THE EFFECTIVENESS OF A CONDUCTIVE COOLING HOOD AT VARIOUS FLOW RATES AND INLET TEMPERATURES

by

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TABLE OF CONTENTS

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LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

A = total inside wetted surface area of tubing (Burton)

A = total contact surface area between tubing and skin (Richardson)

C = heat transferred by convection from the outer surface of the clothed body

c = specific heat of the cooling medium

i = insulation

D = OD of tubing

d = ID of tubing

D_s = heat loss by water vapor diffusion through the skin

f_{cl} = ratio of the surface area of the clothed body to the nude body

h = heat transfer coefficient of the tubing (Btu/hr - ft^2 - F)

h_w = heat transfer coefficient of water next to the tube wall

h_{skin} = heat transfer coefficient of the skin

I_{cl} = heat transfer from skin to outer surface of clothed body

K = heat transfer from skin to outer surface of the clothed body (conduction through the clothing)

k = thermal conductivity (Btu/hr - ft - F)

L = length of a tubing circuit

ℓ = distance between tubing (center to center)

M = internal heat production (via metabolism)

m = mass flow rate of the cooling medium

N = external mechanical efficiency of the body

n = number of tubing circuits connected in parallel in a cooling garment.

Q = heat absorbed by the cooling garment

R_w = wet respiration heat loss

R_d = dry respiration heat loss

R = heat transferred by radiation from the outer surface to the clothed body

S_w = heat loss by evaporation of sweat from the surface of the skin

s = conductive shape factor ($s = 6.3\psi + 1.1$)

T_{in} = inlet temperature of the cooling medium

T_s = average skin temperature over the body

T_{out} = outlet temperature of the cooling medium

u = overall heat transfer coefficient between skin and water (Btu/hr-ft^2-F)

v = air velocity

W = ℓ/ZD

Z = most efficient tube spacing in terms of tube outside diameter

ε = ratio between actual heat transfer and maximum possible heat transfer (garment effectiveness)

ψ = fraction of outside circumference of the tubing that contacts the skin

ϕ = heat flux

INTRODUCTION

Man's ability to maintain his thermal equilibrium determines his ability to survive and function in hostile thermal environments. Man is unique among homeotherms, beings which maintain a constant internal body temperature, in his ability to augment his physiological systems for controlling body temperature. Since the earliest recorded history, man's ability to protect himself from the elements has marked not only the progress of civilization but also the individual's social status among his peers. Today man has learned to survive in space and in the sea, but while we challenge these exotic environments, the mundane environments are challenging us. The economic difficulty of controlling some environments has led to an axiom. "If you can't cool the environment, then cool the man," which might otherwise be stated: "If you can't cool the macroenvironment, then cool the microenvironment."

Devices used to cool the microenvironment are called thermal conditioning systems and, while this terminology may bring to mind a sophisticated space suit, it can be applied with equal validity to an old overcoat. A space suit would likely be an "active" type of thermal conditioning system because it would have an energy input other than the body while an overcoat is of the "passive" type using only insulation or reflectivity (or absorptivity). Active, water cooled garments that cover the entire body, such as used in the space program, have proven very efficient and effective in maintaining the thermal equilibrium of the human body under a wide range of activity levels. Often what is required for use in industrial situations such as tending furnaces and deep mining does not require the sophistication or permit the expense of a full suit.

The idea of cooling the head by conduction was introduced to KSU by Victor Morales (1967). He pointed out some of the advantages of cooling the head as opposed to other parts of the body: 1) the head is relatively accessible (people seldom wear clothes on their heads and what they do wear is easily removed), 2) the head is less subject to vasoconstriction than other parts of the body, and 3) encumbering the head with a cooling garment may not be as restrictive to freedom of movement as it would be to some other part of the body.

LITERATURE SURVEY

Environment

The environmental variables which affect a man's ability to work comfortably are numerous. They have been represented by Rohles (1967) in the form of a cube. The axes represent reciprocative factors, physical factors, and organismic factors.

Reciprocative Factors	Physical Factors	Organismic Factors
Diet	Sound	Age - Sex
Clothing	Light	Rhythmicity
Exposure	Area-Volume	Psyche
Social	Radiation	Drive
Incentive	Inspired Gas	Body type
Activity	Atmospheric Press.	Sensory Process
	Force Field	Genetics
	Air Movement	
	Temperature - R.H.	

To experimentally evaluate all of these variables and their possible interactions would be a momentous task. Thus a model or equation is required.

The "thermal" environment is the smaller part of the total environment described by those variables which are thought to be the most significantly associated with mechanical heat transfer between an organism and its surroundings.

Thermal environmental studies run the gamut from cold to hot. Relevant to this project is only the upper half of this scale - from thermal neutrality, or comfort, to heat stress. The basic questions asked in comfort studies are: 1) What environments are comfortable? Since the experimentation required to answer this question completely would still cost far too much in time and effort the next question asked is 2) What mathematical relationship between the thermal variables can be used to predict thermal comfort? The third question acknowledges individual differences and asks 3) What is the comfort distribution about a given comfortable environment condition?

The heat balance equation (Fanger, 1969) describes the physiological channels through which heat produced by metabolism is transferred to the environment.

$$M - D_s - S_w - R_w \pm R_d = K = + R \pm C$$

M = internal heat production (via metabolism)

D_s = heat loss by water vapor diffusion through the skin

S_w = heat loss by evaporation of sweat from the surface of the skin

R_w = the latent respiration heat loss

R_d = the dry respiration heat loss

K = heat transfer from the skin to outer surface of the clothed body (conduction through the clothing)

R = the heat transferred by radiation from the outer surface to the clothed body

C = the heat transferred by convection from the outer surface of the clothed body.

Fanger (1969) has developed a comfort equation. The principal variables in his equation are:

1) air temperature
2) humidity
3) mean radiant temperature
4) relative air velocity
5) activity level
6) insulation value of clothing.

Although conceptually simple and logical since it is based upon the heat balance equation, the actual solution to Fanger's comfort equation is somewhat involved so he provides graphical solutions that have already been worked on a computer. Figure 1 shows an example of a graphical solution. Fanger points out that, although the proper solution of his equation is necessary for thermal comfort, it is not sufficient. It might be possible, for example, to specify such high values of air temperature, humidity, mean radiant temperature, activity level, and clothing insulation value that no amount of air circulation would allow a person to be comfortable.

Conduction Cooling Suits

Burton and Collier (1964) reported a theoretical comparison of air and water for use in a thermal conditioning garment. Water was calculated to be better than air because its higher specific heat would require much lower pumping power. However, the confinement of water to small

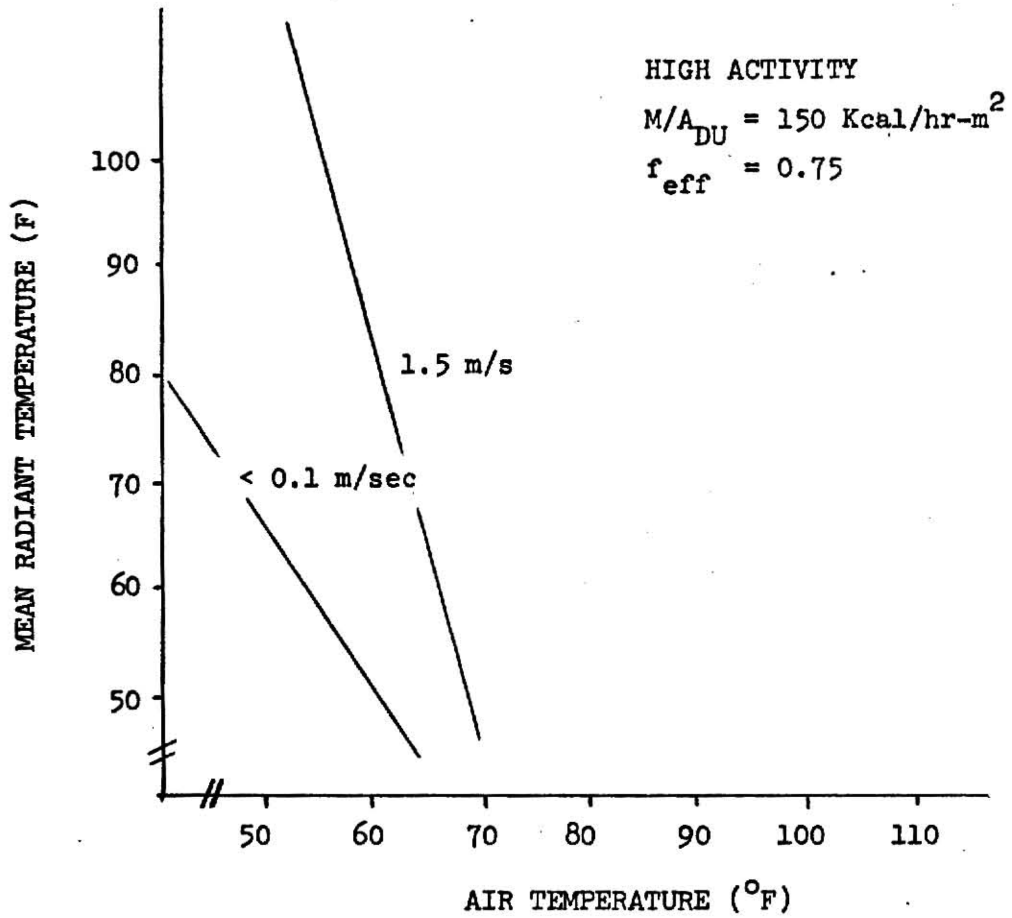


Figure 1. Comfort lines (air temperature versus mean radiant temperature with relative air velocity as a parameter) for persons with light clothing (I_{cl} = 0.5 clo, f_{cl} = 1.1) at three different activity levels (RH = 50%, N $\simeq$ 0).

where:

I_{cl} = Heat transfer from skin to outer surface of clothed body

f_{cl} = Ratio of the surface area of the clothed body to the nude body

N = External mechanical efficiency of the body.

tubes, not necessary in the case of air which could be blown directly on the body, appeared to be a problem of potential significance so they decided to build test suits.

Their first prototype had forty parallel tube circuits, each four feet long with 1/16" ID and 1/32" wall, which were woven into a loose weave one piece garment similar to long underwear. Various pilot experiments were done at flow rates from 16 to 50 lb/hr and temperatures from 55 F (13 C) to 77 F (25 C). Inlet and outlet temperatures, flow rate, pressure drop, and subjective comments by the subjects were recorded. Low metabolic rates were used (sitting and standing work tasks) in conjunction with heavy insulation from the environment. The results were encouraging in that it was found not difficult to keep the subjects comfortable or even cooler than comfortable.

The second prototype was similar to the first with forty parallel tube circuits but six instead of four foot tube lengths. Typical results of testing showed 117 Btu (29 Kcal/hr) absorbed at a flow rate of 22 lb/hr and inlet temperature of 68 F (20 C).

Burton and Collier (1965) developed a mathematical formulation for suit performance:

$$Q = m\ c\ \varepsilon\ (91.5 - T_{in}) \qquad (1)$$

where: Q = heat absorbed by the suit, Btu/hr

m = mass flow rate, lb/hr

c = specific heat of the cooling medium, Btu/lb-F, (1 for water)

91.5 = the assumed average body skin temperature which would then be the outlet temperature (T_{out}) if the water actually warmed to skin temperature before exiting

T_{in} = the inlet temperature of the cooling medium, F

and
$$\varepsilon = 1 - e^{\left(-\frac{A\ U}{m\ c}\right)} \tag{2}$$

Burton (1966) explained ε in the following words:
"The temperature of an element of water which is initially different from skin temperature by the amount ΔT_o, will approach skin temperature exponentially with time. A simple theoretical analysis shows that the local difference in temperature, ΔT, at suit outlet between a stream of water in a pipe, and the skin surface is related to ΔT_o by an equation of the form:

$$\Delta T = \Delta T_o\ e^{\left(-\frac{A\ U}{m\ c}\right)} \tag{3}$$

where: U is the overall heat transfer coefficient between

skin and water $\frac{(Btu)}{(hr\ ft^2\ F\ )}$

A is the total wetted area of the tubes in the suit (ft^2)

Consider a suit whose inlet and outlet temperatures are T_{in} and T_{out}, respectively

$$\Delta T_o = T_s - T_{in}$$

$$\Delta T = T_s - T_{out}$$

where: T_s represents mean skin temperature

$T_{out} - T_{in}$ = the temperature rise through the suit

$T_s - T_{in}$ = the maximum possible temperature rise

Defining the temperature effectiveness, ε, of the suit such that

$$\varepsilon = \frac{T_{out} - T_{in}}{T_s - T_{in}}$$

we obtain $\varepsilon = 1 - \frac{\Delta T}{\Delta T_o}$

and using equation (1)

$$\varepsilon = 1 - e^{\left(- \frac{A\ U}{m\ c}\right)} \tag{4}$$

This is an important result which expresses the relationship between the temperature effectiveness of the suit, as defined above, and its basic thermodynamic properties. It is convenient to visualize the effectiveness, ε, as the ratio between the actual heat transfer and the maximum possible heat transfer of the water stream. The quantity AU can be considered to represent the thermodynamic size of the suit as a heat exchanger." $A = \pi dLn$; d = inside diameter of the tubing, L = length of tubing used per circuit, and n = number of circuits.

Richardson (1967) modified Burton's equation by letting $A = \psi\pi DLn$ where ψ is the fraction of the outside circumference of the tubing that contacts the skin, D is the outside diameter of the tubing and Ln is the length of the tubing. Therefore, in Richardson's equation, A is the total amount of contact area between the <u>outside</u> of the tubing and the skin while Burton's A is the total wetted surface area <u>inside</u> the tubing. Richardson estimates ψ, U, and the components that contribute to U as follows:

"(1) Estimate of ψ, fraction of tube touching skin. The portion of the circumference of a relatively soft plastic tubing in contact with the skin can be measured, and this calculation determined ψ. We carried out experiments with tubes varying in diameter from 1.6 to 10 mm OD and found ψ to vary from about 0.26 to 0.29. An average of 0.27 was chosen.

(2) Estimation of U, heat transfer coefficient, Btu/sq. ft.-hr-F. We estimated U from the design values of the Hamilton-Standard liquid cooled garment (Jennings, 1966, and Kincaide, 1965) and arrived at a conductance of 71 Btu/hr - F for a cooling load of 2000 Btu/hr. The actual heat transfer area, including the ψ values discussed above, gave about 2.56 ft^2. Thus, the experimental value for the overall heat transfer coefficient is $U = 71/2.56 = 27.8$ Btu/hr-ft^2-F. Values of U estimated from more fragmentary data were in the same range.

(3) Estimation of h, liquid-side coefficient of heat transfer. The data used to estimate U were used also to determine a liquid-side heat transfer coefficient from standard correlations. We found the flow to be laminar but h values as high as 250 Btu/hr-ft^2-F. Note that $h \gg U$ so that the liquid-side heat transfer is not controlling.

(4) Tube thermal resistance. The equivalent thermal resistance for that portion of the plastic tube which touches the astronaut is simply k/x, where k is the thermal conductivity of the tube wall and x is the thickness. (k values range around 0.07 to 0.10 Btu/hr-ft-F and x values about 1/32 inch or 2.6×10^{-3} ft.) Thus, k/x ranges from about 27 to 38 Btu/hr-ft^2-F. This value is close to the measured overall heat transfer coefficient ($U = 27.8$ Btu/hr-ft^2-F). Thus, the resistance between the skin and the tube must be small (skin heat transfer coefficient must be large),

and the conductance of the plastic tube wall is heat flow controlling heat-flow resistance."

In his discussion he emphasizes again that it is the tubing wall which offers the most resistance to heat flow from the cooling medium to the body.

Buchberg and Harrah (1968) used a multilayered slab model to develop a relationship between body metabolism and water inlet temperature for a given flow rate which can be expressed as follows:

$$Q = m\,c\,(1 - e^{\left(-\frac{skLn}{m\,c}\right)})(T_s - T_{in}) \qquad (5)$$

where T_s = skin temperature, s = conductive shape factor which was determined experimentally to be related to ψ by $s = 6.3\psi + 1.1$, and k = tube thermal conductivity. Figure 2 shows a graph of the slab model consisting of a skin zone, functional periphery, musculature, and core. Note that the functional periphery is that part of the body next to the skin that is subject to vasoconstriction and vasodilation. Figure 3 shows a graph of the relationship expressed in equation 5 as shown in Buchberg and Harrah's study.

Webb and Annis (1967) reported efforts to evaluate the various physiological and biological responses to change in work rate and therefore metabolic heat production in order to determine which one or ones might be used most advantageously as a cue for an automatic controller for governing heat extraction from the human body in an extravehicular situation. A suit made from 1/16" ID, 1/32" wall, Tygon tubing was used in the study. Flow rates from 26 lb/hr to 260 lb/hr were tested in the

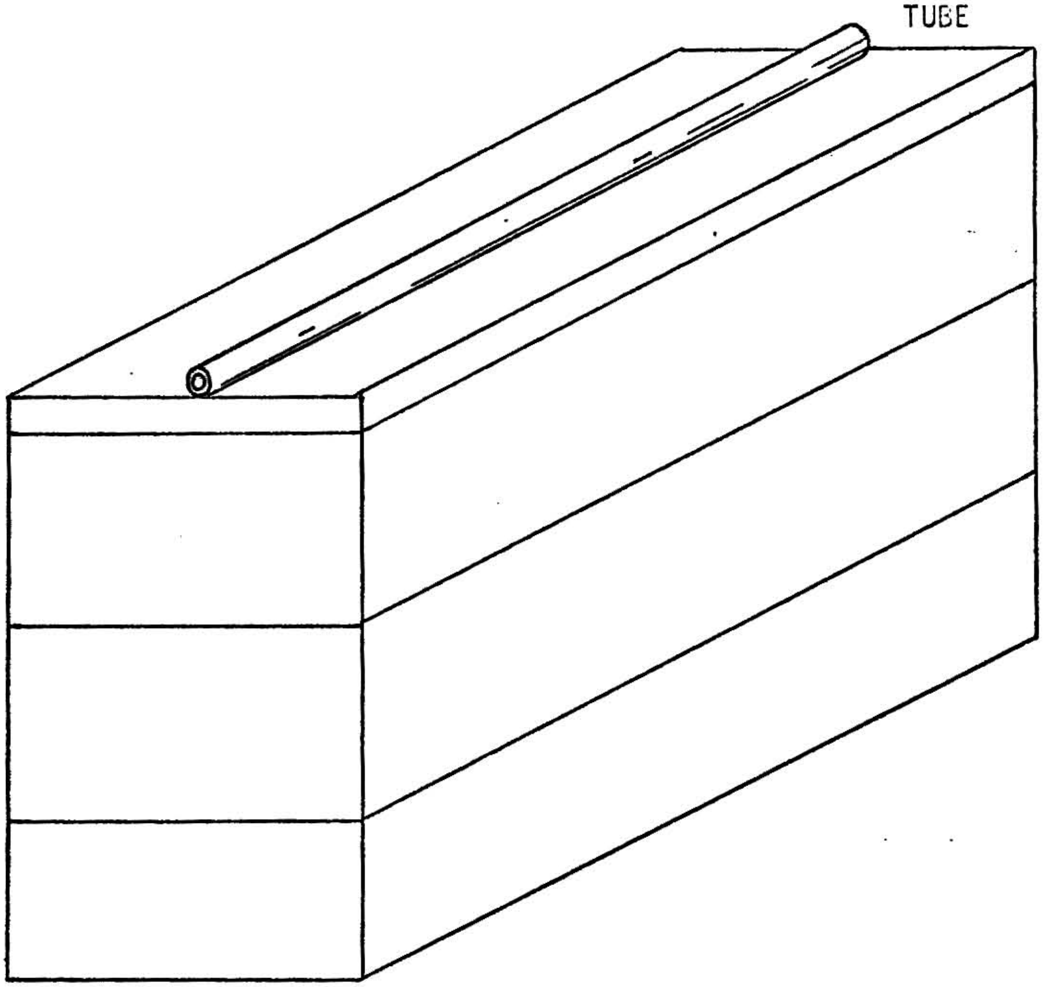


Figure 2. Multilayered slab model used for studying heat flow in the body as depicted by Buchberg and Harrah.

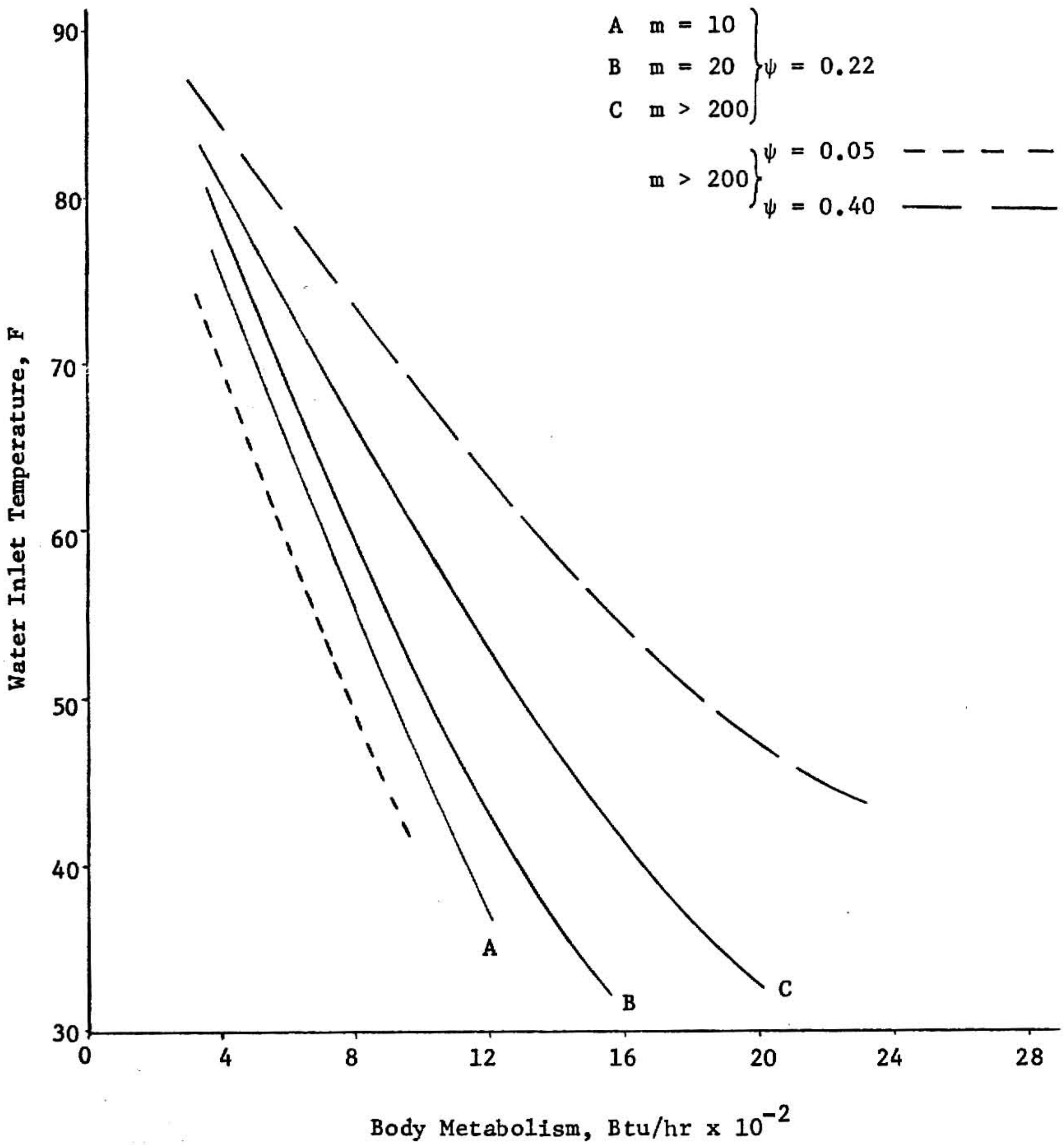


Figure 3. Predicted water inlet temperature as a function of metabolism. ψ refers to the contact surface factor. Buchberg and Harrah (1968).

initial eleven experiments and they were found not to be as critical as inlet temperatures so a constant flow of 200 lb/hr was used for the remaining 19 experiments.

Oxygen uptake was concluded to be the best indicator of the amount of cooling required or at least that a change in cooling may be needed. Sweat production and skin temperatures were also fair indicators. Heart rate was also recognized to be a good indicator except for its potential response to changes other than metabolic heat generation.

Jennings (1966) built and tested a cooling suit that was similar to Burton's in that it was constructed of transparent polyvinalcholride (PVC) tubing stitched into an open weave elastic cotton underwear type garment. Jennings used 300 feet of tubing, 3/16 ID, 1/16 wall, in 48 parallel circuits. Testing indicated that subjects could be maintained in thermal equilibrium to metabolic heat generation rates in excess to 2000 Btu/hr (500 Kcal/hr) and also that sweating could be limited to 0.22 lb/hr (100 gm/hr) or lower for metabolic rates at least up to 1600 Btu/hr (400 Kcal/hr).

Webb and Annis (1968) experimented with both air and water cooling to investigate the feasibility of limiting sweat rate to 0.22 lb/hr (100 gm/hr) at various metabolic heat production rates. Four subjects were tested at three activity levels: 1200, 2400 and 3600 Btu/hr (300, 600, and 900 Kcal/hr). The water cooled suit consisted of loosely knit, 1/16" ID, 1/32" wall, Tygon tubing worn under heavy impermeable insulation. Water inlet temperature was varied according to indications given by:(1) moisture content of a small amount of air forced through the suit under the insulation, (2) skin temperatures, and (3) subjective comment from the subject. Water flow rate was maintained constant at 200 lb/hr.

Results indicated that if the body was cooled sufficiently to remove 80% of the heat produced by the body then sweating could be kept at or below 0.22 lb/hr (100 gm/hr).

Conduction Cooling Hoods

Froese and Burton (1957) measured the heat extracted from the human head using calorimetry techniques. Three subjects were tested while sitting in a temperature controlled room. They wore heavy insulation covering all of the body except the head. The calorimeter apparatus was fitted over the head. Air temperatures from -6 F (-21 C) to 84 F (29 C) were tested. Heat absorption ranged linearly from 20 Btu/hr (5 Kcal/hr) to 168 Btu/hr (42 Kcal/hr) indicating that the head appears not to vasoconstrict in response to the cold under these conditions. Exposure times were from 30 to 45 minutes hence the maximum heat a subject lost would have been approximately 126 Btu (32 Kcal).

Nunneley, Webb, and Troutman (1970) and Nunneley (1970) reported the testing of a water cooled conduction cap. The cap consisted of 43 feet of small bore plastic tubing. Test environments of 68, 86, and 104 F (20, 30, and 40 C) were studied while subjects performed treadmill tasks. Measurements taken were oxygen consumption, heart rate, rectal and ear temperatures, and weights.

Results indicated that the heat absorbed by the cap was more independent of inlet temperature and flow rate than previously expected and more dependent upon the metabolic rate or heat storage within the body. An extraction rate of 120 Btu/hr (30 Kcal/hr) was the maximum attainable while the subjects were at rest. With the subjects working, a maximum heat extraction rate of 360 Btu/hr (90 Kcal/hr) was observed. It was observed also that the maximum heat absorption during work occurred only after the subjects rectal temperature had begun to rise. This implies that the head may be able to vasoconstrict and otherwise control its heat loss.

Morales and Konz (1968) reported the experimental evaluation of the first KSU cooling hood (Model A). Cooling hood A had 23' of polyethelene tubing cemented in two parallel circuits on a shell of medium weight canvas duck. The tubing had a 3/16" ID and 1/32" wall thickness. Hood flow rate averaged 150 lb/hr. Water inlet temperature was not measured at the hood inlet but water was supplied from a tank of ice water and so inlet temperature was likely between 35 and 40 F (1.6 and 3.3 C). (See Morales (1967) for further details.)

The hood was tested on two subjects in two environments with an intermittent pedalling task on a bicycle ergometer. Total exposure time was four hours. One environment was "comfortable" with an air temperature of 76 F and 50% RH. The other environment was "stressfull" with an air temperature of 100 F and 70% RH. The experiment demonstrated the cooling hood to be feasible in reducing physiological stress parameters such as sweat loss, heart rate, skin and rectal temperature, with an accompanying increase in endurance.

Konz and Gupta (1969) at KSU reported the testing of cooling hood Model B. Model B was made from a hot water pad which consisted of two thin flexible rubber sheets with molded channels. (See Gupta 1969 for further details.) Eight male subjects were tested with and without the hood in the same comfortable and heat stress environments used by Morales. The task used was an anagram problem involving creative mental work.

Statistically significant ($p < .05$) physiological differences were found between the hood and no hood conditions in heat stress. Statistically significant ($p < .05$) drops in creativity were noted from neutral environment performances to heat stress performances without the hoods; but with the hood the drop from neutral environment performances to heat stress performances was not significant.

Konz and Duncan (1969) at KSU tested hood Models C and D. Hood C had 27' of Excelon tubing, 3/16" ID and 1/32" wall, sewn into a Nomex cloth shell. The tubing was divided into two equal length circuits. Hood D had 22' of Tygon tubing, 3/16" ID and 1/16" wall sewn into a Nomex cloth shell. The same circuit design was used on both hoods. Both hoods were insulated on the sides with foam rubber. (See Duncan (1969) for further details.) Duncan used an estimated flow rate of 260 lb/hr and an inlet temperature of 60 F (16 C). The test environment was 112 F (45 C) and 58% RH. Twelve sitting subjects were tested. The task was to add columns of numbers on printed sheets.

A typical heat flow calculation showed that a subject when wearing hood C gained 446 Btu/hr (112 Kcal/hr) from metabolic generation, 188 Btu (47 Kcal) from absorbed radiation, and 139 Btu (39 Kcal) from convection. The subject lost 271 Btu (69 Kcal) from sweat evaporation and lost 245 Btu (61 Kcal) due to conduction to the hood. Summing the values in this instance shows that the subject gained or stored heat at a rate of 271 Btu/hr (68 Kcal).

PROBLEM

Briefly summarizing the work at KSU, the Morales and Konz report showed that a water cooled hood was able to provide significant physiological assistance to a heat stressed subject. The Konz and Gupta report showed that a cooling hood could provide psychological benefit, specifically, to enhance creativity in a heat stress environment. The Konz and Duncan report makes a detailed comparative examination of the heat flow between environment, subject, and cooling hood for two slightly different hood designs.

It is the purpose of this project to evaluate a water cooled conduction hood's ability to remove heat from a person in mildly warm environments at a combination of three water inlet temperature and two flow rates and to evaluate the mathematical model

$$Q = m\ c\ \varepsilon\ (91.5 - T_{in})$$

which might be used to compare the effectiveness of different cooling hood designs.

METHOD

Task - A step task (see Figure 4) was used that required a metabolic rate of approximately 1650 Btu/hr (410 Kcal) while walking and 400 Btu/hr (100 Kcal) while seated. (See McNall, et. al. (1968).) The subject made two steps up (9" per step), brought his feet together and made two steps down, and brought his feet together again. A metronome was set to tick once each second and the subjects were instructed to "Step at a rate such that a foot makes contact with each tick of the metronome." A complete cycle took six seconds.

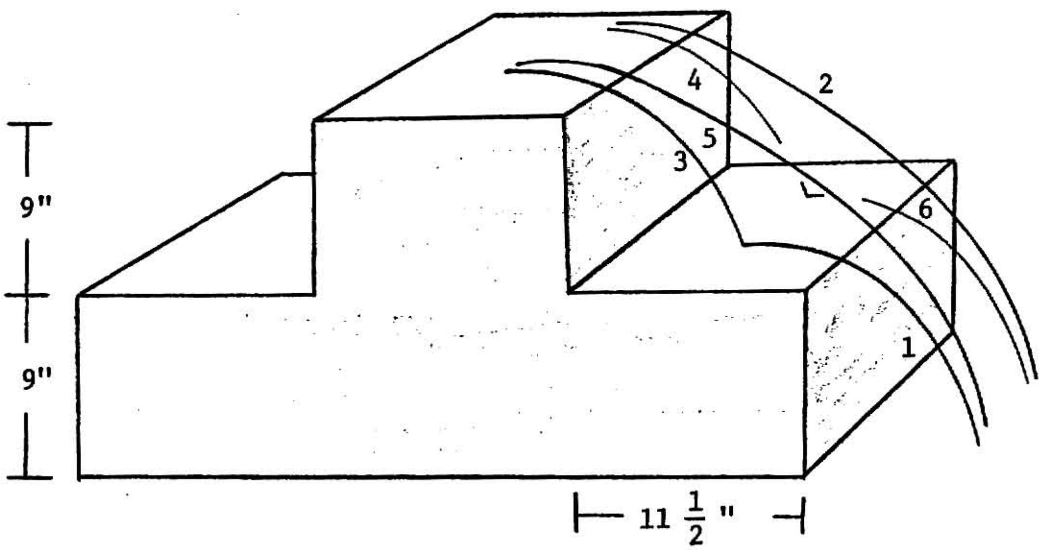


Figure 4. Illustration of step task sequence; the subject took the first step with the left foot.

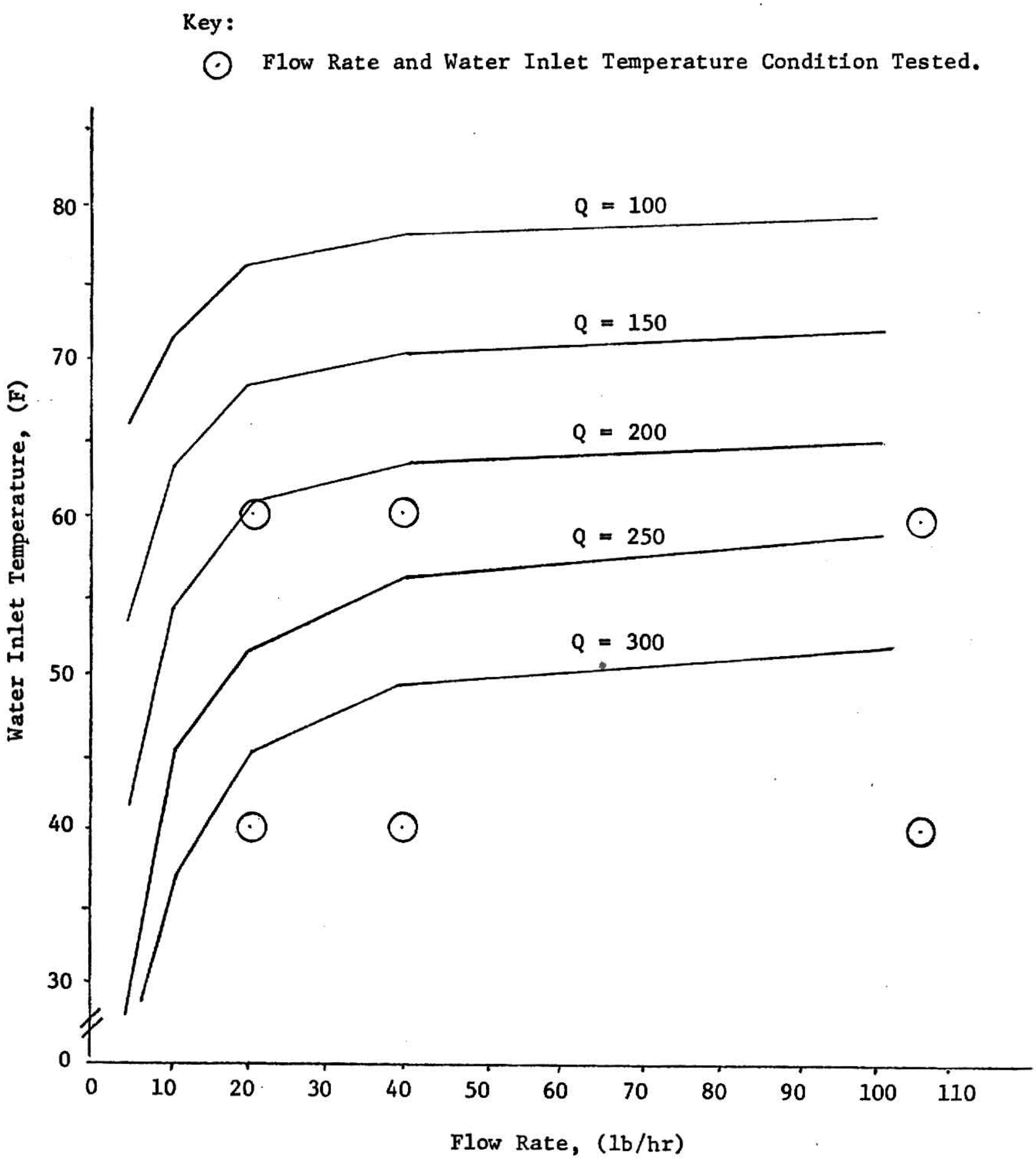


Figure 5. Heat removal as predicted by Burton's suit performance equation. The family of Q curves is developed from the equation

$$Q = (91.5 - T_{in})\, m\, c\, \left(1 - e^{\left(-\frac{A\,U}{m\,c}\right)}\right)$$

where T_{in} is the water inlet temperature, m is the mass flow rate, c is the specific heat of water, and A U = 8 Btu/hr - F.

Subjects - Four U.S. citizen, college age males were used. See Table 1. For all subjects, hair length was from one to two inches long on top of the head and trimmed on the sides.

Procedure - Figure 5 shows a graph of the six combinations of flow rate and inlet temperatures that were tested. The experimental value of A U from the suit performance formula

$$Q = m\ c\ (91.5 - T_{in})(1 - e^{-\left(\frac{A\ U}{m\ c}\right)}) \qquad (5)$$

was found, for Duncan's hood model C, to be close to 8. With A U = 8 the lines of constant heat absorption (Q) would have a pattern shown in Figure 5; there is greater change of Q with respect to inlet temperature for low flow rates than for higher flow rates, hence the rates 20 and 40 lb/hr were chosen for study. The maximum capacity of the flowmeter, 107 lb/hr, was chosen as the third value.

Environment - Two environments were chosen for study, 80 and 100 F dry bulb with 50 % RH.

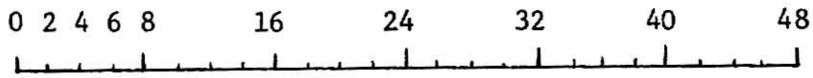
Experimental Design - A test run (see Figure 6) consisted of one subject being tested at one inlet temperature and six flow rates. Two flow rate sequences were used: Type A run used a sequence of 20, 40, 107, 107, 40, and 20 lb/hr while a type B run used a sequence of 107, 40, 20, 20, 40, and 107 lb/hr. During each condition the subject stepped for two minutes, rested sitting for two minutes, stepped for two minutes, and again rested for two minutes. Inlet and outlet temperatures were measured immediately

Table 1.

Age, height, and weight of subjects.

Subject	Age, years	Height, inches	Weight, pounds
RC	24	71	148
SV	25	72	180
GK	25	70	170
AC	21	73	154

TIME (minutes)



TYPE A RUN

20				40				107				107				40				20			
S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R

TYPE B RUN

107				40				20				20				40				107			
S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R	S	R

Key:

20 = 20 lb/hour flow

40 = 40 lb/hour flow

107 = 107 lb/hour flow

S = Step

R = Rest

Figure 6. Illustration of the sequence in which flow rates were studied within a run. During the first eight minutes 20 lb/hr flow rate was tested for a type A run and 107 lb/hr was tested for a type B run.

at the end of each two minute step task and heart rate was measured after one minute of stepping. Approximately twenty seconds of heart rate data was taken at each measurement. Thus there were four readings for each flow rate condition. Table 2 shows the run type used by each subject at each environment and inlet water temperature combination.

EQUIPMENT

Environment - Experimentation was done in a Controlled Environment Room, Model CER 812, produced by the Sherer-Gillett Company.

Hood - The hood used has 48 feet of 1/8" ID and 1/32" wall natural rubber (amber colored) tubing. The tubing was arranged into two symmetrical nonoverlapping circuits, one covering each side of the head. See Figure 7. The tubes spiral outward from the region of the ear and are fastened with hot melt glue to 1/4" elastic strips which run perpendicular to the tubes. Insulation was provided with a separate covering made from a sheet of foam rubber one inch thick. See Figure 8. The hood holds approximately 0.6 lb of water when full. This means that an incremental amount of water circulates through the hood in 108 seconds with a flow rate of 20 lb/hr (0.2 ft/sec), 54 seconds with a flow rate of 40 lb/hr (0.5 ft/sec), and 22 seconds with a flow rate of 107 lb/hr (1.2 ft/sec). See Table 3 for a physical comparison of KSU hoods A, B, C, D, and E.

Water Supply - Cold water temperature was maintained from a water reservoir cooled by a 1/3 hp compressor-condensor unit. Water pressure was supplied by a Little Giant submersible pump, heavy duty, model 3-12R. Flow rate was controlled with a Manostate Predictability Flowmeter with a glass beat float. Control range with the glass bead is from zero to 107 pounds

Table 2.

Type of flow rate sequence used by each subject to test each ambient temperature and inlet water temperature.

Room Temp.	Nominal Water Inlet Temp.	Subjects RC	SV	GK	AC
80	60	A	B	A	B
80	40	B	A	B	A
100	60	A	B	A	B
100	40	B	A	B	A

Figure 7. Photograph of hood E without the insulation.

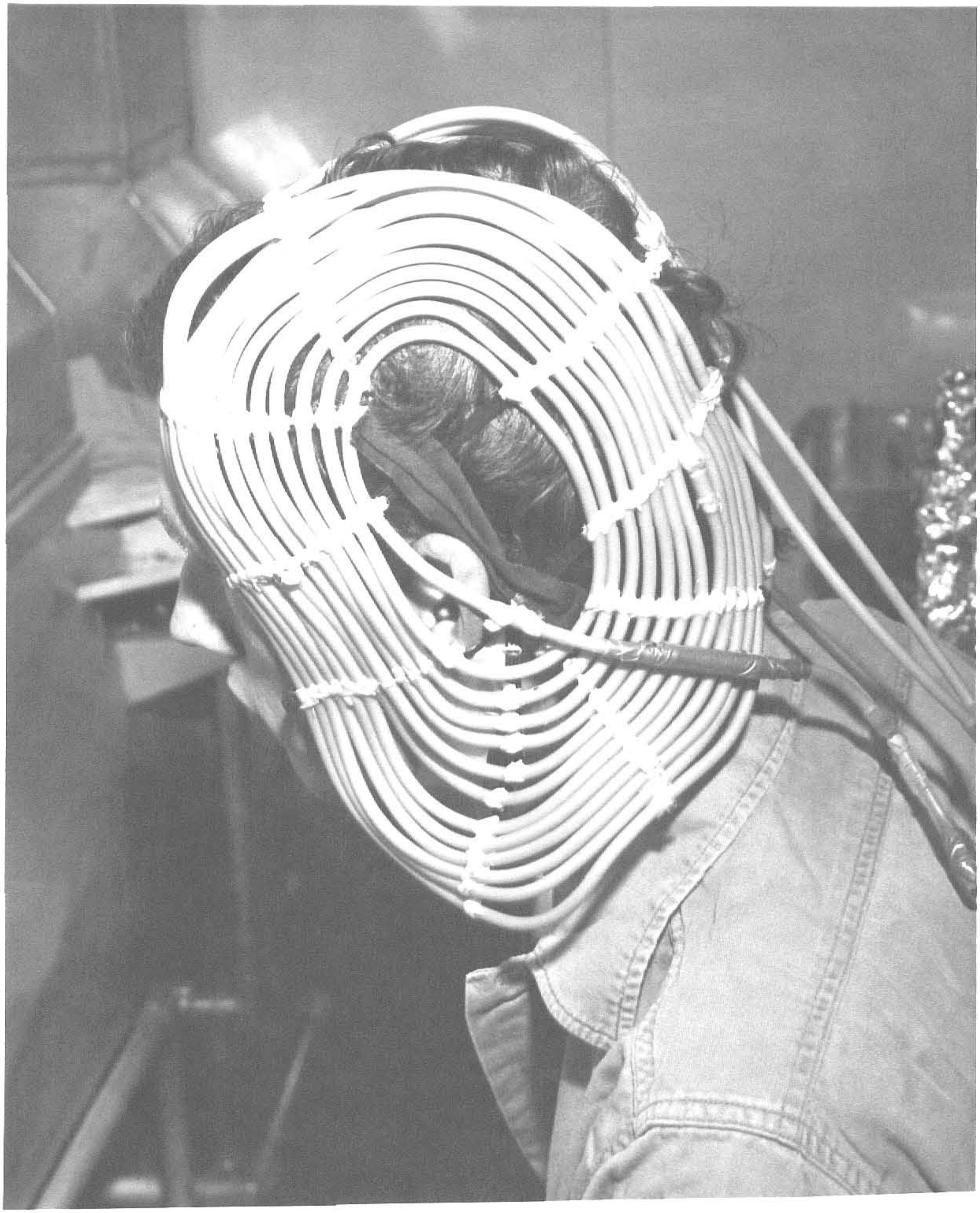


Figure 8. Photograph of hood E showing the foam rubber insulation.


2655
PRE-SOFTENED
Kit
cleaner/wax

Table 3.

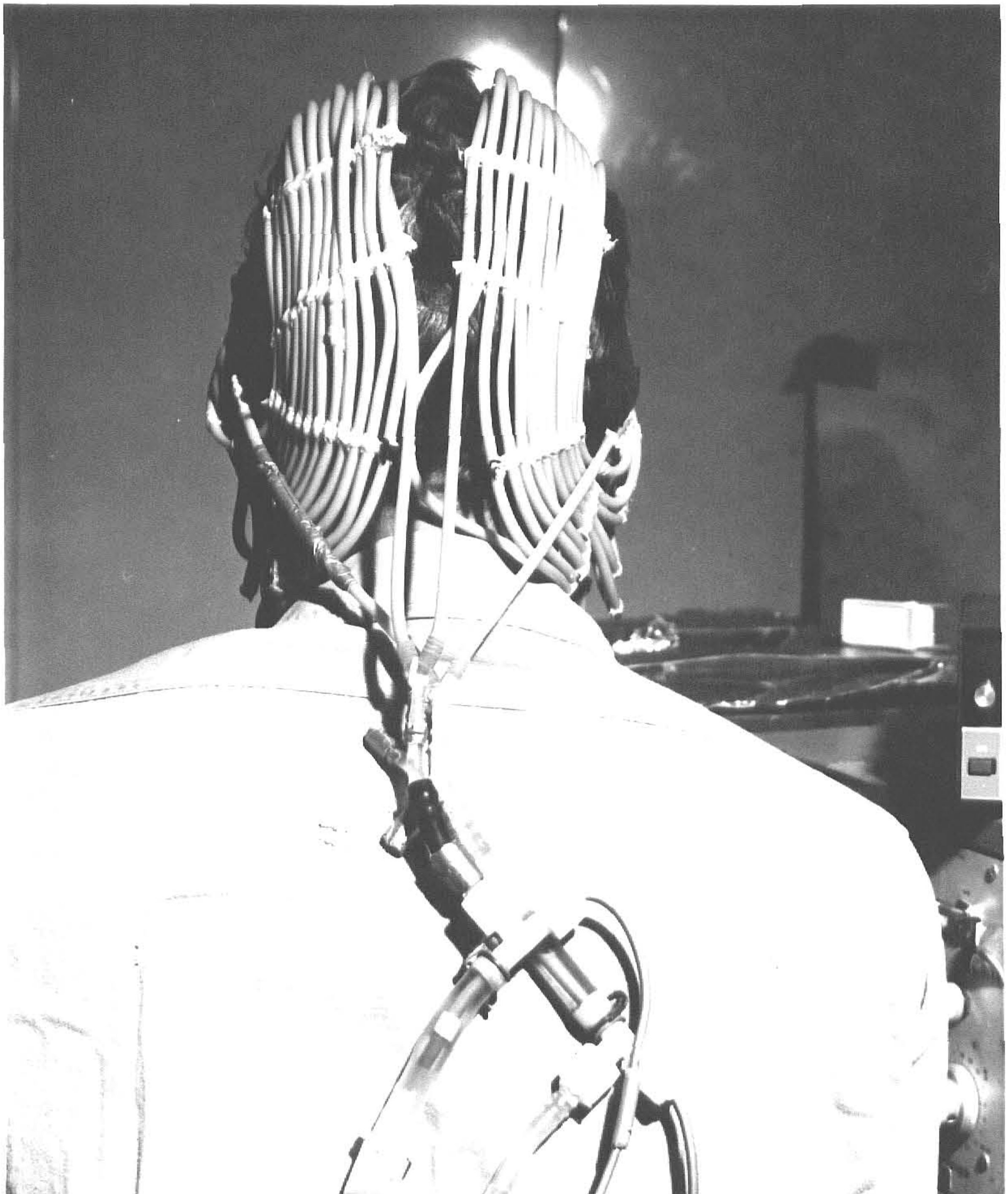
Comparison of KSU hoods.

Variable	A Morales	B Gupta	C Duncan	D Duncan	E Clack
Tubing		Pad type hood			
Length	23'		27'	22'	48'
ID	3/16"		3/16"	3/16"	1/8"
Wall	1/32"	.025"	1/32"	1/16"	1/32"
Material	Polyethelene	Rubber	Excelon	Tygon	Natural rubber
Circuits	2	1	2	2	2
Shell	medium weight canvas duck	Tube molded into shell	Nomex	Nomex	skeleton of 1/4" elastic
Flow Rate, lb/hr.	151	125	260	260	20,40,107
Inlet Temp., F	est. 35 - 40	43	60	60	40,60
Environment:	neutral - 76 F & 50% rh stress 100 F & 70% rh	neutral 76 F & 50% rh stress 100 F & 70% rh	stress 112 F & 60%	stress 112 F & 60%	slight stress 80 F 50% rh and 100 F 50% rh
Subjects	2	8	12	12	6
Task	ergometer	anagrams	addition	addition	step (150 $Kcal/m^2$ hr)

Water Temperature Measurement - Inlet and outlet temperatures were measured with Yellow Springs Instrument Model 409 Thermisters. A Digitec Model 1515 provided a digital readout in degrees Farhrenheit. Accuracy is rated by the factory to within one tenth of a degree. Measurement of the water temperature was made immediately as the water entered and left the hood. The thermister probes were inserted about four inches into the flow stream with the use of plastic T fittings. See Figure 9.

Heat Rate Measurement - Heart rate was measured via E & M Instrument Company telemetery equipment (FM - 1100 - 6 Biotelemetery Receiver, FM - 1100E2 Transmitter, and Self Adhering Surface Electrodes). The EKG was printed out with a Beckman RS-2 strip chart recorder. The incremental heart rate studied was obtained by subtracting the average of the basal values taken before each run from each test reading.

Figure 9. Photograph of thermister probes inserted into the flow stream.



RESULTS

Heat Absorbed

From Subject and Environment: The heat absorbed by the hood, both from the person and from the environment, was calculated from the formula

$$Q = m\ c\ (T_{out} - T_{in}). \qquad (6)$$

On this data, an analysis of variance was performed. See Table 4. The main effects of Environment, Water Inlet Temperature, and Flow Rate significantly ($\alpha < .01$) affected Q.

The average heat absorbed at each condition of each main effect is given in Table 5. Duncan's NMRT was used to determine that each flow rate was significantly ($\alpha < .01$) different from the others.

The heat absorbed, averaged over subjects, at each flow rate is plotted in Figure 10 for each water inlet temperature. The mean heat absorbed, averaged over flow rates for the 40 F and 60 F inlet temperatures, was 264 and 158 Btu/hr, also shown in Figure 10. Figure 11 shows the mean heat absorbed from subject and environment averaged over subjects for each flow rate, given environmental temperature and water inlet temperature.

When water inlet temperature was reduced 20 degrees from 60 to 40 F, total heat absorption increased 80 Btu in the 80 F environment and 50 Btu in the 100 F environment. This was expected because heat absorbed from the environment should be a direct function of the relative difference between the temperature of the environment and the temperature of the water.

Table 4.

Analysis of variance performed on the heat absorption by the hood from both the person and the environment.

Source of Variance	DF	Mean Square	F
Environment	1	245,256.	347.0^{**}
Water Inlet Temperature	1	551,238.	78.0^{**}
Water Flow Rate	2	84,897.	856.0^{**}
Subject	3	1,846.	—
Environment by Water Inlet Temp.	1	4,010.	1.4
Environment by Flow Rate	2	4,974.	1.8
Environment by Subject	3	706.	0.3
Water Inlet Temp. by Flow Rate	2	6,590.	2.3
Water Inlet Temp. by Subject	3	7,028.	2.5
Flow Rate by Subject	6	99.	0.0
Residual	167	2,803.	
Total	191		

$^{**}p < .01$

Table 5.

Mean values of Q from subjects and environments in Btu/hr for environments, water inlet temperatures, flow rates, and subjects.

Environment, F	80 F	100 F
	175	247

Water Inlet Temperature, F	40 F	60 F
	245	158

Flow Rates, lb/hr	20 lb/hr	40 lb/hr	107 lb/hr
	174	214	246

Subjects	RC	SV	GK	AC
	215	216	210	202

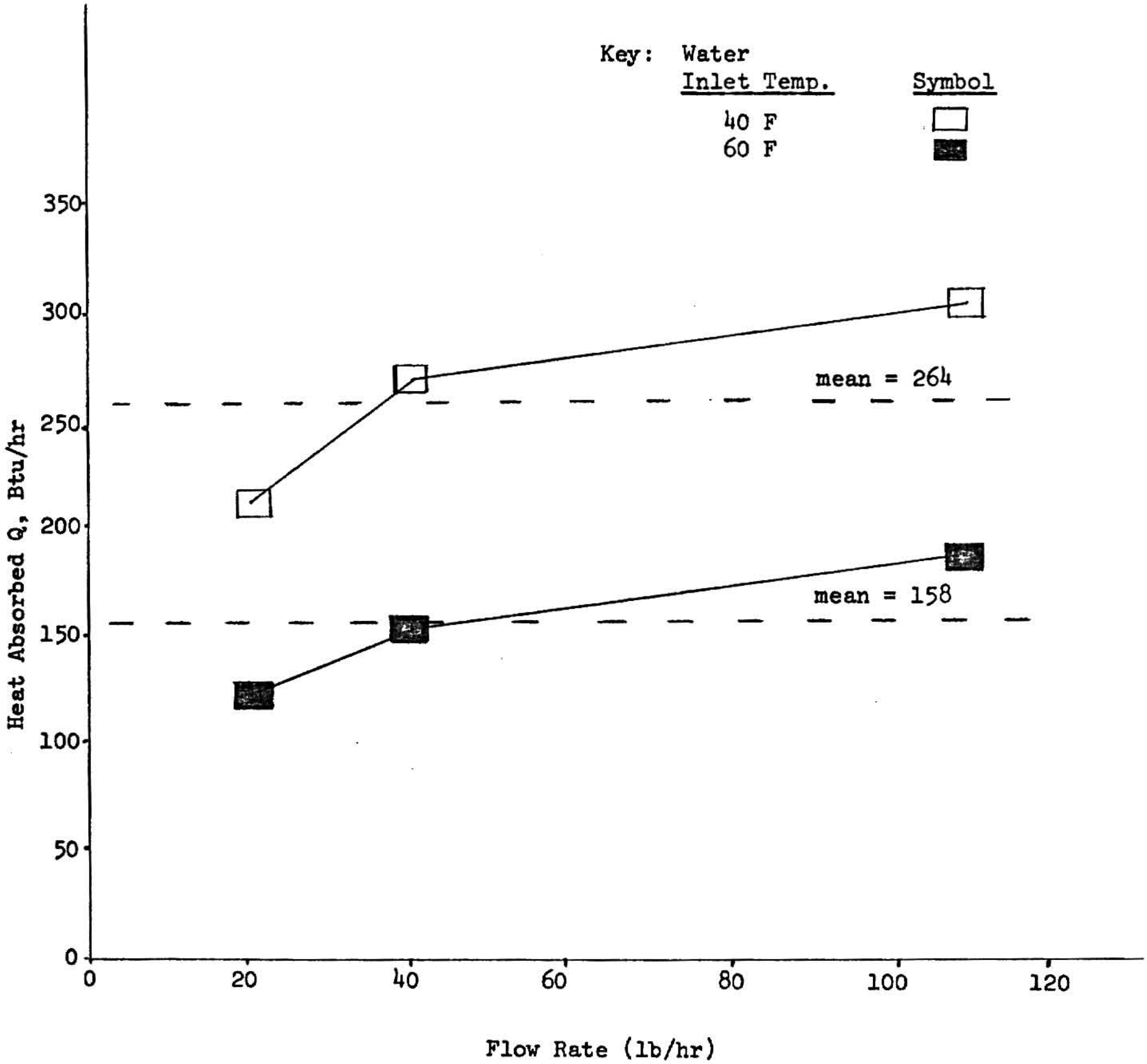


Figure 10. Mean heat absorbed at each flow rate for each inlet temperature condition. Mean absorption for the 40 F water inlet temperature is 150 % of the heat absorption for the 60 F water inlet temperature.

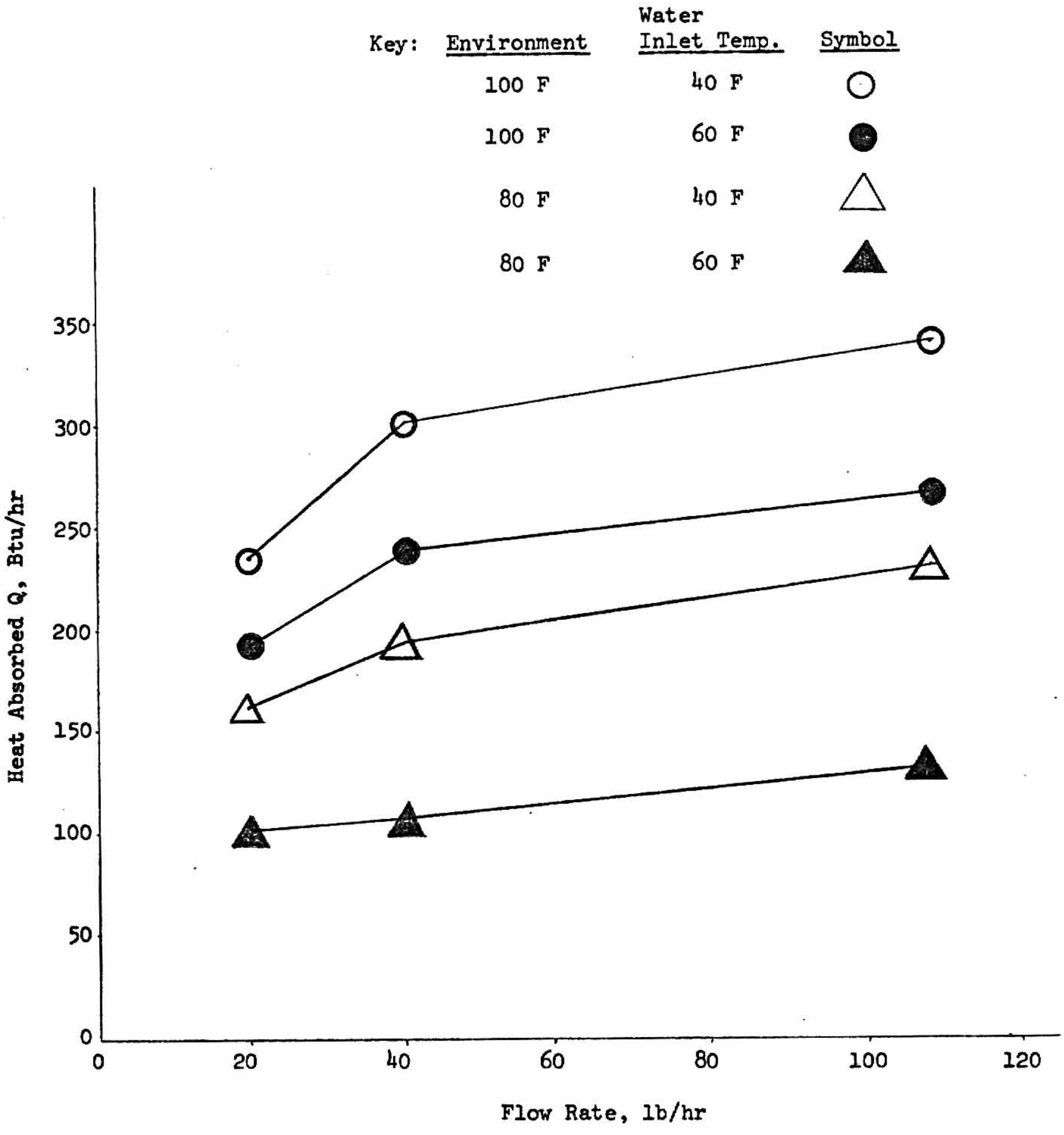


Figure 11. Mean heat absorbed from subject and environment averaged over subjects for each flow rate, given environment temperature and water inlet temperature.

In other words, reducing the water temperature from 60 to 40 F in the 80 F environment represents a change in the temperature difference from 80 - 60 = 20 to 80 - 40 = 40 or an increase of 100 %. Reducing the water temperature from 60 to 40 F in the 100 F environment represents a change from 100 - 60 = 40 to 100 - 40 = 60 or only a 50 % increase. Figure 12 shows a graph of the differences in heat absorbed between the 60 and 40 F water inlet temperatures in both environments and at each flow rate.

From Subject Only: The heat absorbed by the hood from the environment was obtained by testing the hood at each combination of conditions when it was not being worn. For this test the hood was enclosed in the insulation and laid on the step. See Table 6. Note that in the 80 F environment and 60 F water inlet temperature, the Q for the 40 and 107 lb/hr flow rates are inconsistant in that it would be expected that Q for the 40 lb/hr flow rate would be smaller than the Q for 107 lb/hr value. This inconsistancy is likely due to experimental error. To obtain the values of heat absorbed from the subjects only, the values of heat absorbed from the environment were subtracted from the corresponding values of heat absorbed from environment plus subject. Heat from the subjects (with the environmental Q subtracted) was also tested with an analysis of variance; see Table 7. The average heat absorbed from the subject only at each combination of conditions is given in Table 8.

Heart Rate

The incremental heart rate (the number of beats per minute in excess of basal) was also tested with the analysis of variance. See Table 9. There were no significant ($\alpha < .05$) main effects. Significant ($\alpha < .05$)

Table 6.

Heat absorbed, Btu/hr, by the hood from the environment when hood was not being worn by a subject for each environment, water inlet temperature, and flow rate condition.

80 F Environment

	Flow Rate, lb/hr		
Water Inlet Temp.	20	40	107
40 F	99	114	158
60 F	24	35	32

100 F Environment

	Flow Rate, lb/hr		
Water Inlet Temp.	20	40	107
40 F	117	199	238
60 F	49	72	88

Table 7.

Analysis of variance performed on the heat absorption by the hood from the person only.

Source of Variance	DF	Mean Square	F
Environment	1	18807.	22.2*
Water Inlet Temp.	1	1745.	0.2
Flow Rate	2	3768.	23.0*
Subject	3	2933.	—
Environment by Water Inlet Temp.	1	25870.	5.4
Environment by Flow Rate	2	1138.	0.2
Environment by Subject	3	849.	0.2
Water Inlet Temp. by Flow Rate	2	8398.	1.7
Water Inlet Temp. by Subject	3	8604.	1.8
Flow Rate by Subject	6	164.	0.2
Residual	167	4808.	
Total	191		

*$p < .05$

Table 8.

Mean values of Q (Btu/hr) from subjects only for environments, water inlet temperatures, flow rates, and subjects.

Environment, F	80 F	100 F
	100	120

Water Inlet Temperature, F	40 F	60 F
	113	107

Flow Rates, lb/hr	20 lb/hr	40 lb/hr	107 lb/hr
	102	110	118

Subjects	RC	SV	GK	AC
	117	114	108	99

Table 9.

Analysis of variance performed on the incremental heart rate.

Source of Variance	DF	Mean Square	F
Environment	1	2283.	6.5
Water Inlet Temperature	1	0.	0.0
Water Flow Rule	2	106.	3.2
Subject	3	3194.	—
Environment of Water Inlet Temp.	1	158.	1.7
Environment by Flow Rate	2	8.	0.1
Environment by Subject	3	354.	3.7*
Water Inlet Temp. by Flow Rate	2	4.	0.0
Water Inlet Temp. by Subject	3	272.	2.8*
Flow Rate by Subject	6	34.	0.3
Residual	167	95.	
Total	191		

*$p < .05$

two way interactions were ones that involved the subject effect. The average incremental heart rate at each condition of each main effect is given in Table 10. The data was split according to environment so that the main effects were Water Inlet Temperature, Flow Rate, and Subject. Again there were no significant differences in either the 80 F or 100 F environment. See Table 11. Next the data was split according to Water Inlet Temperature so that the main effects were Environment, Flow Rate, and Subject. This time Environment was found to be significant in the 40 F Inlet Water Temperature. See Table 12. Figure 12 shows a graph of the incremental heart rate averaged over the four subjects for each flow rate, given environmental temperature and water inlet temperature.

The lack of significant effect on the incremental heart rate by environment, water inlet temperature, and flow rate was unexpected. It was expected that the heat lost to the hood would cost less physiologically than the heat lost to the environment by normal physiological methods and thus the different heat absorption rates would have correspondingly different heart rates.

Table 10.

Mean incremental heart rate at each main effect condition.

Environment, F	80 F	100 F
	29.5	36.4

Water Inlet Temperature, F	40 F	60 F
	32.9	33.0

Flow Rates, lb/hr	20	40	107
	33.5	33.9	31.5

Subjects	RC	SV	GK	AC
	28.8	32.1	44.7	26.3

Table 11.

Analysis of variance of incremental heart rates taken in the 80 F environment and the 100 F environments.

		Environment			
		80 F		100 F	
Source of Variance	DF	Mean Square	F	Mean Square	F
Water Inlet Temperature	1	68.	0.8	90.	0.2
Flow Rate	2	32.	1.2	82.	2.2
Subject	3	1898.	—	1650.	—
Water Inlet Temp. by Flow Rate	2	12.	0.5	1.	0.0
Water Inlet Temp. by Subject	3	87.	3.5*	571.	11.3*
Flow Rate by Subject	6	28.	1.1	37.	0.7
Residual	78	25.		51.	
Total	95				

*$p < .05$

Table 12.

Analysis of variance of incremental heart rates taken while using a water inlet temperature of 40 F and again while using a water inlet temperature of 60 F.

		Water Inlet Temperature			
		40 F		60 F	
Source of Variance	DF	Mean Square	F	Mean Square	F
Environment	1	620.	35.0**	1820.	2.5
Flow Rate	2	68.	2.4	43.	3.1
Subject	3	1019.	—	2447.	—
Environment by Flow Rate	2	0.	10.0	16.	0.0
Environment by Subject	3	18.	13.0	722.	0.3
Flow Rate by Subject	6	28.	14.0	14.	0.4
Residual	78	65.		34.	
Total	95				

*p < .05

**p < .01

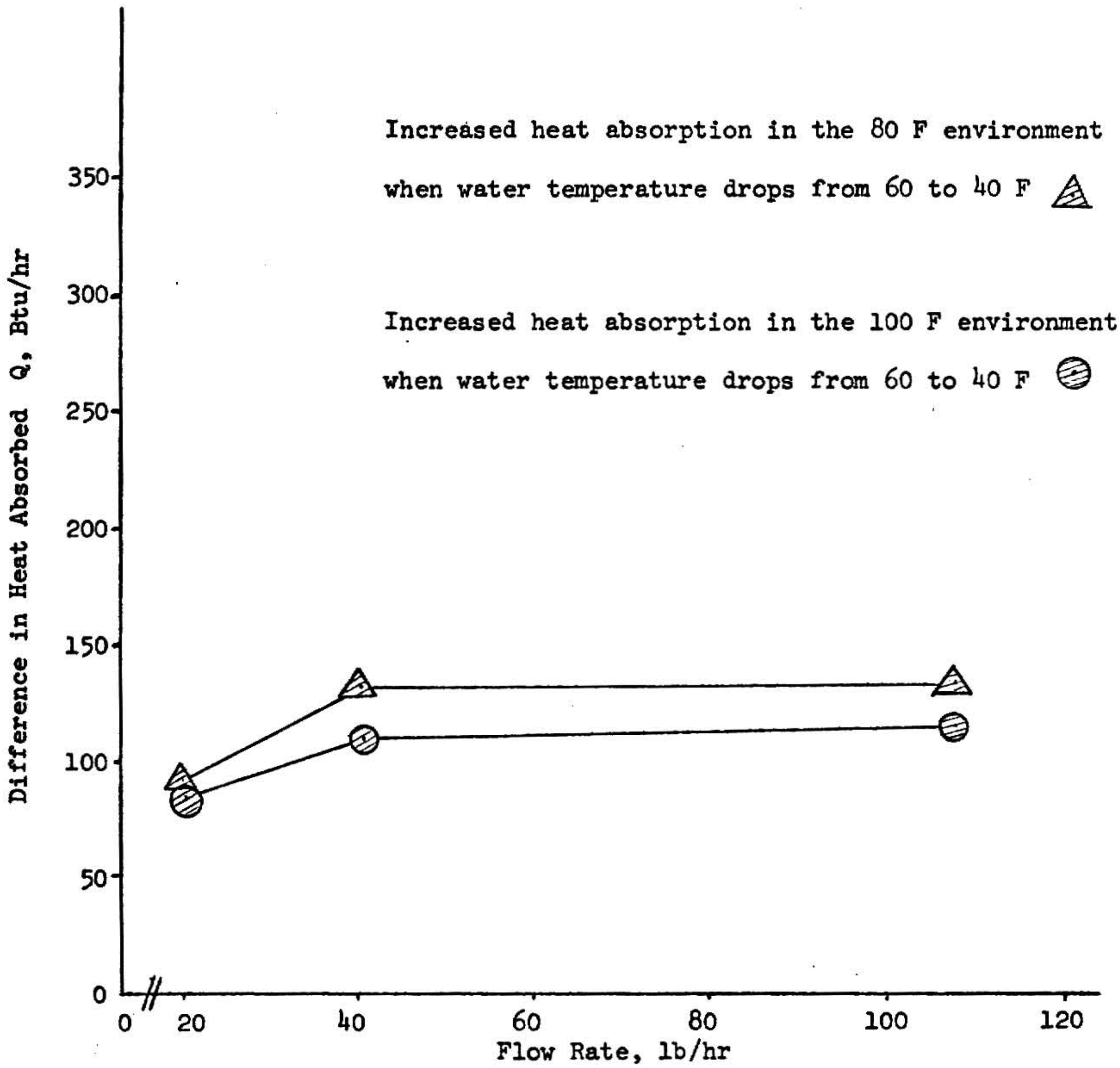


Figure 12. Difference in heat absorbed when inlet water temperature was reduced from 60 to 40 F for the 80 and 100 F environments.

DISCUSSION

Analysis of Suit Performance Equations and Their Application to Cooling Hood E:

The suit performance equations mentioned in the literature review (Burton, Richardson, and Buchberg and Harrah) are observed to be of the form:

$$Q = m\ c\ (1 - e^{\left(-\frac{f(k)}{m\ c}\right)})(T_s - T_{in}) \tag{7}$$

Burton let $f(k) = \pi dLnU$. πdLn is the total inside wetted surface area of the tubing which is 1.57 ft^2 for hood E. U is the overall heat transfer coefficient (Btu/hr - ft^2-F) which is related to the heat transfer coefficient of the tubing (h), the heat transfer coefficient of the skin (h_{skin}), and the heat transfer coefficient of the outer surface of the water next to the tube wall (h_w) in the following way:

$$\frac{1}{U} = \frac{1}{h} + \frac{1}{h_{skin}} + \frac{1}{h_w} \tag{8}$$

Hood E uses natural rubber with a conductivity, $k = 9.2 \times 10^{-2}$ Btu/hr - ft - F. The tubing wall thickness is 2.6×10^{-3} ft so the tube heat transfer coefficient (h) = $(9.2 \times 10^{-2})/(2.6 \times 10^{-3})$ = 35.4 Btu/hr - ft^2 - F. Richardson estimates the heat transfer coefficient of the skin (h_{skin}) to be large enough that the $1/h_{skin}$ term can be disregarded in equation (8), and the heat transfer coefficient of the water next to the tube wall (h_w) to be about 250 Btu/hr - ft^2 - F. Thus when equation 8 is solved, U = 31.0 Btu/hr - ft^2 - F. For hood E, f(k) then is equal to 48.7 Btu/hr - F.

Richardson let $f(k) = \psi\pi DLnU$. For hood E, $\psi\pi Ln$, representing the contact area between the tubing and the skin, equals 0.63 ft^2 if ψ (the contact surface factor) is estimated to be 27%. Allowing U = 31.0 Btu/hr - ft^2 - F as explained above, f(k) would equal 19.6 Btu/hr - F.

Buchberg and Harrah let f(k) = skLn. The shape factor $s = 6.3\psi + 1.1 = 2.8$ for a contact surface factor ψ of 27%. The conductivity of the tubing $k = 9.2 \times 10^{-2}$ Btu/hr - ft - F. f(k) then equals 12.4 Btu/hr - F.

These theoretical values of f(k) contrast with the average experimental values of 2.7 Btu/hr - F obtained in the 80 F environment and 3.5 Btu/hr - F obtained in the 100 F environment. In comparing hood E with hood C it is observed that their difference in tubing length was the single most important difference with regard to the theoretical calculation of f(k). For example, for hood C, using Buchberg and Harrah's model (f(k) = skL), s would be the same, 2.8, k is approximately 7.2×10^{-2} Btu/hr - ft - F, and Ln = 27', so f(k) = 5.4 Btu/hr - F as compared to 12.4 for hood E. So, while the theoretical value of f(k) for hood E is almost twice as large as f(k) for hood D, the experimental values (2.7, and 3.5 Btu/hr - F) for hood C are smaller than the experimental value for hood D. Hence there is some doubt about the applicability of the suit performance equation to hoods.

The values of heat absorbed from subjects only (mean values shown in Table 8) were inserted into equation (7) to solve for f(k). When these experimental values of f(k) were tested in an ANOVA, the effects of environment and flow rate were significant ($\alpha < .05$). Yet the equation was supposed to eliminate the effect of flow rate. It does not account for environment at all. Thus the suit performance equation in its present form cannot be successfully applied to cooling hoods.

It is noted that the theoretical value of f(k) becomes more pessimistic with each new modification. The difference between the value of f(k) obtained from the latest modification, by Buchberg and Harrah, and the experimental values obtained from the hood are likely due to the basic differences between a cooling garment on the torso and the head. A suit covers the torso and limbs and the assumption was made that the man and the suit were perfectly insulated from the environment as they would be in an extravehicular space application. Therefore the amount of heat picked up by the suit (Q) is also a close representation of the metabolic heat. In the test of hood E, on the other hand, the environment has an important effect on Q. The hood can absorb or lose heat in direct exchange with the environment through the imperfect insulation or indirectly transmitted through the subject. Also metabolic heat generated by the task could be lost directly to the environment through normal physiological mechanisims instead of being absorbed by the hood. There is evidence that heat absorption is a function of heat storage within the body. Nunneley noted that heat absorption from the head appeared to be related to metabolic rate. In Nunneley's work 120 Btu/hr was the maximum heat absorption rate attainable with the subject at rest while up to 360 Btu/hr was obtained while the subjects were working.

Another assumption made by the suit performance equations that doesn't apply to hood E concerns the tube spacing. Burton doesn't make a specific tube spacing assumption but does point out that as spacing decreases the efficiency is decreased. Richardson's analysis assumes a tube spacing of one inch and Buchberg and Harrah's analysis assumes a spacing of 0.74

inches. In terms of tube diameter, assuming 3/16" OD tubing, Richardson's analysis assumes a tube spacing of 5.3 diameters and Buchberg and Harrah's analysis assumes a spacing of 4.0 diameters. Hood E has a tube spacing of approximately one fourth of an inch or 1.3 diameters which would very likely cause efficiency to be reduced. Schnurr and Rogers (1970), in a study of the effect of heat transfer for snow removal, found the heat flux, ϕ, was

$$\phi = 1 + .00365\ (\ell/D)^{2.38}$$

where ϕ = heat flux required, Btu/hr - ft^2

ℓ = center to center spacing of tubes

D = outside diameter of the tubing

The critical number to note is the .00365. This indicates that changing the ratio of ℓ/D from 5 to 1 would decrease the flux required by only 10%. In other words, closer spacing buys relatively little. This was for heat transfer in concrete not flesh but it tends to indicate that close spacing is not especially beneficial.

Other ways in which suits would differ from a hood are: 1) Insulation that hair provides. This may significantly reduce the value of the heat transfer coefficient (U) which would in turn lower the values of f(k) and 2) Fit of the hood. The head has a complex shape and it is difficult to obtain a good fit all over such as behind the ears or in convex areas such as the temples.

Hood Performance Equation

A heat exchange equation for a conduction cooling hood that uses tubes would have to consider flow rate, coolent temperature environment, body heat storage, and hood design. An equation of this sort, considering the heat absorbed directly from the environment Q_1 and from the body Q_2 separately might be of the form:

$$Q = m\ c\ (T_s - T_{in})(1 - e^{\left(-\frac{WkLn}{m\ c}\right)})$$

$$+ m\ c\ (T_{env} - T_{in})(1 - e^{\left(\frac{F(i,v)}{m\ c}\right)})$$

where:

Q = heat absorbed by the cooling medium

m = mass flow rate of the cooling medium

c = specific heat of the cooling medium

T_s = skin temperature

T_{in} = water inlet temperature

$T_{env.}$ = environment temperature

s = conductive shape factor of the tubing described by Buchberg and Harrah

k = conductance of the tubing material

L = length of a tubing circuit

ℓ = distance between tubing circuits, center to center.

n = number of circuits

D = OD of tubing

i = hood insulation

W = ℓ/ZD

v = air velocity of the environment

Z = most efficient tube spacing in terms of tube diameters.

The term $W = \ell/ZD$ comes from Schnurr and Rogers and represents the efficiency of heat removal with regard to tube spacing. In the second part of the equation (i) in F(i,v) represents the insulation factor and (v) represents the air velocity.

At this stage of development of a hood performance equation it should be recognized that it cannot yet be used to reliably predict hood performance. The temperature of the skin (T_s) is not yet well enough defined. Is the heat being transfered to the cooling fluid, which begins at temperature T_{in}, from the skin immediately beneath the tubing at temperature, for example (T_{in} + 5F), or from an average skin temperature (T_{in} + 30F)? Or perhaps the heat transfer should be considered to be the cooling fluid at T_{in} from the body core temperature at $\simeq$ 98.6 F. Further it should be recognized that T_s itself is some function of metabolic rate, heat storage, psychological state, and the environment.

It is emphasized that again this equation is emperical and untested. "The proof is left to the student."

CONCLUSIONS

1. Flow rates in excess of 40 lb/hr for hood E are subject to rapidly diminishing returns because heat absorption has approached an asmptotic maximum. This relationship is represented mathematically by the suit performance equation

$$Q = m\ c\ (T_s - T_{in})(1 - e^{-\left(\frac{f(k)}{m\ c}\right)})$$

where $f(k) \simeq 3$ for hood E.

2. Hood E did not absorb as much heat as was expected which may imply that the head does vasoconstrict or otherwise regulate the amount of heat it gives up. Hoods C and D tested by Duncan absorbed around 250 Btu/hr from subjects sitting in an environment at which radiation and convection added to the stress and evaporation was difficult. In the present study, hood E absorbed around 130 Btu/hr from working subjects in a mildly stressful environment.

3. Heart rate was not affected by environment, flow rate, or water inlet temperature. This seems due to the fact that heat which was not absorbed by the hood could be lost through normal physiological means such as evaporation.

4. In order to apply suit performance equations to hoods, metabolic rate, heat storage, environment, and tube spacing must be considered.

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THE EFFECTIVENESS OF A CONDUCTIVE COOLING

HOOD AT VARIOUS FLOW RATES AND INLET TEMPERATURES

by

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B.S., Kansas State University, 1968

AN ABSTRACT OF A MASTER'S THESIS

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requirements for the degree

MASTER OF SCIENCE

Department of Industrial Engineering

KANSAS STATE UNIVERSITY

Manhattan, Kansas

1970

ABSTRACT

A water cooled cooling hood was tested at three flow rates (40, 80, and 107 lb/hr) and two inlet temperatures (40 and 60 F) in two environments (80 and 100 F dry bulb with 50% RH). Four student subjects stepped at a metabolic rate of approximately 1000 Btu/hr. The physiological index measured was heart rate. The heat absorbed by the hood was obtained by measuring the flow rate and the inlet and outlet temperatures.

The hood tested had 48' of 1/8" ID, 1/32" wall, natural rubber tubing. A previous one built and tested by Duncan (1969) at Kansas State University had 27' of 3/16" ID, 1/32" wall, Excelon tubing. Duncan's subjects were tested sitting in a 112 F, 58% RH environment. His flow rate was 260 lb/hr and inlet temperature was 60 F. Estimated heat absorbed by his hood from the subject was 245 Btu/hr.

Results of the present experiment showed that environment, water inlet temperature, and flow rate significantly ($\alpha < .05$) affected heat absorption. The average heat absorbed for an inlet temperature of 60 F, flow rate of 107 lb/hr, and environment of 80 F was 140 Btu/hr, with 110 Btu/hr from the subject and 30 Btu/hr from the environment. At the 100 F environment, other conditions the same, the average heat absorbed was 235 Btu/hr, with 150 from the subject and 85 from the environment.

Water inlet temperature and flow rate did not significantly ($\alpha < .05$) affect heart rate. The effect of environment on heart rate was significant in the 40 F water inlet temperature condition.

An equation predicting the effect of inlet temperature, flow and environment on heat absorption is given.