INTO CREW SPACE OF TEST COMBAT VEHICLE

	Test Number								
	1	2	3	4	5	6	7	8	9
Solar radiation, Btu/(hr)(ft^2)									
Meter aimed at sun		87	73	61	103	126	28	82	12
Meter vertical		66	61	50	89	110	26	74	11
Meter horizontal		63	44	39	66	73	13	63	9
Air-cycle refrigeration unit									
Air temp., °F, entry	77	78	76	63	69	79	62	77	65
exit	10	-4	-21	-44	-42	-39	-52	-43	-47
Turbine temp. drop, °F	67	82	97	107	111	118	114	120	112
Turbine press ratio	2.03	2.4	2.8	3.27	3.7	3.76	3.65		3.36

(1°-5°F) change in armor temperature, probably due mainly to change in atmospheric conditions. The minimum and maximum cooling-air velocities in the crew space are reported in Table X and were believed to be too high for comfort. It is noted that the test air velocities usually exceed the 20-fpm cooling-air velocity recommended in Reference 1, page 9 of this report.

TABLE X

MINIMUM AND MAXIMUM AIR VELOCITIES (FPM) MEASURED DURING COMBAT TANK CREW-SPACE COOLING TESTS*

		Test Number			
		2	5	6	8
Commander region,	face	25-50	20-120	35-85	
	waist	20-45	16-150	120-400	50-55
	knee	25-65	18-65	10-130	20-25
Gunner region,	face	15-30	17-200	45-100	10-13
	waist	20-50	15-80	45-100	20-28
	knee	15-30	16-80	25-60	15-22
Loader region,	face	30-55	38-200	30-100	20-45
	waist	60-95	26-130	80-250	14-20
	knee	85-130	30-100	70-120	20-35
Driver region,	face	10-20	15-40	10-60	25-30
	waist	10-17	15-25	18-25	14-18
	knee	10-19	15-35	16-20	18-25
Machine gunner region,	face	10-30	18-35	22-50	18-28
	waist	10-15	15-24	17-27	16-19
	knee	10-21	15-20	15-22	15-21

*Air velocity measurements made only for the cooling tests reported in table.

INSULATION AND REFRIGERATION REQUIREMENT

Colburn and Hougen[4] demonstrated that for either natural convection or for forced convection where the velocities are low, the film coefficient of convection for gases heated on vertical plates is defined by

$$h_c = 0.128 \left(\frac{k^3 \rho^2 c \beta g [\Delta t]}{\mu}\right)^{1/3} . \qquad (8)$$

Griffiths and Davis[5] found that the film coefficient for horizontal plates facing upward is about 27 percent higher than for vertical plates; and for horizontal plates facing downward, about 33 percent lower.

The combined coefficient for convection and radiation from flat or cylindrical surfaces in a room is given by

$$h_{cc} = h_c + \epsilon h_r . \qquad (9)$$

The expression of Colburn and Hougen and the findings of Griffiths and Davis may be used to calculate the film coefficient of convection for heated vertical and horizontal flat surfaces located in a space having a given air temperature. If the surfaces are insulated with insulation of known emissivity, then the combined coefficient may be found using

$$h_{cc} = h_c + \epsilon h_r .$$

This was done for an assumed insulation emissivity of 0.9 and for heated surfaces located in a room or space where the air temperature is 70°F.[6] Data from these calculations are given in Figure 10.

The combined coefficient is a function of the temperature of the free surface of the insulation, but for practical purposes useful values of the combined coefficient may be obtained by assuming a reasonable surface temperature. More accurate predictions for the surface temperature then are obtained by trial-and-error solution. Average temperature of the insulation is assumed as the arithmetic average of the insulated surface and the room air temperatures.

For vertical cylindrical surfaces of large radius compared to surface thickness, the values of the combined coefficient almost agree with values calculated for the vertical flat plate. For flat surfaces the rate of the heat loss through insulation is

$$q = \frac{k A(t_{is} - t_a)}{x + \left(\frac{k}{h_{cc}}\right)} . \qquad (10)$$

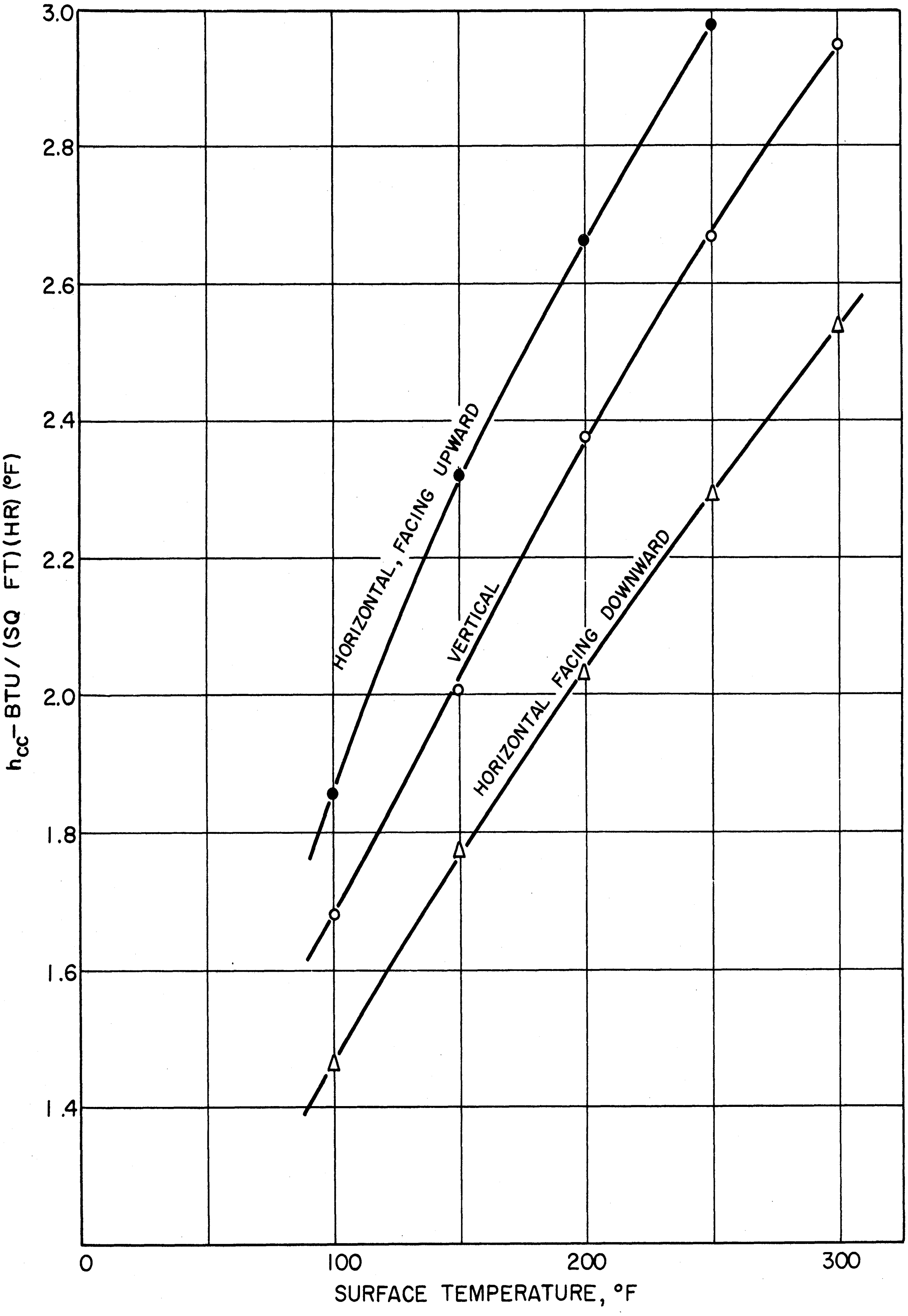


Figure 10. Combined coefficient h_{cc} for convection and radiation from flat insulated surfaces in a room at 70°F; assumed ϵ = 0.90.

The cylinder analogy will be used to calculate the refrigeration requirement of a combat tank and the effect of insulation thickness on the refrigeration requirement for the initial condition of elevated armor temperature accompanied by high atmospheric temperature. Limited observation and the literature concerned with high temperature conditions suggest that armor temperatures of the order of 150°F are not unreasonable. The calculations are based on this armor temperature of 150°F which is assumed to exist due to solar radiation when the atmospheric temperature is 120°F. The desired crew-space temperature is taken to be 80°F. When insulation is considered, the calculations are for insulation on the crew side of the armor. Insulation of fair quality is considered for which a typical value of thermal conductivity is used as follows:

$$t_{average} = 100°F, \; k = 0.039 \text{ Btu/(hr)(ft}^2\text{)(°F/ft)}$$

$$= 200°F, \; k = 0.041.$$

Based on the outlined considerations, the refrigeration requirement, due to heat loss through the armor surrounding the crew space of a combat tank, has been calculated as a function of insulation thickness. The data of the calculations are listed in Table XI and shown in Figures 11 and 12.

TABLE XI

COMBAT TANK CREW-SPACE REFRIGERATION CAPACITY REQUIRED DUE TO LOSSES THROUGH THE ARMOR FOR SEVERAL THICKNESSES OF INSULATION

$k \cong 0.04$ Btu/(hr)(ft^2)(°F/ft) for
atmospheric temperature = 120°F
armor surface temperature = 150°F
crew-space air temperature = 80°F

REFRIGERATION REQUIREMENT DUE TO LOSSES THROUGH ALL SURFACES SURROUNDING CREW SPACE

Insulation Thickness, in.	Heat Flow q, Btu/hr	Refrigeration Capacity, Ton	% of Zero Insulation Load
0	64,900	5.4	100
1/32	55,300	4.6	85
1/16	48,800	4.1	75
1/8	40,300	3.4	62
1/4	29,000	2.4	45
1/2	19,200	1.6	30

TABLE XI (continued)

DISTRIBUTION OF HEAT LOSSES THROUGH THE ARMOR SURROUNDING THE CREW SPACE

A. Losses for Deck Surface and Turret Deck Region

Insulation Thickness, in.	Heat Flow q' Btu/(hr)(ft^2)	Heat Flow q, Btu/hr	Refrigeration Capacity Ton	% of Total Armor Loss
0	124	11,800	0.98	17.6
1/32	108	10,300	0.86	18.6
1/16	97	9,240	0.77	18.8
1/8	80	7,650	0.64	19.0
1/4	61	5,760	0.48	19.9
1/2	41	3,880	0.32	20.2

B. Losses for Hull and Turret Side-Wall Armor

Insulation Thickness, in.	Heat Flow q' Btu/(hr)(ft^2)	Heat Flow q, Btu/hr	Refrigeration Capacity Ton	% of Total Armor Loss
0	145	37,700	3.14	58.3
1/32	123	32,000	2.68	58.1
1/16	109	28,300	2.36	58.1
1/8	91	23,600	1.97	58.6
1/4	64	16,700	1.39	57.6
1/2	43	11,100	0.92	57.7

C. Losses for Hull Armor Facing the Ground

Insulation Thickness, in.	Heat Flow q' Btu/(hr)(ft^2)	Heat Flow q, Btu/hr	Refrigeration Capacity Ton	% of Total Armor Loss
0	162	15,400	1.28	23.9
1/32	137	13,000	1.08	23.4
1/16	119	11,300	0.94	23.1
1/8	95	9,050	0.75	22.4
1/4	68	6,500	0.54	22.5
1/2	45	4,240	0.35	22.1

The refrigeration capacity required due to heat loss through the armor is not the entire refrigeration requirement. An additional load is introduced due to leakage. The maximum air leakage occurs when the turret ventilation blower is operating at maximum capacity, which for the test tank is 1500 cfm. Total refrigeration requirement may include capacity accounting

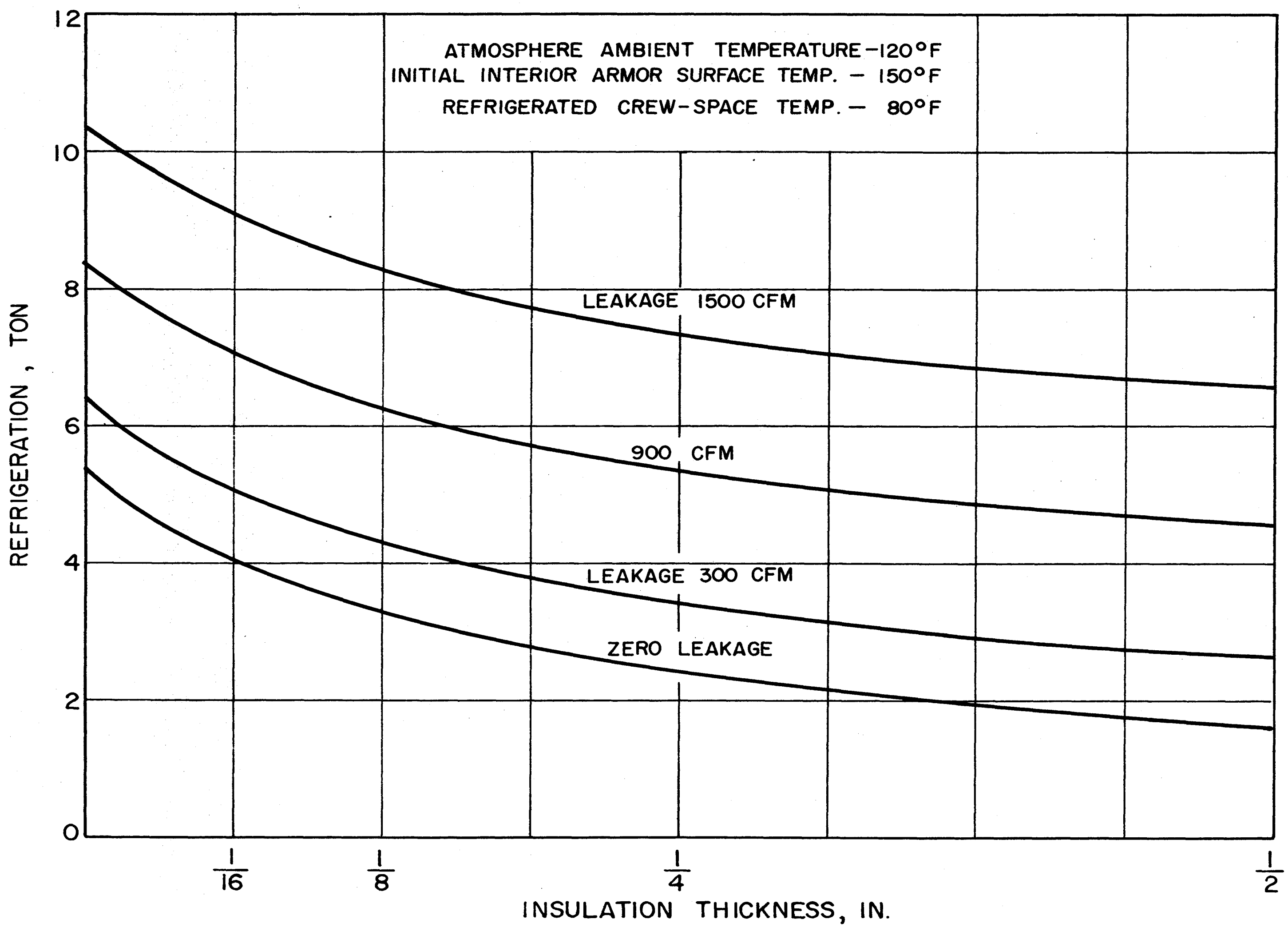


Figure 11. Total refrigeration requirement for a combat tank as a function of crew-space air leakage and insulation thickness.

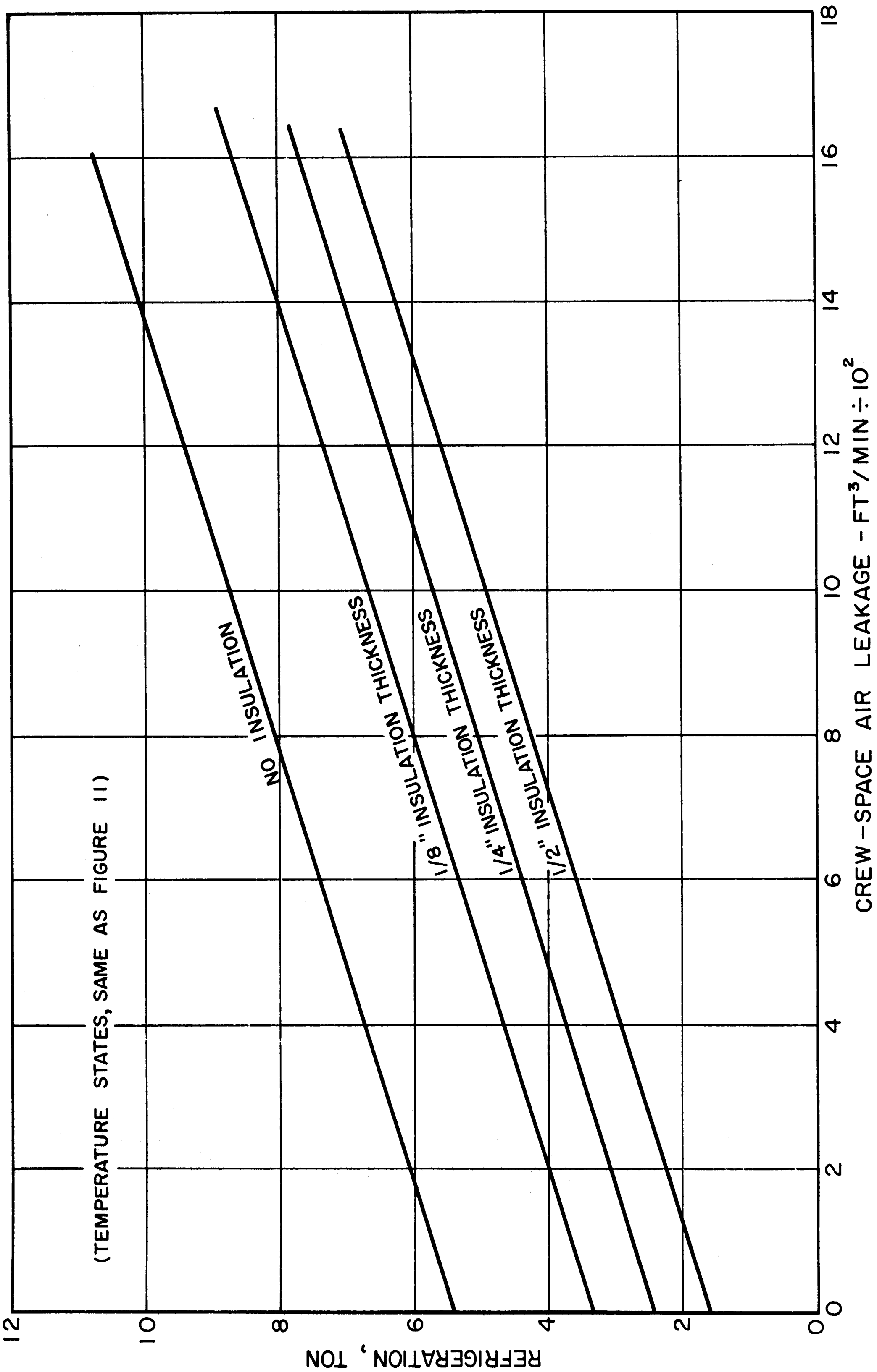


Figure 12. Total refrigeration requirement for a combat tank as a function of insulation thickness and crew-space air leakage.

for this air leakage. However, for a crew space having a cooled-air introduction, the turret ventilation blower probably operates only infrequently, perhaps only to expel gun fumes.

Refrigeration capacities equivalent to convective air-leakage rates of 1500, 900, and 300 cfm are added to the zero-leakage refrigeration requirement; these overall refrigeration requirements also are shown in Figures 11 and 12. For the 40°F air temperature difference, (120-80), the refrigeration capacity to account for air-leakage rates of 1500, 900, and 300 cfm is about 5, 3, and 1 tons, respectively.

AIR CONDITIONING OF BUSES, AIRCRAFT,[7] AND PASSENGER CARS

An outline of some current air-conditioning practices in the transportation field follows, as such practices may be compared to predicted combat tank cooling requirements.

INTERCITY BUSES

The capacity of the average refrigeration system in an intercity bus is about 4 tons. This capacity usually is sufficient to maintain an inside condition of 78°F dry-bulb temperature, and 67°F wet-bulb temperature, when the atmospheric condition is 95°F dry-bulb and 78°F wet-bulb temperature.

Bus cooling requirements are minimized by reduction of air infiltration and the use of body insulation. Heat-absorbing glass is used to reduce the solar radiation load. Outer surfaces of window shades and the exterior surfaces of the vehicle are treated to obtain high reflectivity.

A typical 35-foot-long intercity bus with a surface area approximately 1250 square feet weighs about 20,000 pounds, of which 1200 pounds represent air-conditioning equipment. Mechanical compression, usually with Freon-12 as the refrigerant, is the commonly used system. The refrigeration compressor is driven either by the main propulsion or an auxiliary engine. The auxiliary engine, for a 4-ton refrigeration capacity, develops about 20 brake horsepower at 2000 rpm.

About 10 cfm of fresh air per passenger is introduced into the bus. The air introduction maintains a small pressure within the vehicle so that only air leakage out of the vehicle body occurs. No exhaust-air openings are provided. Cooled-air distribution throughout the passenger space is accomplished by the use of perforated ceiling ducts; there are no return-air ducts. The air-conditioning controls are simple and as few as possible, to reduce nonoperation due to control failure.

AIRCRAFT

In aircraft, refrigeration capacities vary from about 1 ton for the fighter airplane with a cockpit volume of 40 to 60 cubic feet up to about 10 tons for commercial pressurized aircraft having some 6000 cubic feet of cabin volume.

Satisfactory airliner cabin air conditions are 73° to 75°F dry-bulb temperature, a relative humidity of 50 percent, an air movement of 15 to 25 fpm about the passenger's body, and no heat loss or gain from the body to the surrounding surfaces.

PASSENGER CARS

Air conditioning in passenger cars is a recent development. Main engine powered mechanical compression with Freon-12 as the refrigerant is the common system. Total weight of the system is about 200 pounds and the installed cost approximates $500.00. Typical data are:[8]

car speed, mph	20	40	60
compressor, rpm	1090	2185	3280
Refrigeration effect, Btu/hr	13,750	25,250	33,000
Refrigeration effect, ton	1.1	2.1	2.7

AREA OF AIR-LEAKAGE OPENINGS IN THE ARMOR SURROUNDING THE CREW SPACE

Thermal losses occur for the crew space of a combat tank due to convective air flow through openings in the armor surrounding the crew space. The magnitude of such losses may be estimated only after the area of leakage is known. If the area of air leakage is large, in the case of crew-space heating, thermal losses could be reduced by minimizing the area for air leakage. A combat tank having a crew space which could be made airtight at will could have important military significance. For these reasons, it is important that the area of the air-leakage openings in the armor surrounding the crew space of the combat tank be known.

Direct measurement of the areas of the several small openings is difficult and probably not accurate. An indirect approach is made where a known air flow is introduced into the buttoned-down crew space of the test vehicle through an opening in a steel plate covering the well remaining after removal of the turret air ventilation blower. Subsequent to the establishment of a steady-state condition for the known air introduction, the air pressure difference was measured between the crew space and the atmosphere and pressure and temperature parameters so that air densities may be calculated. Actual parameters measured for each test point were:

atmospheric temperature and pressure,
air pressure difference between crew space and atmosphere,
average air temperature in crew space, and
air introduction parameters including
(a) ASME orifice air pressure,
(b) ASME orifice differential air pressure, and
(c) air temperature.

The test apparatus and instrumentation are listed in Table XIV, and test conditions for each test are given in Table XII.

The calculations are based on consideration of the Bernoulli equation

$$P_1 + \frac{\rho_1 V_1^2}{2g} = P_2 + \frac{\rho_2 V_2^2}{2g}, \tag{11}$$

where subscript ($_1$) denotes air conditions within the crew space, and subscript ($_2$) refers to conditions immediately downstream of the air-leakage opening.

If the assumption is made that the air velocity within the crew space approaches zero, then Equation (11) reduces to

$$V_2 = \sqrt{\frac{2g}{\rho_2}(P_1 - P_2)} = \sqrt{\frac{2g}{\rho_2}}\sqrt{\Delta P}\,. \tag{11a}$$

But

$$m' = 60\,\rho_2\,A_2\,V_2 \tag{12}$$

and

$$m' = 60\,A_2\sqrt{2g}\sqrt{\rho_2\,\Delta P};$$

$$m' = C_1\sqrt{\rho_2\,\Delta P}\,. \tag{13}$$

TABLE XII

AIR-LEAKAGE TEST CONDITIONS

Test Conditions Common to All Tests	
Vehicle location	Test vehicle standing in the open, east of Building Number 8-1/2, Willow Run Airport, Ypsilanti, Michigan.
Vehicle orientation	Driver cockpit and main gun facing north.
Vehicle condition	Turret ventilation blower removed and well-capped with a steel plate. All hatches buttoned-down, canvas cover of gun muzzle and gun breech shroud in position.
Air supply	Air from high-pressure storage flows through a pressure reducing and regulating valve and then through as ASME-type calibrated square-edge flat plate measuring orifice. Air then flows either to the air-cycle refrigeration unit and into the crew space through an entry in the steel cap on the well of the air ventilation blower or directly from the orifice to the entry in this cap.

Test Conditions Particular to Tests

Tests 1 and 2	Test 3	Test 4	Test 5
Air supplied to air-cycle turbine sitting on turret bustle and discharged from turbine into crew space, main and auxiliary engines not operating; air filter passages between crew space and engine space blocked.	Similar to tests 1 and 2 except that air-filter passages between crew space and engine space open.	Similar to Test 3 except that main engine operating at idle speed measured by vehicle tachometer.	Air directly into crew space; no air-cycle refrigeration unit; no engines operating; air-filter passages between crew space and engine space closed.

TABLE XIII

EXPERIMENTAL DATA, CREW-SPACE LEAKAGE AREA TESTS

Time	Corrected Atmospheric Pressure P_A, lb/ft²	Crew Space to Atmosphere ΔP, lb/ft²	Crew-Space Air Pressure P_i, lb/ft²	Orifice Air Temperature, °R	Orifice Air Pressure, psia	Orifice Pressure Drop $\Delta P'$, in. $CH_3(OH)$	Orifice Air Flow m', lb/min,	Orifice Air Flow W cfm	Crew-Space Average Air Temperature, °R	Crew-Space Air Density ρ, lb/ft³	$\left(\frac{\Delta P}{\rho}\right)^{1/2}$	C	$\bar{A}_2 \left(\frac{C}{294}\right)$ in.²
						Test No. 1, October 8, 1954							
15.50 - Start test													
15.55	2083	0.140	2083	509	30.47	0.35	6.67	82.3	510	0.077	1.35	66.1	32.4
16.00	2083	0.312	2083	505	47.67	0.56	8.95	117.2	507	0.077	2.01	57.8	28.3
16.05	2084	0.51	2083	501	59.27	0.68	11.08	145	504	0.077	2.56	54.3	26.6
16.10 - Stop test													
						Test No. 2, October 11, 1954							
13.20 - Start test													
13.26	2048	0.078	2048	528	24.02	0.25	4.14	54.1	528	0.073	1.03	54.5	26.7
13.31	2048	0.187	2048	524	34.42	0.43	6.59	86.1	526	0.073	1.6	57.2	28.0
13.36	2048	0.328	2048	521	46.62	0.59	8.96	117.3	523	0.074	2.21	57.4	28.1
13.37 - Stop test													
						Test No. 3, October 11, 1954							
13.50 - Start test													
13.55	2112	0.094	2112	525	25.12	0.26	4.41	57.6	527	0.075	1.12	52.3	25.6
14.00	2112	0.166	2112	523	34.12	0.4	6.4	83.6	526	0.075	1.49	57.1	28.0
14.05	2111	0.296	2111	521	43.91	0.49	7.93	103.8	525	0.076	1.98	52.8	25.9
14.06 - Stop test													
						Test No. 4, October 11, 1954							
14.18 - Start test													
14.25	2111	0.281	2111	524	24.31	0.26	4.3	56.6	530	0.075	1.94	29.6	14.5
14.30	2111	0.354	2111	522	33.31	0.4	6.25	82.2	529	0.075	2.17	38.1	18.7
14.35	2111	0.477	2111	520	44.66	0.55	8.48	114	529	0.075	2.52	43.8	21.4
14.36 - Stop test													
				(Main engine idle speed set at 1200 rpm at 14.18; at 14.37 main engine speed was 1500 rpm.)									
						Test No. 5, October 13, 1954							
15.53 - Start test													
15.58	2069	1.45	2068	518	25.6	3.7	16.6	217	535	0.073	4.45	52.5	25.7
16.03	2069	3.09	2066	508	35.6	5.4	24	313	532	0.073	6.5	50.3	24.6
16.08	2069	5.77	2063	506	46.8	9.0	38.6	505	529	0.073	8.87	59.5	29.1
16.13	2069	8.88	2060	505	64.3	11.5	45	588	528	0.073	11.0	56.0	27.4
16.18	2069	11.53	2057	503	70.3	13.8	54	710	526	0.073	12.6	54.8	26.8
16.19 - Stop test.		(Air supply check showed an additional point could be run.)											
16.21 - Start test													
16.26	2069	15.15	2054	505	78.8	15.7	61	796	526	0.073	14.4	58.1	28.5
16.27 - Stop test													

TABLE XIV

INSTRUMENTS AND EQUIPMENT USED IN EXPERIMENTS

1. M-47 combat tank, Designation USA 30169766(DE-29).

2. Brown electronik temperature recorder; 16 points; Model No. 153X(67)-P16-X-(106)A4K; Serial No. 746819, range, -150° to 200°F in 2°F subdivisions.

3. Brown electronik temperature recorder; 16 points; Model No. 153X62P16-X-13; Serial No. 568497; range, 0° to 350°F in 2°F subdivisions.

4. Mercury-filled thermometers; range, 0° to 220°F in 2°F subdivisions, Cenco Cat. No. 19297.

5. Toluol-filled thermometers; range, -100° to 50°C in 1°C subdivisions; Cenco Cat. No. 19370.

6. 24-gauge, polyvinyl, insulated, double-conductor, iron-constantan wire; M-H Cat. No. 9B3N4.

7. Laboratory-type mercury-filled barometer for measurement of atmospheric air pressure; Cenco Cat. No. 76890.

8. Meriam inclined manometer, calibrated in 0.001 inch of water; Serial No. 47B253; Model A-763. This instrument used to determine air pressure difference between crew space and atmosphere.

9. ASME-type flat plate orifice, calibrated by A. Weir, Jr.[9] The orifice was used to measure the air flow into the crew space of the tank.

10. Vacuum-thermocouple type radiation meter, calibrated in 0 to 2 gram-calories per square centimeter per minute; Cenco Cat. No. 81085. Instrument used to measure solar radiation.

11. Hot-wire anemometer; Type 8500; Serial No. TA-387; manufactured by Illinois Testing Laboratories; dual range: 5 to 300 fpm and 70 to 2000 fpm. Instrument used to measure air movement.

12. Air-cycle refrigeration unit, Assembly No. 52022-150; Serial No. 9-104; manufactured by AiResearch Manufacturing Company. Equipment used to obtain cooled air for introduction into crew space.

TABLE XIV (continued)

13. Miscellaneous instruments and equipment include a high-pressure air-storage facility along with nitrogen-controlled air-pressure regulating valves, manometers, pressure and temperature gauges.

14. A Brown recorder and gas-welded iron-constantan thermocouples suspended in air were used to measure crew-space air temperatures. Thermometers were used as a check.

15. Armor surface temperatures were measured with equipment similar to (14) above except that the thermocouples were fastened to the surface with a mixture of powdered copped metal DuPont Duco cement. No attempt was made to remove surface finish or oxidation.

Equation (13) may be rewritten:

$$W = C_1\sqrt{\frac{\Delta P}{\rho_2}} \; . \tag{14}$$

With reference to Equation (13), if m' and $\sqrt{\rho_2 \; \Delta P}$ are determined by experiment, the constant C, is known and the air stream area immediately deownstream of the leakage opening is

$$A_2 = \frac{C_1}{60\sqrt{2g}} \; .$$

The air stream area is related to the area of the leakage openings in the armor by assuming that the openings are sharp-edged orifices for which the air stream vena contracta effect is known. The cross-sectional area of a contracted air stream flowing from a sharp-edged orifice is about 0.62 of the sharp-edged orifice area and the average air velocity in the contracted stream section is about $0.99\sqrt{2g \; \Delta P/\rho_2}$. Equation (13), for these values, reduces to

$$m' = 294 \bar{A}_2 \sqrt{\rho_2 \; \Delta P} = C\sqrt{\rho_2 \; \Delta P} \; , \tag{13a}$$

and Equation (14) becomes

$$W = 294\ \overline{A}_2 \sqrt{\frac{\Delta P}{\rho_2}} = C \sqrt{\frac{\Delta P}{\rho_2}} . \qquad (14a)$$

The area of the openings in the armor surrounding the crew space is

$$\overline{A}_2' = \frac{C}{294} . \qquad (15)$$

A graph of Equation (13a) or of Equation (14a) is a straight line with slope C.

The air density ρ_2 was assumed to correspond to air density ρ_1 in the experimental work. The experimental data of Table XIII were used to plot Figure 13. The armor leakage coefficient C determined from Figure 13 is C = 56, and by Equation (15), $\overline{A}_2 \cong 27$ square inches.

The area $\overline{A}_2$ may be used to estimate the heat loss due to convective air leakage from the crew space of the buttoned-down combat tank when the personnel heater is in operation. The maximum heat loss by convective air leakage occurs when air leakage from the crew space leaves at highest velocity and temperature. Data from Test A, Table IIa, are used as an example. Highest crew-space ambient temperature recorded in Test A is 110°F, atmospheric temperature is 53°F, and the highest crew-space air velocity is 40 fpm. Using these data, the $\overline{A}_2$ value, and standard atmospheric pressure in the crew space, the heat loss from the crew space due to convective air leakage is 430 Btu per hour.

CONCLUSIONS

1. Operation of the personnel heater installed in the test vehicle results in a 16° to 19°F temperature rise in the turret-basket region of the crew space and a 38° to 49°F temperature rise in the cockpit regions. The air velocity of the personnel heater outlet duct approximates 1000 fpm. During heater operation the air velocity in the cockpit region was up to 31 fpm and usually less than 20 fpm in the turret-basket region.

2. During heater operation the distribution of the warmed air throughout the crew space is not uniform as indicated by the air temperature variation and low air velocities recorded in Tables IIa, b, and c.

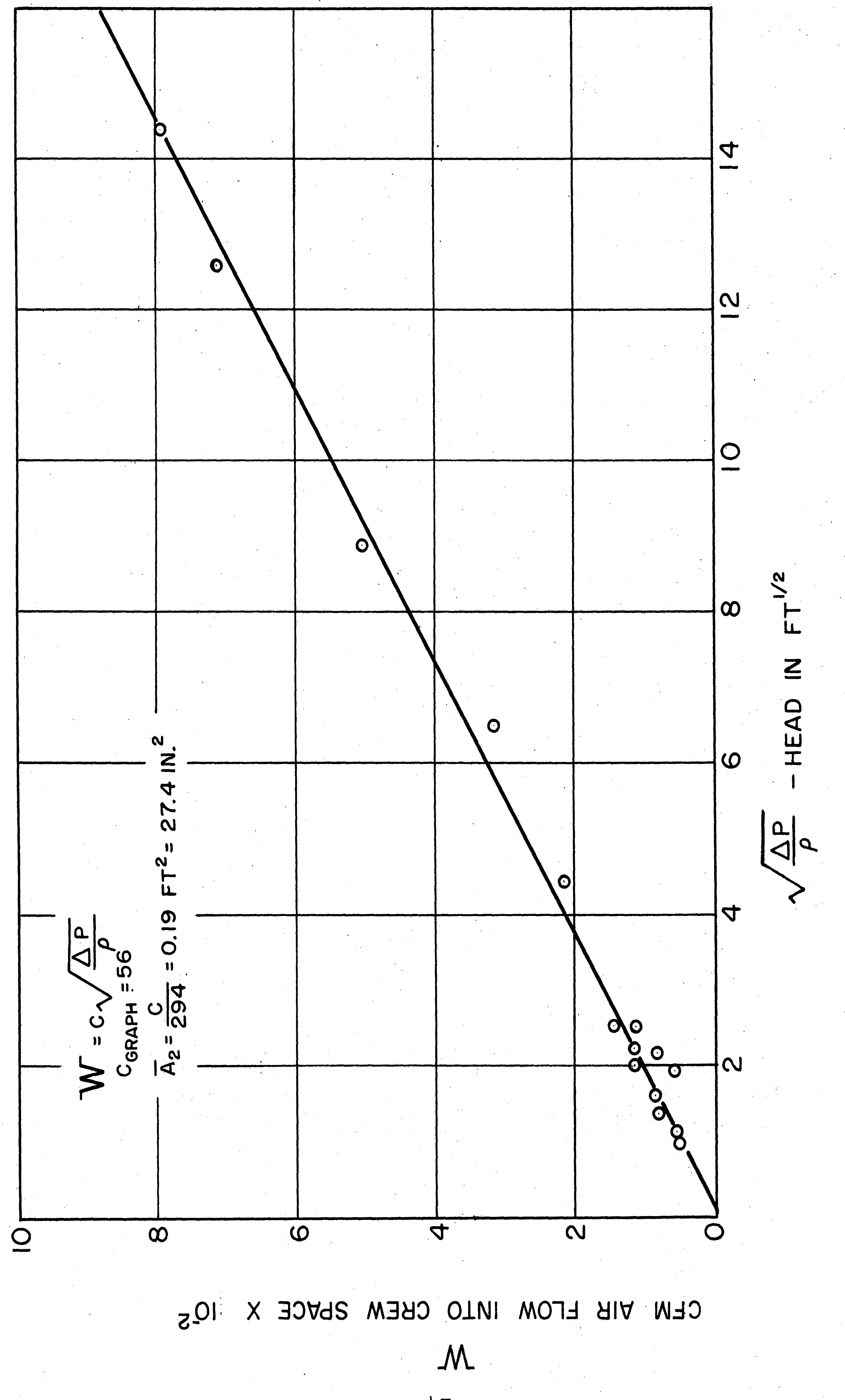


Figure 13. Air flow into crew space vs $\sqrt{\Delta P/\rho}$.

3. Heater performance as installed in the test tank is approximated by the expressions:

$$q_t = 675(t_t - t_A),$$

$$t' = 574\, W\left(\frac{\Delta t_h}{q_h}\right) - 460, \text{ and}$$

$$t'' = t' + \Delta t_h \,.$$

4. Personnel heater fuel consumption in U.S. gallons per hour is approximated by the expression $F = 0.125(10)^{-4} q_h$. If the maximum allowable fuel consumption is 1 gallon per hour, a heater with thermal capcity up to 80,000 Btu per hour may be used.

5. The total of the leakage openings in the buttoned-down test combat tank is 27 square inches. The expression $W = 56\sqrt{\Delta P/\rho}$ predicts the air-leakage rate through these openings. For one test, the heat loss by convection through the openings was 430 Btu per hour.

6. A heat addition of 60,000 Btu per hour in the crew space results in a calculated maximum temperature rise for the noninsulated armor surface surrounding the crew space of less than 2°F. In the absence of winds and other thermal effects, the 2°F temperature rise occurs in 34 minutes.

7. In the case of insulated armor, where the insulation is 1/4-inch-thick cork on the interior armor surface, a heat addition of 60,000 Btu per hour to the crew space results in a calculated maximum temperature rise for the exposed cork surface of 133°F. In the absence of winds and other thermal effbcts this temperature is reached in 16 minutes.

8. The crew-space cooling tests in which up to 1-1/2 tons of refrigeration effect were obtained and up to 145 cfm of cooled air were introduced into the crew space through a single opening in the turret bustle, produced a maximum crew-space temperature drop of 18°F with turret-basket air velocities up to 250 fpm and air velocities in the cockpit of 10 to 50 fpm. These maximum conditions did not give conditions satisfactory for body comfort due to nonuniform cooled air distribution and the high (over 50 fpm) crew-space air velocities.

9. At atmospheric conditions near the upper limit obtained in practice, and for an air-leakage rate from the crew space of 300 cfm, 6-1/2 tons

of refrigeration are required to maintain an 80°F crew-space ambient. For the same conditions, 3-1/2 tons of refrigeration effect are required to maintain the 80°F crew-space ambient when 1/4 inch of insulation [k = 0.04 Btu/(hr)(ft^2)(°F/ft)] is applied to the crew-space armor.

10. Insulation is thermally effective when placed on either surface of the crew-space armor. Insulation on the exterior surface of the armor introduces a time lag for both heating and cooling in a cold atmosphere. Insulation on the exterior armor surface, when the insulation has a reflectivity greater than that of the armor, assists crew-space cooling when solar radiation is an important thermal load.

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APPENDIX

ELECTRIC ANALOGS FOR COMBAT TANK HEAT TRANSFER

No application is made in this report of an electrical analogy for combat tank heat flow, but preliminary consideration was given to this area and is reported. The analogs were prepared in consultation with Dr. Myron Tribus, visiting Assistant Professor of Chemical Engineering and Director of Icing Research, 1952-1954, University of Michigan; now Associate Professor, University of California, Los Angeles.

A combat tank is visualized as a metal container having walls of variour thicknesses and orientations. The crew space and the engine space are considered to be adjacent subcontainers, as in Figure 14.

Consider the case where the crew space is cooled and the armor facing the crew is colder than human body temperature. The temperature state is maintained by refrigeration.

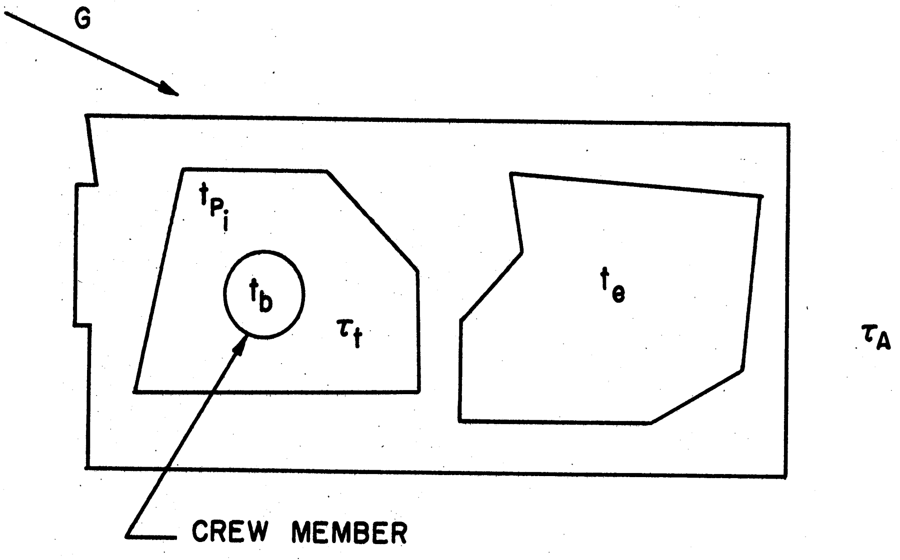

Figure 14. Schematic of a metal box as a thermal analog of a combat tank.

The possible ways for heat transfer to occur are:

(a) radiation - sun to tank armor, etc.,
(b) re-radiation - from armor to earth, surroundings and space,
(c) conduction - flow through the armor,
(d) convection - transport through openings,
(e) radiation, conduction and convection from main power plant,
(f) radiation, conduction and convection from auxiliary equipment and instrumentation,
(g) radiation and convection from human body to the armor, and
(h) evaporation of human body perspiration.

The general forms of the equations representing the transfers are:

$$\text{conduction:}\quad q = \frac{k\ A\ \Delta t}{\ell}\ ; \tag{16}$$

$$\text{convection:}\quad q = h_c\ A\ \Delta t;\ \text{and} \tag{17}$$

$$\text{radiation:}\quad q = \sigma\ \epsilon\ A\ \Delta T^4. \tag{18}$$

The equation for heat flow due to radiation may be expressed as

$$q = h_r\ F'\ A\ \Delta t. \tag{19}$$

For a refrigerated crew space where the armor facing the crew is colder than human body temperature, the radiation from the human body to the interior armor plate surface is

$$q_R = h_r A_b (t_b - t_{p_i}) . \tag{20}$$

The average body surface area of a human adult male usually is taken at somewhat less than 20 square feet (19.6). The radiation coefficient for the human body is assumed to be

$$0.9 \leq h_r \leq 1.2.$$

Cooling due to evaporation of human body perspiration is expressed by

$$q_V = h_b AL (e_b - e_t), \tag{21}$$

and for the resistance concept of conductive heat transfer

$$q_V = \frac{k\ A\ \Delta t}{L} = \frac{\Delta t}{L/kA} = \frac{\Delta t}{\bar{R}} . \tag{22}$$

Then

$$\bar{R}_c = \frac{\Delta t}{q_c} = f_1 \left[\frac{1}{h_c\ A_b}\right] , \tag{17a}$$

$$\bar{R}_R = \frac{\Delta t}{q_R} = f_2 \left[\frac{1}{h_r\ A_b}\right] , \text{ and} \tag{19a}$$

$$\bar{R}_V = \frac{\Delta t}{q_V}$$

$$= \frac{\Delta t}{h_b\ AL\ (e_b - e_t)}$$

$$= \frac{1}{h_b\ A} \left[\frac{t_b - t_t}{e_b - e_t}\right] \frac{1}{L} \tag{23}$$

By virtue of Equations (17a, 19a, and 23), a crude electrical analog representing the heat transfer concerned with the human body in the crew space of a combat tank is shown in Figure 15.

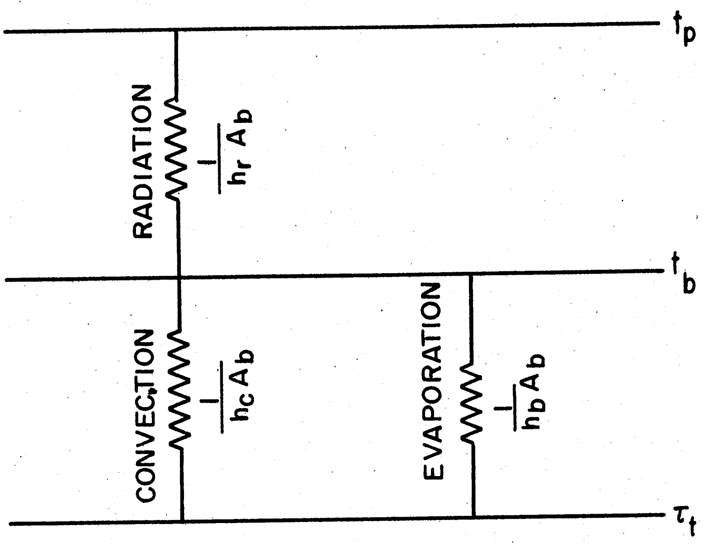

Figure 15. First electric analog for body heat transfer of tank crew member.

Heat flow for the "body circuit" of Figure 15 due to convection and radiation could proceed in either direction, but the heat flow from the human body due to the evaporative process must result in body cooling.

A more sophisticated "body circuit" is obtained by consideration of an idealization of the human body shown in Figure 16.

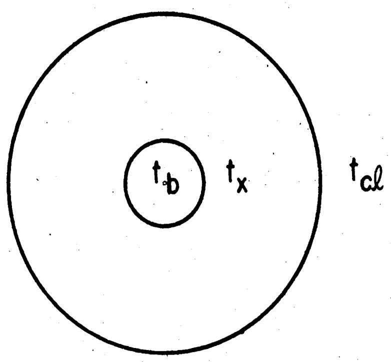

Figure 16. Thermal idealization of human body.

When the idealization of the human body of Figure 16 is surrounded by the combat tank crew space, the human "body circuit" could appear as shown in Figure 17.

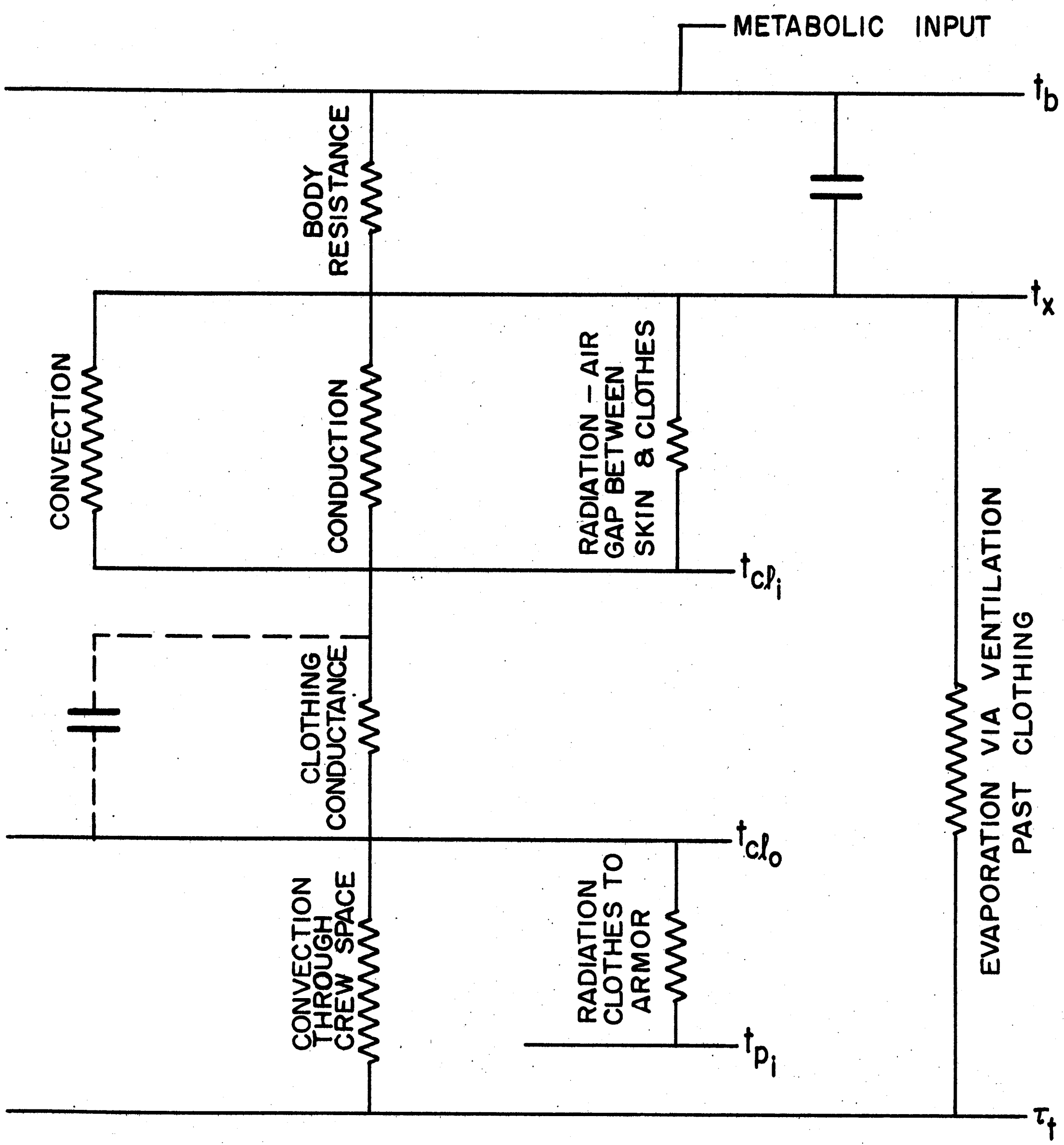

Figure 17. Second electric analog for body heat transfer of tank crew member.

The circuit in dotted line of Figure 17 is to account for transient conditions and considerably more of such accounting is required for an analogy for transient conditions.

Considerations of a similar nature yield a "crew-space circuit" of the form shown in Figure 18.

The subcircuits representing the inputs to the "crew-space circuit" may be complex. The electric analog for the input to the crew space from the main power plant is given as an example. Development not given previously

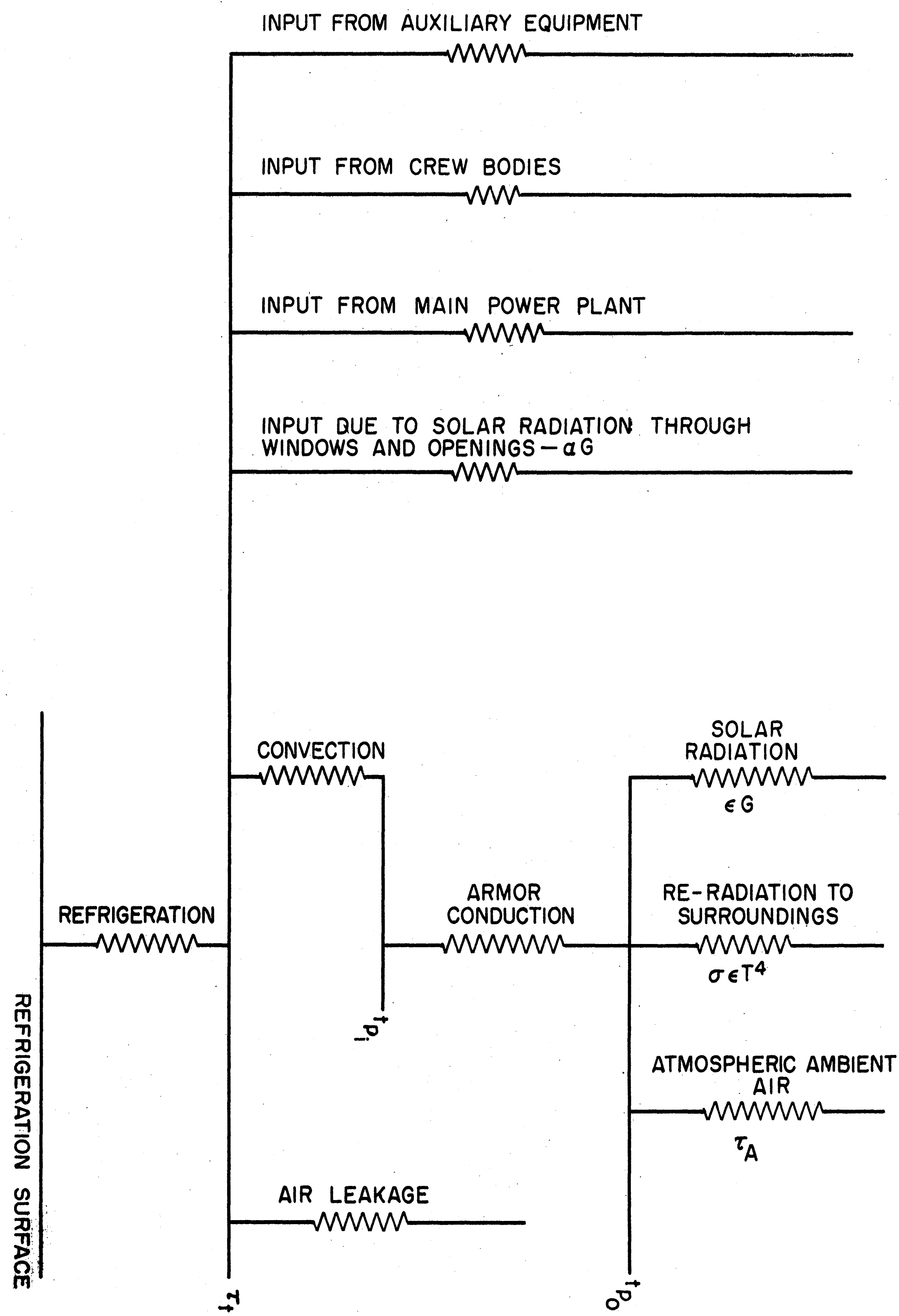

Figure 18. Electric analog for crew-space heat transfer.

concerns heat addition to the crew space due to passage of heated air from the engine compartment to the crew space via openings. The heat addition is

$$q = m''c\ (t_e - t_t). \tag{24}$$

In the electric analogy, the resistance representing the air leakage in question is proportional to 1/m"c. The electric analogy for the heat input to the combat tank crew space due to the main power plant is given in Figure 19.

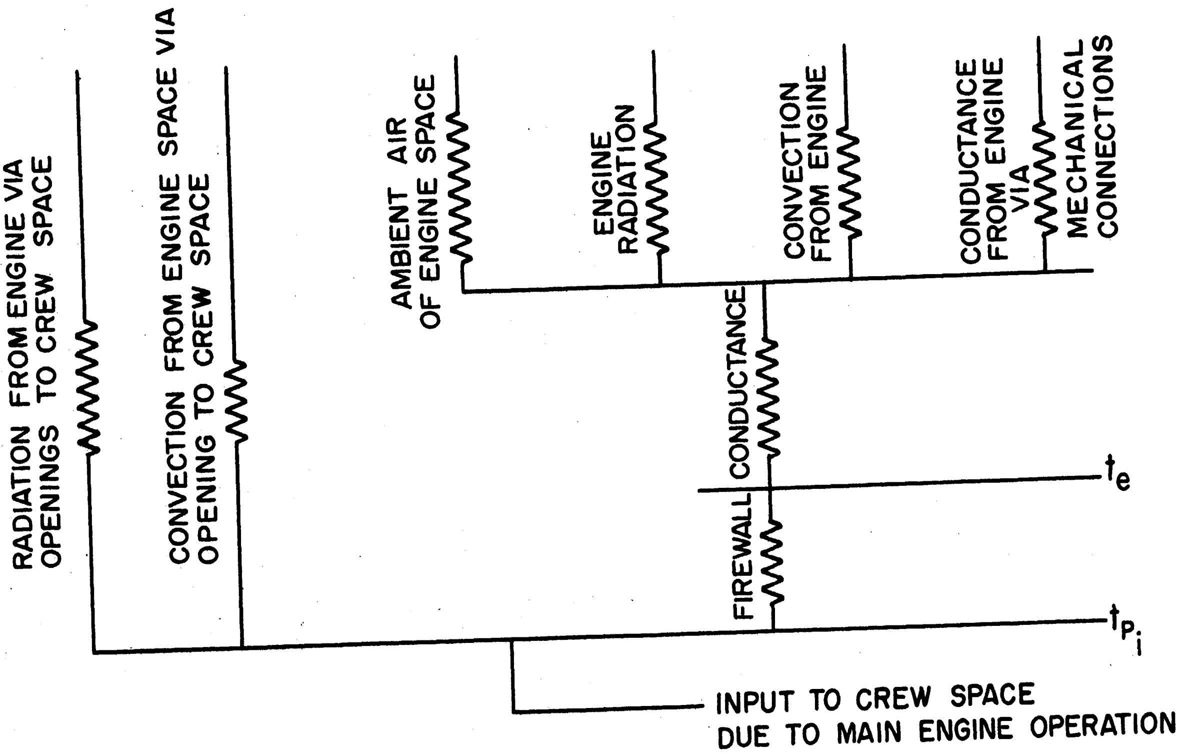

Figure 19. Electric analog for heat flow to crew space due to main power plant operation.

An analytic solution of the heat transfer for the crew space of a combat tank for even one set of conditions is a tedious operation. An electric analog of the heat flow permits ready solution of both the transient and steady-state conditions for any given problem. The analog circuits may be adapted for several combat tank designs. The complexity of the required circuits is a function of the degree of accuracy demanded. The major heat flows may be found by ignoring circuits representing comparatively low-valued heat transfers. Circuits which may be neglected are determined by comparison of calculated or feasible maximum values.