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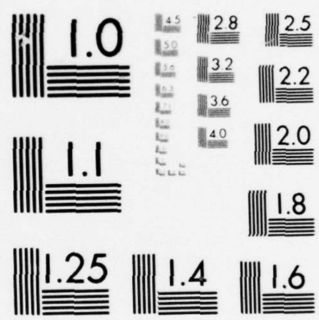

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THE USE OF THE PHYSIOLOGICAL THERMAL INDEX (PTI) AS AN ALTERNATE TO EFFECTIVE TEMPERATURE FOR CIVIL DEFENSE PLANNING

FINAL REPORT—PART B
DEFENSE CIVIL PREPAREDNESS AGENCY
DEPARTMENT OF DEFENSE

CONTRACT NO. DAHC 20-69-C-0128
WORK UNIT 1224D

by
Richard K. Pefley
James T. Macdonald
November 1977

The University of Santa Clara • California

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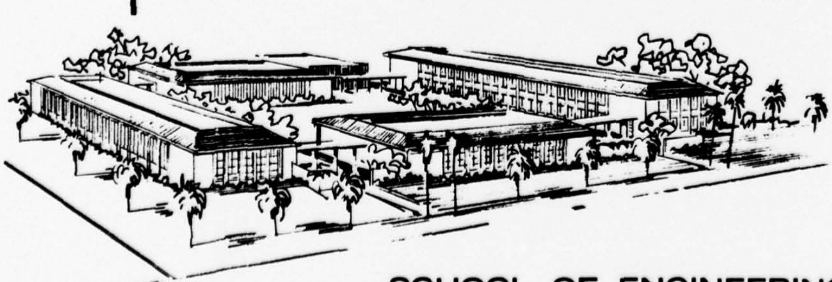

SCHOOL OF ENGINEERING

ENGINEERING AND APPLIED SCIENCE RESEARCH

UNIVERSITY OF SANTA CLARA

SANTA CLARA, CALIFORNIA AREA CODE 408-984-4467

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SUMMARY

Ventilation planning for survival shelters had been based on the effective temperature (ET) concept established in the literature of the American Society of Heating, Refrigeration and Air Conditioning Engineering (ASHRAE). The ET concept has been repeatedly subjected to criticism by several research workers and alternatives have been proposed. One, the new effective temperature (ET*) has replaced the ET concept in the current ASHRAE literature. Another, the Physiological Thermal Index (PTI) has been developed by University of Santa Clara researchers under DCPA sponsorship.

A comparison is presented here among three human thermal models that relate to these new indicies of comfort. The comparisons are in the context of suitability for predicting ventilation requirements for survival shelters. Some inadequacies are found in all three models. The principal conclusion is that the ET* index of comfort found in current ASHRAE literature predicts warm hotcomfort conditions at combinations of temperature and humidity that are greater than those recommended by the PTI and ET scales. Hence, caution is advised and more study recommended before the ET scale in the DCPA literature is supplanted by the ET*. Adoption of the PTI scale appears to be a more satisfactory alternative.

ME 77-3

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INTRODUCTION

The thermo-regulatory system of the human being involves a very complex and sophisticated control network, capable of responding to a wide variety of environmental conditions while residing within the limits of human thermal tolerability. In order to maintain a deep body temperature that is within an acceptable limit, the human body must continually reject a prescribed quantity of heat to the environment that is in keeping with a given activity level. Sedentary equilibrium conditions which are characteristic of shelterees will be studied presently, and for this condition the rate of heat rejection must exactly balance the human's sedentary metabolism. To this end the individual may respond to the environmental conditions through a combination of conductive, convective, evaporative, and radiative heat exchange mechanisms.

In a variety of disciplines it is often necessary to understand and predict typical human thermal responses to specific environmental conditions. A great deal of work has been done in this area by physiologists and engineers in an attempt to develop a sound understanding of man's thermal response characteristics. The Monoman Calorimeter at the University of Santa Clara which has been supported throughout its life by DCPA has contributed significant data in this field for the past twelve years. The contributions have been primarily associated with the conditions of high temperatures and humidities and low ventilation rates associated with survival shelter planning.

The original index of human comfort in the air conditioning literature is named Effective Temperature (ET). This index is extensively used in DCPA literature. However, the authors of the ET index of comfort and other researchers have pointed out its inadequacies. The American Society of Heating Refrigeration and Air Conditioning Engineers (ASHRAE) have generated a new index of comfort (ET*) which is a variation of the original effective temperature. This index is displacing the former ET index of comfort in all of their literature. The Monoman Calorimeter Project has generated the Physiological Thermal Index (PTI) as an alternate to the ET concept. These and other recent indicies of comfort have been supported by thermoregulatory models of the human coupled with his surroundings.

It is the purpose of this study to compare and contrast three of the most prominent models in this field of study starting from the most fundamental levels. The three models chosen are the Physiological Thermal Index (PTI) developed by the University of Santa Clara, the New Effective Temperature Scale (ET*) developed under a research project of the American Society of Heating, Refrigeration, and Air Conditioning Engineers, and the predicative model developed by P. O. Fanger of the Technical University of Denmark.

The ultimate objective of this study is to provide information that may eventually lead to the development of a more universally accepted model which can be incorporated in future revisions of DCPA literature. By exposing the more controversial issues of this field, the developers of these models will be provided with a basis for working toward a resolution to these discrepancies. The first section will be devoted to analyzing the Physiological Thermal Index as well as incorporating effects into this model that have not been considered previously. The remaining sections will discuss the two additional models.

SECTION I: PHYSIOLOGICAL THERMAL INDEX

The basic thermal model of the human thermo-regulatory system used with the Physiological Thermal Index (PTI) has been previously developed. Additions to the model that are included here are the incorporation of the effects of air velocity, clothing, and an assumed mean radiant temperature equal to the ambient temperature. The last of these additions necessarily restricts the application of the model's results to those conditions that would be approximated indoors. In the following sections each of these terms as well as those previously derived will be discussed and analyzed.

The model is a three node (formerly two node without clothing effects), lumped parameter type, using the core, skin, and clothing temperatures as the nodal points of concern. As shown in Fig. 1, the model considers the human body to be composed of a central core and a peripheral skin layer. Heat which is generated deep within the core must be continuously removed in order to maintain thermal equilibrium. A portion of this heat is transferred directly from the core to the ambient air via the respiratory tract, while the remainder is dissipated from the body surface. Heat flows from the core region to the skin surface through a complex combination of arterial blood flow and thermal conduction through the body tissues. Assuming respiratory losses negligible, the model considers all of the rejected body heat to flow through a variable thermal resistance to the skin. Subsequent exchange mechanisms

THREE NODE MODEL

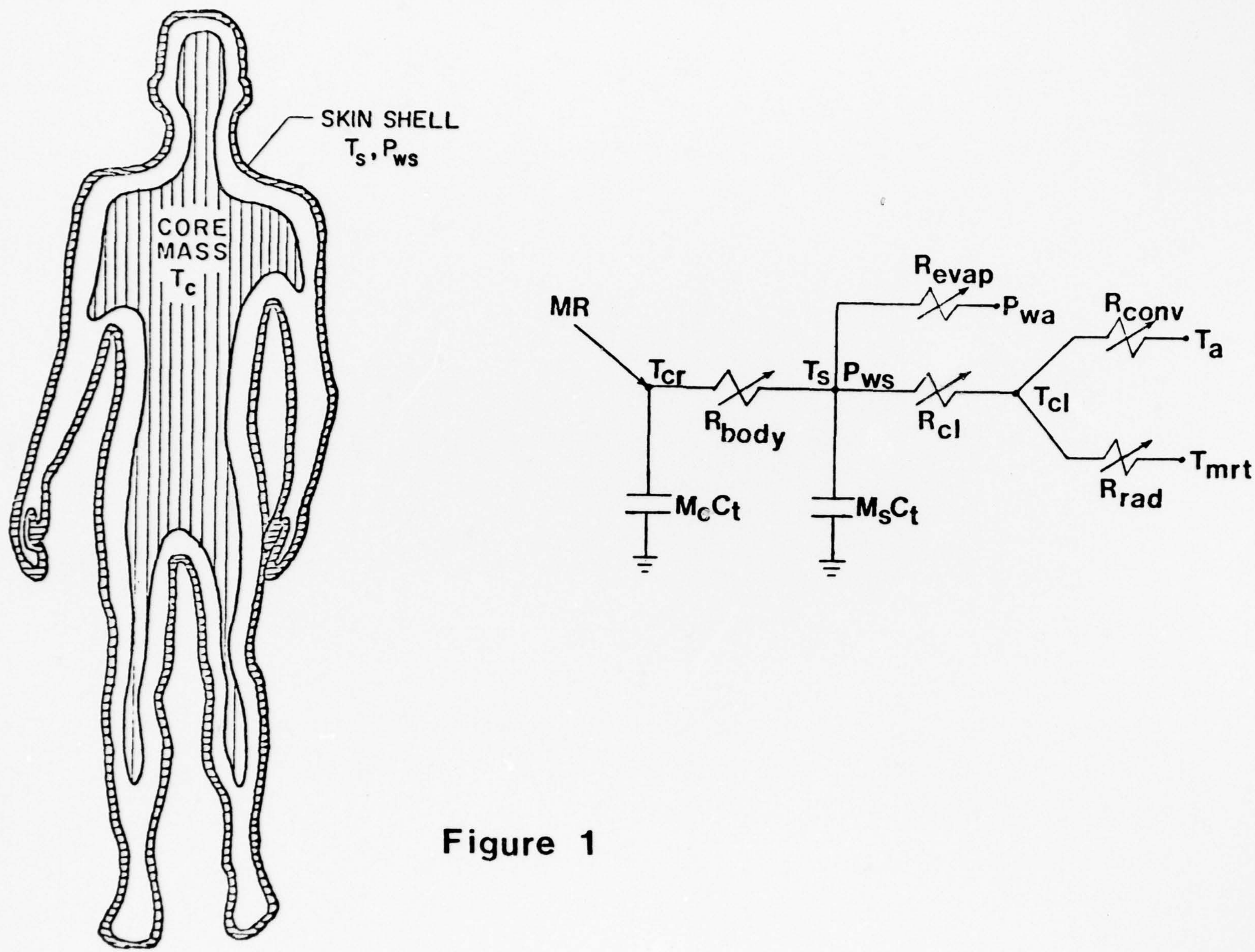

Figure 1

include evaporation, conduction through clothing, radiation and convection. Thermal conduction downstream of the clothing node, T_{cl}, is not considered here since it is generally a localized effect with contacting bodies and does not typically account for a significant portion of a person's metabolic heat rejection. It will, therefore, not be considered further as an available heat transfer mode.

The sections that follow will describe each of the components of the thermal circuit shown in Fig. 1.

I. 1 Internal Heat Production

The metabolic rate, MR, is generated from the release rate of energy by the oxidation processes in the human body. This quantity is sometimes partly converted to external mechanical power, but is mainly converted to internal body heat. Since only the sedentary activity level will be considered here, this power term will be equal to zero and, therefore, all of the internal energy produced will be converted to heat.

I. 2 Internal Heat Flow

Considering only equilibrium conditions and neglecting the cyclic variations in body water content, there will be no change in the internal energy storage of either the core or skin shell. We can thereby omit the core and skin shell storage terms, M_cC_t and M_sC_t, respectively. With these restrictions the variable thermal body resistance, R_{body}, is described by the application of Ohm's Law to the thermal circuit:

$$R_{body} = \frac{T_{cr} - T_s}{MR} \tag{1}$$

where MR is the human's sedentary metabolic rate (Btu/hr)

R_{body} is the thermal resistance of the body to the flow of internal heat (hr - °F/Btu)

T_{cr} is the average core temperature (F)

T_s is the body area averaged skin temperature (F)

Normalizing the sedentary metabolic rate with respect to the DuBois surface area generates the following relationship:

$$R'' = \frac{T_{cr} - T_s}{MR / A_{du}} \tag{2}$$

where A_{du} is the DuBois surface area (the surface area of the nude body) (ft^2)

R' is the unit area thermal body resistance (hr- °F -ft^2/Btu)

Having performed a large number of experimental tests, the functional relationship between R" and T_s was developed. The solid line of Fig. 2 was determined from a linear Gaussian least squares curve fit and for males was found to be:

$$R'' = 4.71-0.0476\ T_s \qquad (3)$$

An alternative method for determining the thermal body resistance, which will be employed here for comparative purposes, can be gained from the following relationship:

$$T_c = 2/3\ T_r + 1/3\ T_s \qquad (4)$$

where T_r is the measured rectal temperature (°F). This equation is assumed to be valid under equilibrium conditions.[1] It has been shown, over the range of skin temperatures considered here, that the deep body temperature, T_r is constant.[2] This temperature was found to be approximately 99.6 F. Combining equations (2) and (4) generates the following thermal resistance expression in the form of Eq. 3:

$$R'' = 3.32 - .033\ T_s \qquad (5)$$

where MR of Eq. 2 is equal to 400 Btu/hr for a male subject at sedentary conditions and A_{du} is equal to 20 ft^2 for a male subject.[3]

This equation has been plotted in Fig. 2 as a broken line. It is apparent that considerable discrepancy exists between the two expressions of the thermal resistance. The core temperature relation cited in Eq. 4 is suspected of giving inaccurate results,[1] but Eq. 3 of the PTI model may also be a possible error source. Introducing the expression for R" given by Eq. 3 into Eq. 2, the core temperature, T_{cr}, can be found as a function of the skin temperature, T_s. Performing this substitution and simplifying generates the following relationship:

$$T_{cr} = 95.1 + 0.039T_s \qquad (6)$$

If this expression is evaluated at neutral thermal comfort conditions, a state point where researchers universally agree on the values of T_c and T_s, insight to the validity of this expression can be gained. It is agreed that at comfort conditions T_s equals 93.4 F and T_c equals 97.9 F.[1,4,5,9] Evaluating this expression for T_s equal to 93.4 F predicts a core temperature

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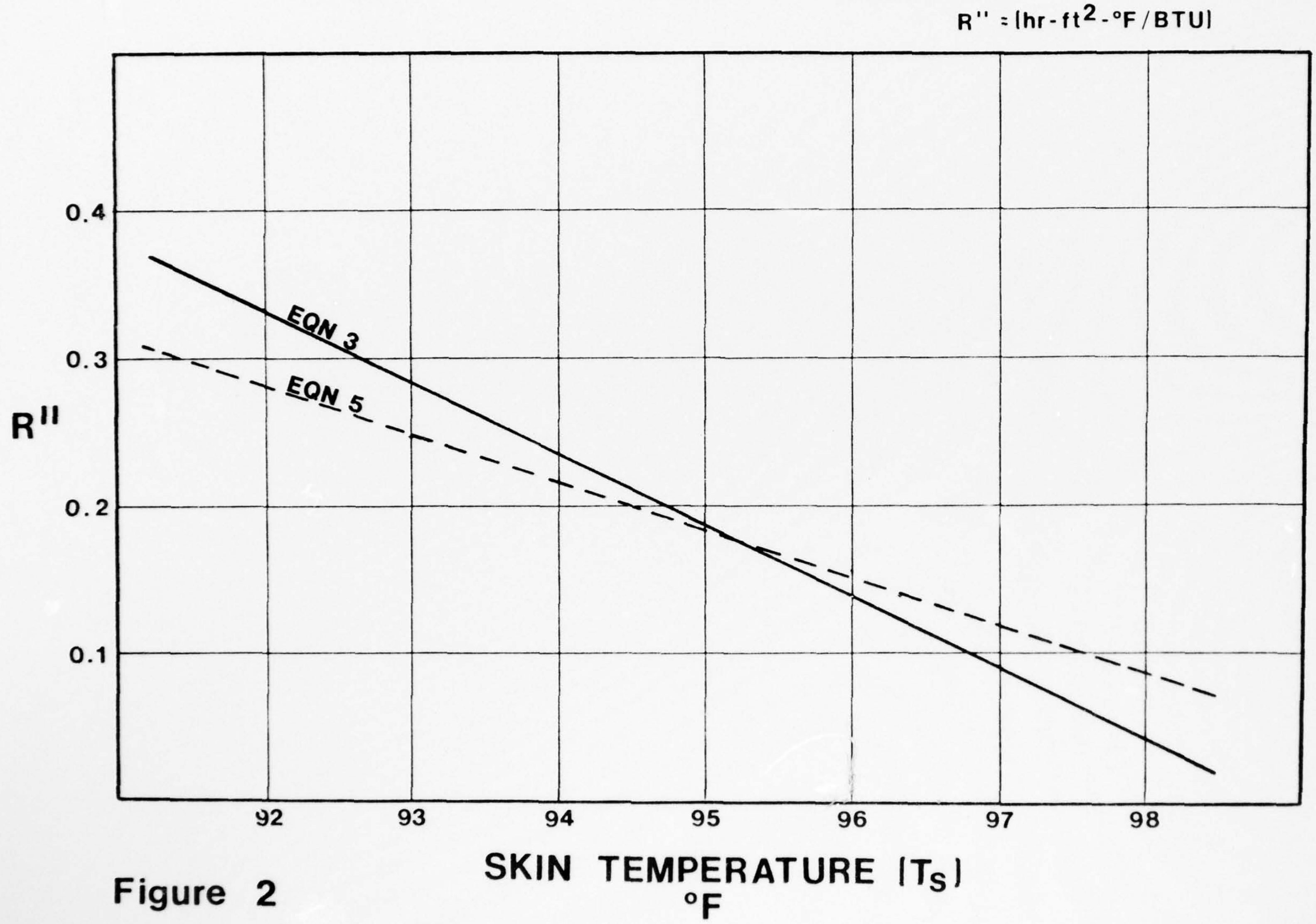

Figure 2

of 98.7 F. For the same value of T_s, Eq. 4 predicts a core temperature of 97.5 F. It is evident that Eq. 4 generates a closer approximation to the agreed value. From this analysis it appears that additional experimental work in the PTI model is needed in this area. More will be said about this particular topic in Section III, dealing with the Physiological Thermal Index and the Effective Temperature Scale analyses.

I. 3 External Heat Rejection

The energy transferred to the skin surface that is not stored as an increase in the internal energy content of the skin shell must be rejected to the surrounding environment.

The heat balance equation describing the heat exchange between the human body and its surroundings is based on the law of conservation of energy. Alternatively, applying Kirchoff's Current law to the thermal circuit of Fig. 1 yields the same result:

$$MR - W - E - S = K = (R + C) \quad (Btu/hr) \qquad (7)$$

where MR is the net rate of metabolic heat production

- W is the rate of mechanical work accomplished
- E is the rate of evaporative heat loss
- S is the rate of gain or loss in the heat content of the body (indicated by the capacitive terms, M_cC_t and M_sC_t)
- K is the heat transfer from the skin to the outer surface of the clothed body (conduction through the clothing)
- R is the rate of thermal energy exchange by radiation from the outer surface of the clothed body
- C is the rate of thermal energy exchange by convection from the outer surface of the clothed body.

The dual heat balance statements given by Eq. 7 indicate that the metabolically produced heat (MR) minus the rate of mechanical work done (W) the evaporative losses (E), and the rate of heat storage (S) is equal to the heat conducted through the clothing (K) and the energy dissipated at the outer surface of the clothing by radiation and convection (R + C). The evaporative loss term, E, combines the effects of moisture diffusion through the dry skin area, evaporation of perspired water from the wetted skin surface, and the latent respiration heat loss. In generating the previous equations relating to body temperatures, it has been assumed that the dry respiratory heat loss is negligible. For all practical purposes this term induces negligible significance in the results of the model (less than 2%). Some of the other researchers, however, have concerned themselves with it. In the light of the major deviations that are present in the models dealing with this subject, many of which will be exposed later in this paper, it seems senseless to concern ourselves with what are at this point trivialities by comparison.

The power term, W of Eq. 7 is taken as zero since only the sedentary activity level is being considered. This term was included in Eq. 7 since ongoing research at the University of Santa Clara will be concerned with the magnitude of this parameter in the analysis of different activity levels. In addition, the present discussion will be limited to conditions of equilibrium and, therefore, the storage term, S, is also taken as zero.

The following sections will discuss and quantify each of the remaining terms in Eq. 7.

I. 4 Heat Conduction Through Clothing: The incorporation of clothing effects in the Physiological Thermal Index model will be implemented here, as the original model concerned itself only with nude subjects. This is requisite to make comparisons to the New Effective Temperature Scale since it has only concerned itself with clothed subjects. In studying the various models that are reviewed and analyzed here it is found that there are two different ways of handling the addition of clothing to the skin surface. The method employed by Fanger will be utilized here for reasons that will be discussed later in this section.

The transfer of dry heat between the skin and the outer surface of the clothed body is quite complicated involving internal convection and radiation processes in intervening air spaces, and the conduction through the cloth itself. To simplify calculations, Gagge et al. introduced the term I_{cl} to represent the total thermal resistance from the skin to the outer surface of the clothed body. This term is defined by:

$$I_{cl} = \frac{R_{cl}}{0.881} \tag{8}$$

where the thermal resistance, R_{cl}, is the total heat transfer resistance from the skin to the outer surface of the clothed body (-hr-°F ft^2/Btu). The resistive term, I_{cl}, has been determined for different clothing ensembles[7] in a rather unique way using a life size heated manikin dressed in an actual clothing ensemble.

Having expressed the resistance term in this manner and applying Ohm's law to the thermal circuit, the dry heat transfer from the skin to the outer surface of the clothed body can be described by the following relation:

$$K = A_{du} \frac{T_s - T_{cl}}{0.881\ I_{cl}} \text{(Btu/hr)} \tag{9}$$

where T_{cl} (°F) is the temperature of the outer surface of the clothing and the other parameters are as defined previously.

The introduction of the clothing conduction equation in the PTI model required that the third node, T_{cl}, be incorporated into the thermal circuit as shown in Fig. 1. The variable resistance between the T_s and the T_{cl} nodes is indicative of different clothing ensembles which correspondingly have different clo (I_{cl}) values. For nude subjects, as in the original model, this resistance goes to zero and $T_s = T_{cl}$.

The alternative method for handling the clothing conduction, which has been utilized in the ET* scale, implements an efficiency factor to account for clothing effects. Known as Burton's efficiency factor, the coefficient eliminates the need for the T_{cl} node in the thermal circuit. This is achieved by incorporating the effects of clothing between the skin temperature node and the ambient temperature and, therefore, a clothing temperature is never considered. The following equation indicates the use of the term:

$$R + C = f_{cl_1}\ h\ A_{du}\ (T_s - T_a)\ \text{(Btu/hr)} \tag{10}$$

where h is the combined heat transfer coefficient consisting of $h_c + h_r$ and the clothing thermal efficiency factor, f_{cl_1}, is given by:

$$f_{cl_1} = \frac{1}{1 + 0.881\ h\ I_{cl}} \quad \text{(N.D.)} \tag{11}$$

where h has the units Btu/hr - ft^2 - °F and I_{cl} is in clo units.

Introduction of Fanger's clothing conduction concept alters Eq. 10 to:

$$R + C = hA_{du}\ (T_{cl} - T_a)\ f_{cl_2}\ \text{(Btu/hr)} \tag{12}$$

where the f_{cl_2} term, not to be confused with Burton's efficiency factor,[7] is the ratio of the surface area of the clothed body to the nude body. This parameter has also been determined for different clothing ensembles and is used to adjust the nude area of the body, A_{du}, by taking into account the increase in the effective outer surface area when clothing is worn. This approach was preferred over the New Effective Temperature Scale technique since it includes the variables of the intermediate processes which, when used in the computer program of the Appendix, can be printed out and examined for validity. Another, more important reason, as will be seen, is that the New Effective Temperature Scale fails to take into account the area ratio term of the Fanger model.

By combining equations (9) and (12), one can solve for T_{cl} in terms of T_s, T_a, etc. (noting that K= R + C from Eq. 7). Substituting this value into Eq. 12, combining it with Eq. 10, and solving for f_{cl_1}, generates the equivalent thermal efficiency factor implied in Fanger's model. The following is the result:

$$f_{cl_1} = \frac{1}{1 + 0.881\ h\ I_{cl}\ f_{cl_2}} \quad \text{(N.D.)} \qquad (13)$$

Comparing this result to Eq. 11 indicates that the only difference is the inclusion of the area ratio term, f_{cl_2} (indicated with the subscript 2 to minimize confusion as both the ET* and Fanger's models use the same expression, f_{cl}, to represent the two different quantities expressed here as f_{cl_1}, and f_{cl_2}, respectively). Fanger's area ratio factor can affect Burton's efficiency factor of Eq. 13 by as much as 40% for a heavy clothing ensemble. It is, therefore, evident that Fanger's method is to be preferred.

<u>I. 5 Convective Heat Loss</u>: The heat loss by convection from the outer surface of the clothed body can be expressed by the following equation:

$$C = A_{du}\ f_{cl}\ \ h_c\ (T_{cl} - T_a)\ \text{(Btu/hr)} \qquad (14)$$

where f_{cl} is the ratio of the surface area of the clothed body to the surface area of the nude body (f_{cl_2} of the previous section)

h_c is the convective heat transfer coefficient (Btu/hr - ft^2 - °F)

T_a is the logarithmic mean temperature of the air stream (°F) as defined by:

$$T_a = T_s - \frac{T_{ex} - T_{in}}{\ln \dfrac{T_s - T_{in}}{T_s - T_{ex}}}$$

T_{ex} is the dry bulb temperature of thoroughly mixed air after passing over the body length (°F)

T_{in} is the dry bulb temperature of the air stream prior to traversing the body (°F)

It should be noted that since the original PTI model was concerned only with very low air flow rates, the logrithmic mean ambient temperature, T_a, was a necessary requisite for precision. In the experimental study the inlet and exhaust temperatures, T_{in} and T_{ex}, respectively, were measured in addition to the skin temperature, T_s[8]. In this manner the ambient temperature was computed. For modeling purposes, however, we need only concern ourselves with the ambient temperature, T_a. It is this parameter, in addition to the ambient humidity, W_a (described by a following section), that will be used to define lines of constant human thermal comfort on the psychrometric chart.

For the thermal circuit, an equation can be derived to describe the variable convective resistance of the circuit. Rearranging Eq. 14 in the form of Ohm's law and noting that the temperature potential, $T_s - T_{cl}$, is analogous to the voltage potential and the heat flux, C/A_{du}, is analogous to a current flux, we find that:

$$R_{Conv.} = \frac{1}{f_{cl}\, h_c} \quad (hr - °F - ft^2/Btu) \qquad (15)$$

The next section will be used to describe in detail the convective heat transfer coefficient, h_c, since considerable controversy exists over its magnitude for various conditions.

<u>I. 6 Coefficient of Convective Heat Transfer</u>: One of the purposes of this report is to incorporate the effects of greater air velocities into the PTI model originally developed. The earlier model was intended for use in very low air ventilation rate applications, where it was shown that the classical approach of natural convection analysis does not apply.[1] A natural motion limiting value for the coefficient of convective heat transfer was obtained from an experimental study of low air flow rates performed in the Monoman calorimeter. It was concluded that the natural body movements of a sedentary human induce boundary air motions which overrides the weaker fluid forces produced by boyant means which would prevail for inanimate objects at low forced convection velocities. The average value of h_c for all tests performed was found to be equal to 0.76 Btu/hr - ft^2 - °F. It should be noted that when buoyant or artifically induced fluid forces create heat transfer coefficients that exceed this value, the natural motion limit concept no longer applies. Since air velocities will be dealt with in this model that will exceed the

limiting value, a suitable method must be found for incorporating these effects.

Synthesizing the work of many people before him, Kerslake generated an equation for the convective heat transfer coefficient as a function of air velocity. This equation is as follows:

$$h_c = 0.104\ V^{.5}\ (\text{Btu/hr - ft}^2 - {}^\circ\text{F}) \qquad (16)$$

where V is the air velocity in feet per minute.

It was recommended in an earlier Santa Clara University study[3] that the relationship of Fig. 3 be used for the convective heat transfer coefficient versus air velocity. Reference[9] (refer to Fig. 3), is Kerslake's relationship characterized by Eq. 16 compared to other references[10,11,12] that suggest correlations similar to Kerslake's.

The use of Eq. 16 is restricted by Kerslake to air velocities greater than or equal to 20 fpm. This corresponds to a minimum h_c value of 0.465 btu/hr - ft^2 - °F. For air velocities less than this, he suggests the use of natural convection techniques to determine h_c.

Since the natural motion limit concept has been verified by a large number of experiments, it will be used as a limiting value for h_c, and, therefore, natural convection techniques will not be considered. However, if Eq. 16 is to be used for the values of h_c that are prescribed by air velocity and if the natural motion limit value of 0.76 Btu/hr - ft^2 - °F is also to be used, it is seen that their interraction occurs near 50 fpm. This implies that the random, natural motions of a human being induce an effective relative air velocity of 50 fpm for agreement with Kerslake. This appears high since it is generally agreed that "still air" corresponds to about 30 fpm. It is also seen in the figure that the line recommended for use by an earlier Santa Clara study has a transition from the natural motion limit value starting near 10 fpm and terminating near 50 fpm at a value well above Kerslake's value for this velocity.

This evidence indicates that there is considerable uncertainty about the value of the convective coefficient in the natural motion region, in the transition region and the correlation where the coefficient is dependent upon air velocity. Further study needs to be given to this issue. However, Kerslake used the results of several independent workers and the natural motion limit value was determined from many tests. Hence it is

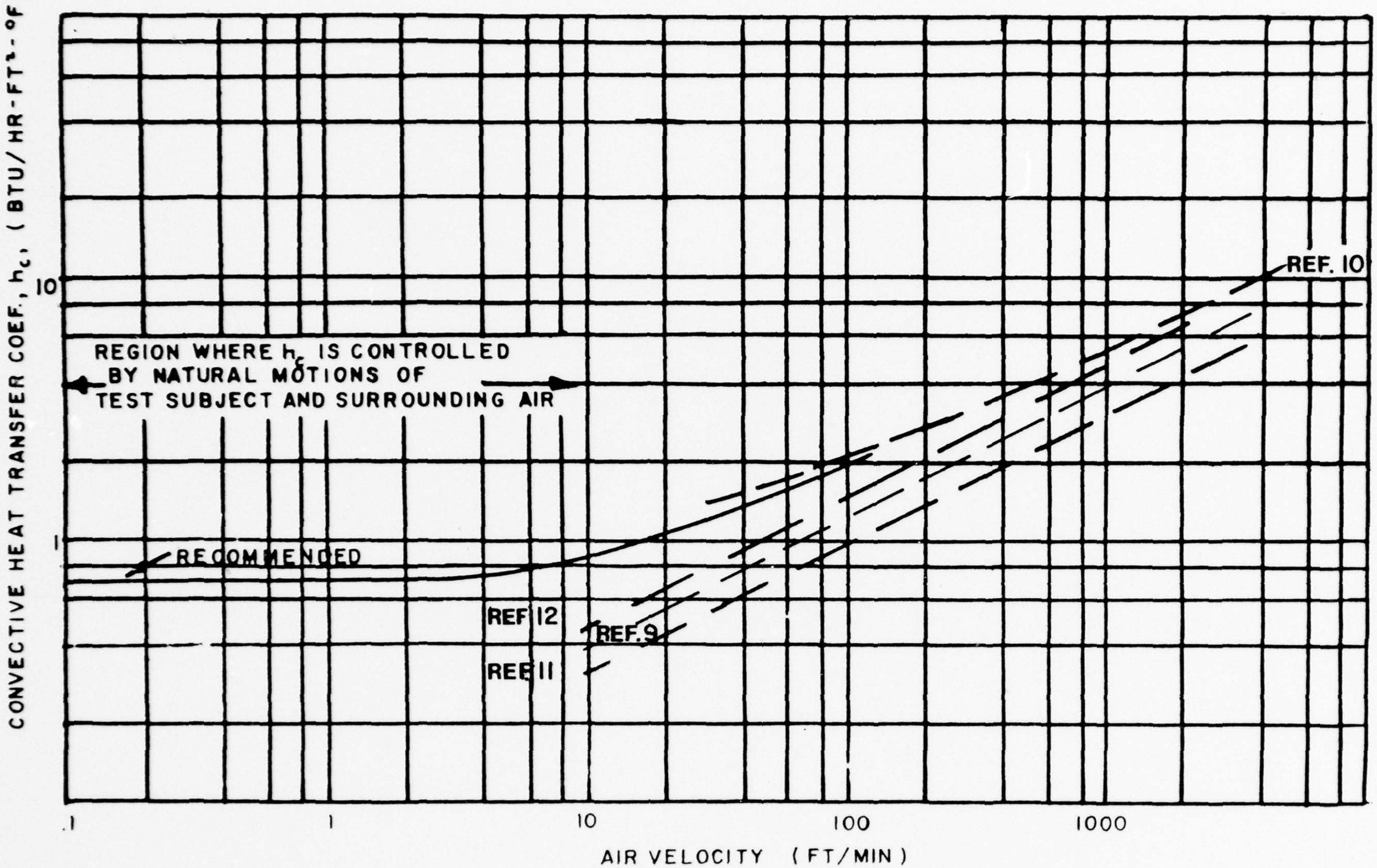

Figure 3 AVERAGE CONVECTIVE HEAT TRANSFER COEF. FOR A SEDENTARY HUMAN VERSUS AIR VELOCITY

difficult to deny the validity of these two pieces of evidence.

For want of better evidence for this study, the natural motion limit line is projected to 50 fpm and the natural motion limit value of h_c will be used in this region. For velocities in excess of 50 fpm the value of h_c given by Kerslake's equation (Eq. 16) is used. This method of resolving the controversy will be used in the PTI model, although, as previously suggested, additional research may produce a more satisfying resolution.

I. 7 Radiant Heat Exchange: Radiant heat exchange takes place between the human body and its surroundings, just as between any two physical objects. The heat loss by radiation from the outer surface of the clothed body can, therefore, be expressed by the Stefan-Boltzmann Law:

$$R = A_{eff}\ \varepsilon\ \sigma\ (T_{cl} + 460)^4 - (T_{mrt} + 460)^4\ (\text{Btu/hr}) \qquad (17)$$

where A_{eff} is the effective radiation area of the clothed body (ft^2)

ε is the emittance of the outer surface of the clothed body (N.D.)

σ is the Stefan-Boltzmann constant: 1.714×10^{-9} Btu/hr - ft^2 - $°R^4$

T_{mrt} is the mean radiant temperature (°F). The mean radiant temperature, in relation to a given person placed at a given point with a given body position and a given clothing, is defined as that uniform temperature of a black enclosure which would result in the same heat loss by radiation from the person as the actual enclosure under study.

The geometry of the radiant exchange between the body and the surroundings is complex due to the irregularity of the body. Since the body is not everywhere convex there will be some reradiation on the body surfaces and also between adjoining appendages and the torso. The area in Eq. 17 will, therefore, not be the actual surface area of the clothed body but a reduced area, called the effective radiation area. It is given by the following relationship:

$$A_{eff} = f_{eff}\ f_{cl}\ A_{du}\ (ft^2) \qquad (18)$$

where f_{eff} is the effective radiation area factor, i.e. the ratio of the effective radiation area of the clothed body to the surface area of the clothed body.

The value of f_{eff} has been determined by Fanger[4] and for the purposes of this study it was found to be 0.696 for sedentary body posture. The f_{cl} factor has been determined for various clothing ensembles. This study will consider subjects ranging from nude to those clothed in a light working ensemble. This range corresponds to I_{cl} values of 0, 0.5, and 0.6 with respective f_{cl} values of 1.0, 1.1, and 1.1.[4] These values were chosen to correlate with those used in the other models under study. Although the study is limited to these values, the computer program is written such that any correlating values of I_{cl} and f_{cl} can be used. It is recalled that I_{cl} is an important parameter in the clothing conduction equation.

Since the emittance for human skin is close to 1.0[4] and most types of clothing have emittances of about 0.95, a mean value of 0.97 is suggested for use according to Fanger.

The remaining parameter of concern is the mean radiant temperature. This variable is easy to define but very difficult to evaluate. For modeling purposes, however, it will be assumed that the mean radiant temperature is equal to the ambient temperature. This is obviously an invalid assumption if the subject is irradiated by some high intensity radiant field, such as that generated by the sun, but it is a reasonable assumption for an occupant subjected to typical indoor conditions. The two analytical models that are being studied in addition to the PTI model have made the same assumption for modeling, indicating that this parameter will not be of concern if (and when) discrepancies arise in the comparisons to follow.

The variable radiant resistance of the thermal circuit is given by:

$$R_{rad} = \frac{1}{f_{eff}\, f_{cl}\, \varepsilon\, h_r} \quad (hr - {}^\circ F - ft^2\ /Btu) \qquad (19)$$

where h_r is the linearized radiant heat transfer coefficient. The previous equation was derived in a manner analogous to that used to derive Eq. 15. In the computer program of the Appendix, the linear radiant heat transfer coefficient, hr, is tested and refined for every iteration of the program to within 1% error of the value that would be given by Eq. 17.

<u>I. 8 Evaporative Heat Loss:</u> The evaporative heat rejection from a human is composed of latent heat exchange within the respiratory tract, moisture diffusion through the dry skin area, and evaporation of the actively perspired water film from the wetted skin surface. The water is assumed to evaporate from the skin surface which is a good approximation even when the clothing becomes saturated. Since water is a good conductor compared to the clothing, the outer clothing temperature will approach the temperature of the skin when the clothes become saturated. Hence, the water will evaporate at a temperature

not too different from the skin temperature. However, it is noted that in this event, the conductive property of the clothing should no longer be considered when dealing with the clothing conduction equation. Since the clothing is saturated its conductive property will be a function of the water and cloth and it will, therefore, be increased significantly. At this point the PTI model will handle this problem only qualitatively for the sake of making its existence known with the hope that a feasible solution may be sought in the future. It should be emphasized that this problem arises only when the clothing of the body is at or near saturation. For survival shelters most clothing will be removed for such conditions and the issue is avoided. In the passive sweat region no inaccuracy will be incurred as a result of this problem.

Combining the effects of the sweat and diffused moisture evaporative exchange mechanisms into a single parameter, X, yields the following energy transfer rate equation:[1]

$$E = h_m \, h_{fg} \, A_{du} \, X \, f_{pcl} \, (W_s - W_a) \text{ Btu/hr)} \qquad (20)$$

where h_m is the average convective mass transfer coefficient based on a wet body area of XA (lb_{mwv}/hr - ft^2) (lb_{mwv}/lb_{mda})

h_{fg} is the latent heat of vaporization of saturated water vapor at T_s (an average value of 1039 Btu/lb_m, corresponding to a mean skin temperature of 95 °F, is used as it only varies 1% in the expected temperature range of T_s)

X is the fraction of surface body area, A_{du}, that is considered completely wet and combines the effects of moisture diffusion and active sweat losses (moisture loss from the body area is imagined to occur as though the body were divided into two distinct regions--one that is entirely wet and the remainder dry)

f_{pcl} is the permeation efficiency factor[7] for clothing given by:

$$f_{pcl} = \frac{1}{(1 + 0.813 \, h_c \, I_{clo})} \quad \text{(N.D.)} \qquad (21)$$

W_s is the humidity ratio that corresponds to saturated air at T_s (lb_m wv/lb_m da)

W_a is the logarithmic mean average of air outlet and inlet humidity ratios and is calculated by the following equation:[1]

$$W_a = W_s - \frac{W_0 - W_i}{\ln \frac{W_s - W_i}{W_s - W_0}} \; (lbm_{wv}/lbm_{da}) \qquad (22)$$

where W_0 and W_i are the humidity ratios of the air at exhaust and inlet conditions respectively.

As previously mentioned, we need only be concerned with the ambient specific humidity, W_a, for modeling purposes. This is the case since the ambient reflects the averaged conditions of the inlet and exit and will, therefore, be the condition that the human effectively feels.

It has been assumed, in deriving the evaporative equation of the PTI model given by Eq. 20, that the latent respiratory exchange is negligible. It will be shown in the following section that, negligible or not, this exchange mechanism is included in the h_m X product. One assumption involved in combining the evaporative exchange mechanisms into a single term, X, is that all evaporative exchange takes place at a common specific humidity ratio, W_s (corresponding to saturated air at T_s and, therefore, only a function of T_s). This is accurate for the evaporation of sweat at the skin surface. With regard to the moisture diffusion through non sweating skin in which the water is vaporized a very short distance beneath the skin surface, this assumption is believed to induce insignificant error. It is argued, therefore, that combining all these moisture loss effects into a single parameter is justified.

It is recognized that the mass transfer coefficient, H_m, is related to the convective heat transfer coefficient, h_c. It was argued previously that the convective heat transfer coefficient is a function of air velocity when the natural motion limit coefficient no longer applies. It is recognized that the air velocity is having an affect on both h_c and h_m although h_m is only associated with the wet skin fraction, X. As will be seen in the study of Fanger's model, there is some controversy over whether or not air velocity affects h_m for the sweating skin area to the same degree as it does h_c which is related to the entire body area. In the original PTI model, which was concerned only with very low air flow rates, the h_mX product was defined and evaluated such that it was identically influenced by all three evaporative exchange mechanisms.[8] For air velocities that produce h_m values exceeding the limiting case, a method will be discussed which utilizes Fanger's respiratory exchange equation to examine the influence of larger h_m values on the respiratory component of the h_m X product. This was done since it is obviously not valid to weigh the respiratory loss by a mass transfer coefficienty governed by air velocity. The analysis is accomplished in the following section dealing with the comparison of the PTI model to Fanger's model.

The variable evaporative resistance of the thermal circuit defined in Eq. 20 is given by the following relation:

$$R_{evap} = \frac{1}{h_m h_{fg} f_{pcl} X} \text{ (hr - (lb}_{mwv}/\text{lb}_{mda}\text{) - ft}^2/\text{Btu)} \qquad (23)$$

I. 9 Coefficient of Mass Transfer: The coefficient of mass transfer is also subject to the natural motion limits for very low air flow rates. In previous studies at Santa Clara University, the h_m X product of the evaporative exchange equation was found as a function of the skin temperature, T_s. A curve fit of the experimental data generated this relationship, which is reproduced in Fig. 4a. This curve is only valid for the natural motion limiting conditions. Since the skin wetness fraction, X cannot exceed unity, the natural motion limiting value of h_m can be determined from the figure. This value was found to be 4.22×10^{-4} lb_m/hr - lb_f or in units consistent with the computer program developed, it is equal to 1.4 lb_m/hr - ft^2. Dividing the ordinate of Fig. 4a by h_m, which was assumed constant for the tests, the X - T_s curve of Fig. 4b was generated. This relationship is not dependent on the natural motion argument. It is believed to be dependent only upon the activity level of the human subject.[1] It will be shown in a subsequent section that the X - T_s curve defines a scale of human thermal comfort, namely, the Physiological Thermal Index.[8]

Before leaving the subject of mass transfer coefficients we must provide a means for handling the change in h_m that will occur when air velocities induce relative motion that cause the value of h_m to exceed the natural motion limiting value. Kerslake, again synthesizing the works of others, generated the following relationship for h_m as a function of air velocity:

$$h_m = 0.416\, V^{.5} \text{ (lb}_m\text{/hr - ft}^2\text{)} \qquad (24)$$

where V is the air velocity in feet per minute.

It is seen that when the natural motion limiting value of h_m is used in this equation, it corresponds to an air velocity of about 11 fpm. This value correlates better with what was recommended in Fig. 3 as an upper limit

UNIT AREA EVAPORATIVE BEHAVIOR

$h_m = (LB_M/HR\ LB_F)$

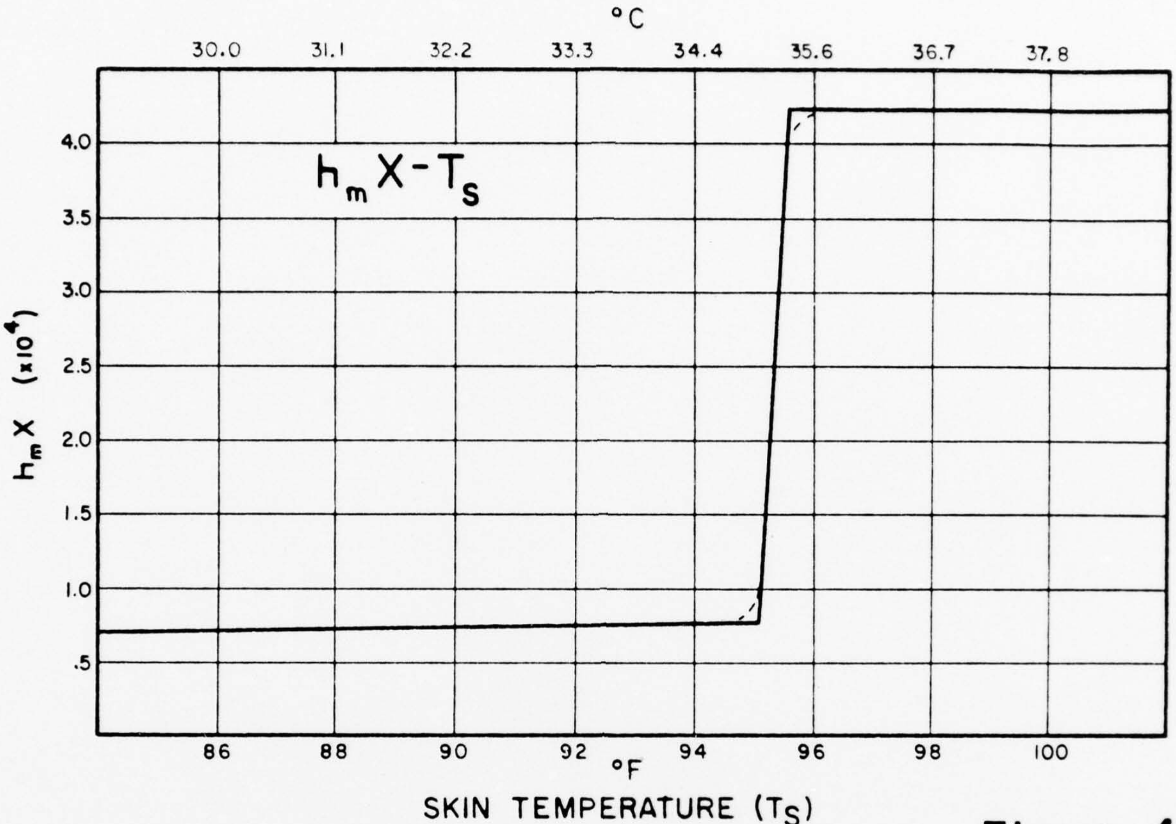

Figure 4a

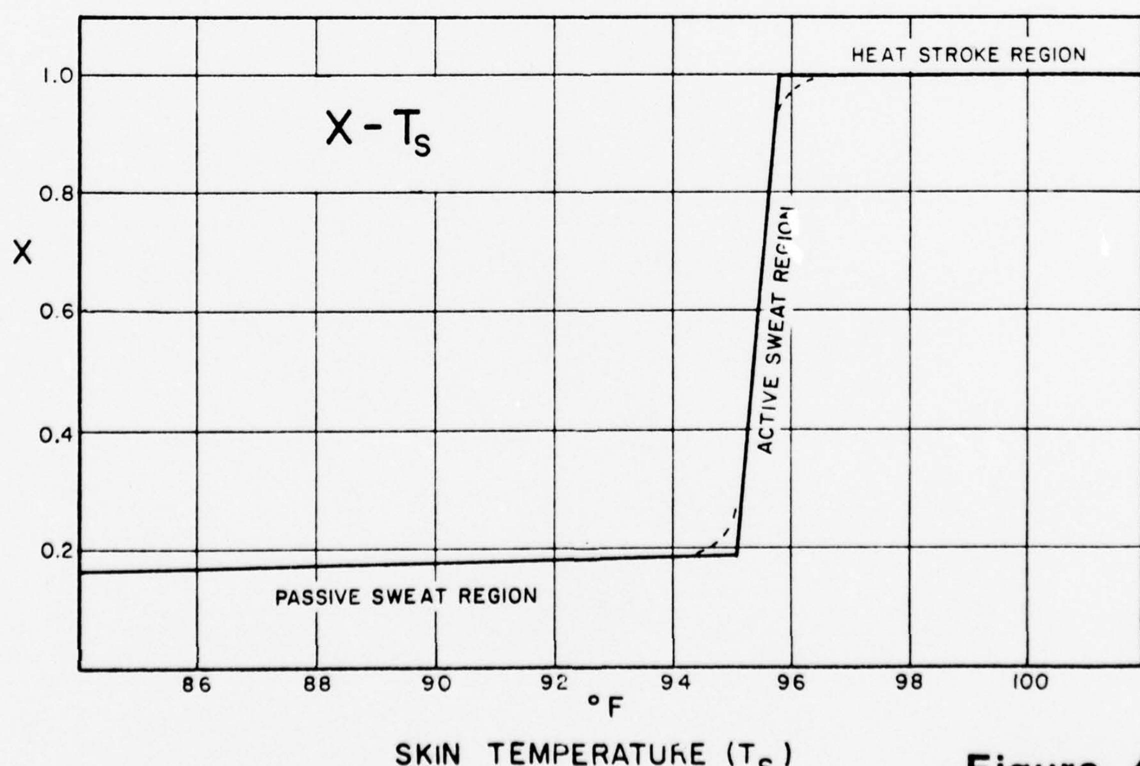

Figure 4b

air velocity for "still" air than did the natural motion convective coefficient.

If this equation were to be used for air velocities exceeding 11 fpm, and Eq. 14 for air velocities exceeding 50 fpm, it should be noticed that this would result in a non-constant Lewis Relation over the 11 - 50 fpm air velocity range. It is generally accepted that the coefficients of heat and mass transfer correlate linearly (i.e., L.R. is constant) for animate subjects, as it is known that they do for inanimate objects. However, the physiological functions of the body may preclude this assumption, justifying a non-constant relationship. Since to date there is no evidence to support this latter concept, the PTI model will adopt the policy of using the natural motion limit coefficients for air velocities up to 50 fpm. For air velocities exceeding this value, it will use the Lewis Relation determined by the ratio of the natural motion limit coefficients and the convective heat transfer coefficient will be given by Kerslake (equation 6). These two quantities determine the mass transfer coefficient for air velocities exceeding 50 fpm given by the following relation:

$$h_m = 0.190\ V^{.5}\ (lb_m/hr - ft^2) \qquad (25)$$

It is noted that there is considerable discrepancy between this expression and that given by Kerslake's equation 24. Equation 25 has used Kerslake's convective heat transfer coefficient equation which can be justified to some extent since it is agreed that the value of h_c is known to a greater degree of certainty than is the relationship of h_m to air velocity.[9] It is important, however, to recognize the discrepancy between Equations 24 and 25. The only present justification for using the procedure outlined is that it generates good comparative results with the other human thermal comfort predictive models studied in this report. However, to let the inference of the previous statement be the sole guide from which results of the PTI model are to be secured is unsatisfactory. An in depth comparative analysis is performed in Section III which deals with the New Effective Temperature Scale. There the effects of all the important parameters, including the Lewis Relation and the heat and mass transfer coefficients are examined.

The previous statements should further indicate a need for additional research in the complex area of heat and mass transfer coefficients and their relationship to one another in the study of human thermal comfort.

I. 10 Thermal Circuit: Having described and defined each of the components of the energy balance given by Eq. 7, which correlate with the thermal circuit through the application of Kirchoff's current law, and substituting those expressions into Eq. 7, generates the following double heat balance equation for thermal equilibrium:

$$\frac{MR}{A_{du}} - h_m h_{fg} \times f_{pcl} (W_s - W_a) = \frac{(T_s - T_{cl})}{0.881\ I_{cl}} =$$

$$f_{cl}\ h_c\ (T_{cl} - T_a) + f_{eff}\ f_{cl}\ \varepsilon\ h_r\ (T_{cl} - T_{mrt}) \qquad (26)$$

where, as explained previously, the work term, W, is equal to zero (sedentary state).

Equation 26 is the desired general comfort equation for thermal equilibrium. The comfort equation contains the following variables:

I_{cl}, f_{cl} which are functions of the type of clothing

MR/A_{du} is a function of the activity level

T_a, W_a, T_{mrt} are environmental variables

V is partially affected by the type of activity but is primarily a function of the environment (V determines magnitudes of h_c and h_m when natural motion limit coefficients are not in force)

f_{eff} is a function of body posture

It will be shown in the following section that for a specified comfort indicated by the X - T_s curve of Fig. 4b, a series of ambient dry bulb temperatures, T_a, and correspondingly a series of specific humidities, W_a, can be used to generate the same feeling of comfort (or discomfort). The X - T_s relationship is one of the primary inputs of Eq. 26 defined by the PTI model. This relationship is valid only for the sedentary activity level and, therefore, the discussion and application of the equation will be limited to this condition.

The researchers at Santa Clara are presently working to extend the PTI model to different activity levels. The result will be a series of X - T_s curves determined by the various activity levels which will allow the utilization of Eq. 26. Note this requires reinsertion of the work term of Eq. 7 over the range of activity levels.

I.11 The Physiological Thermal Index and Lines of Constant Comfort: The relationship described by Fig. 4b provides the basis for the Physiological Thermal Index (PTI). The Index is taken as one's physiological location on the X - T_s curve. The value assigned from the physiological scale to any point on the X - T_s curve represents a fixed physiological state regardless of the particular environment which produced this state. The index was defined by weighing equally the metabolic energy dissipative power that is related to the skin wetness, X, and the skin temperature, T_s. The resulting PTI scale appears in Fig. 5.[8]

Considering the PTI values of Fig. 5 as independent variables, the corresponding values of X and T_s are fixed and are, therefore, dependent variables for use in Eq. 26. The following is a list of the independent and dependent variables and constants of appropriate units to be used for this study in Eq. 26:

Independent	Dependent	Constant
PTI No.	X, T_s, W_s	MR/A_{du} = 20 Btu/hr - ft^2 (sedentary male)
V	h_c, h_m, h_r	h_{fg} = 1039 $^{Btu}/lb_m$
I_{cl}	f_{cl}, f_{pcl}, T_{cl}	f_{eff} = 0.696 (sedentary)
W_a	T_a	ε = 0.97

In the previous list it has been assumed, as stated before, that the mean radiant temperature, T_{mrt}, is equal to the ambient temperature, T_a. For this reason it has been omitted. It should also be noted that the coefficients h_c and h_m are dependent on the air velocity, V, only when the natural motion limit values do not apply. The linear radiant heat transfer coefficient, h_r, is considered dependent since it will vary as the difference between T_{cl} and T_a varies. In actually generating the constant comfort lines the independent variables consisting of the PTI No., V, and I_{cl} were taken as parameters and, therefore, specified for the generation of each line. It is further noted that having specified each of these parameters there are three unknowns in the two equality statements of Eq. 26, namely, W_a, T_a, and T_{cl} which indeed indicates that an infinite number of solutions exist.

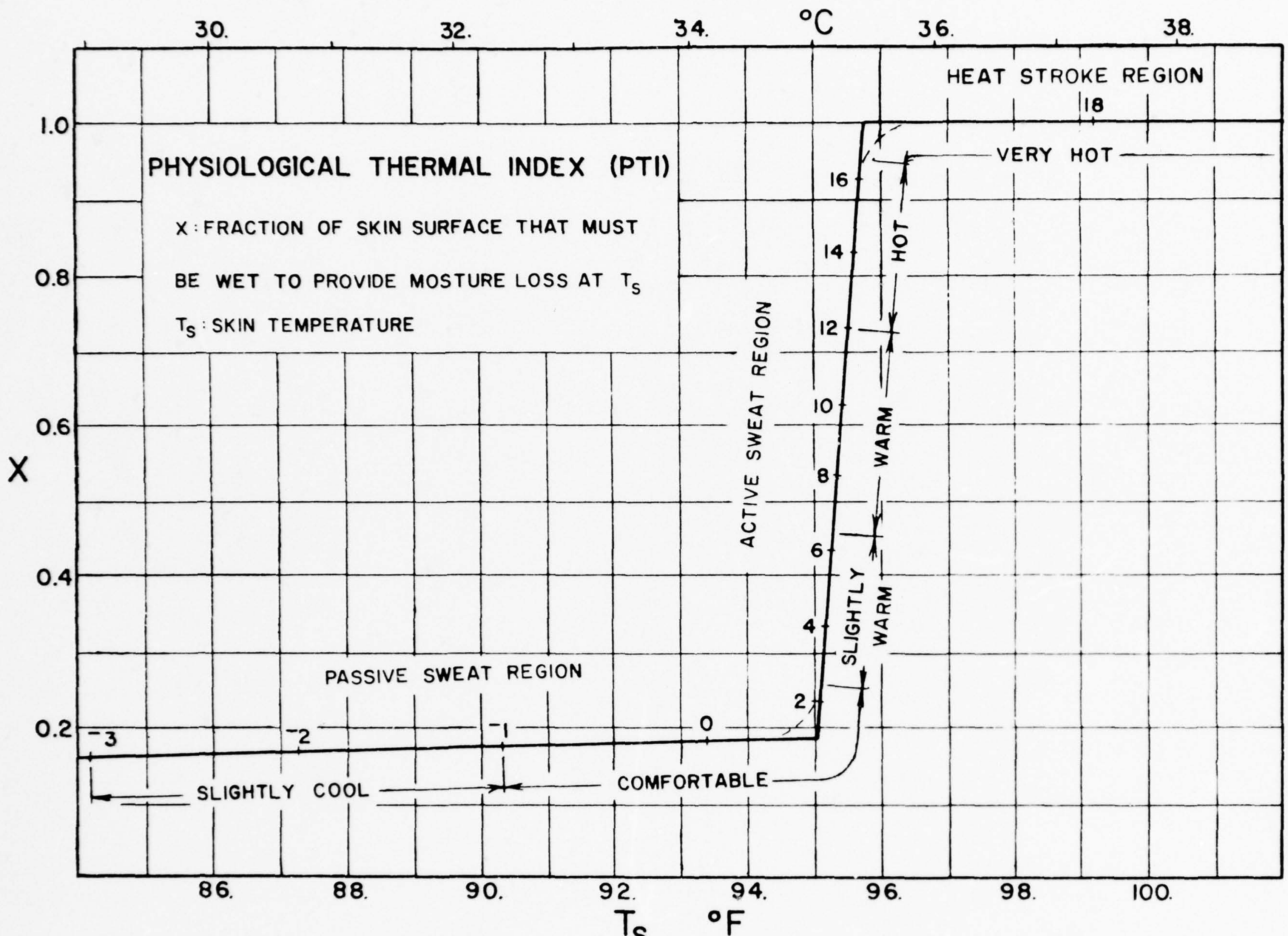

Figure 5 BODY EVAPORATIVE BEHAVIOR V.S. SKIN TEMPERATURE

The specific equations used to generate the lines of constant comfort on a psychrometric chart can now be developed. Solving first for the evaporative heat loss of Eq. 20:

$$\frac{E}{A_{du}} = h_m h_{fg} X f_{pcl} (W_s - W_a) \qquad (27)$$

where, for a given PTI value, V, and I_{cl}, the following are knowns:

X, W_s, h_m and f_{pcl}; h_{fg} is constant and W_a is the independent variable for which a number of arbitrary values are taken. The results are calculated values of E/A_{du}.

The sensible heat transfer of Eq. 7 is then given by:

$$\frac{K}{A_{du}} = \frac{MR}{A_{du}} - \frac{E}{A_{du}} \qquad (28)$$

from which the temperature of the outer surface of the clothing, T_{cl}, is solved from Eq. 9:

$$T_{cl} = T_s - \frac{(K/A_{du})\, I_{cl}}{1.135} \qquad (29)$$

The ambient temperature, T_a, from Eq. 26 is then given by:

$$T_a = T_{cl} - \frac{K/A_{du}}{(h_c + h_r f_{eff}) f_{cl}} \qquad (30)$$

The complete set of equations can be found in the computer program of the Appendix. Using this program the average air states, (comprised of W_a and T_a), sensible and evaporative heat outputs, inlet and exit air states that correspond to constant skin temperature and constant skin wetness (i.e., air states of equal PTI or comfort value) were computed.

The resulting sets of lines appear in Figs. 6 and 7 and are based on an average male metabolic rate of 400 Btu/hr and DuBois surface area of 20 ft^2. The air velocity and clothing effects are as indicated. The PTI values represent the range from a cool sensation to the maximum tolerable limit where the body is completely wet. The zero PTI line indicates neutral thermal comfort. It is noticed that in the cool region the PTI lines are steep and as the value of the PTI line increases the slope decreases to a minimum value which is represented by the 17.1 PTI line. Beyond this line the slope remains the same since the surface

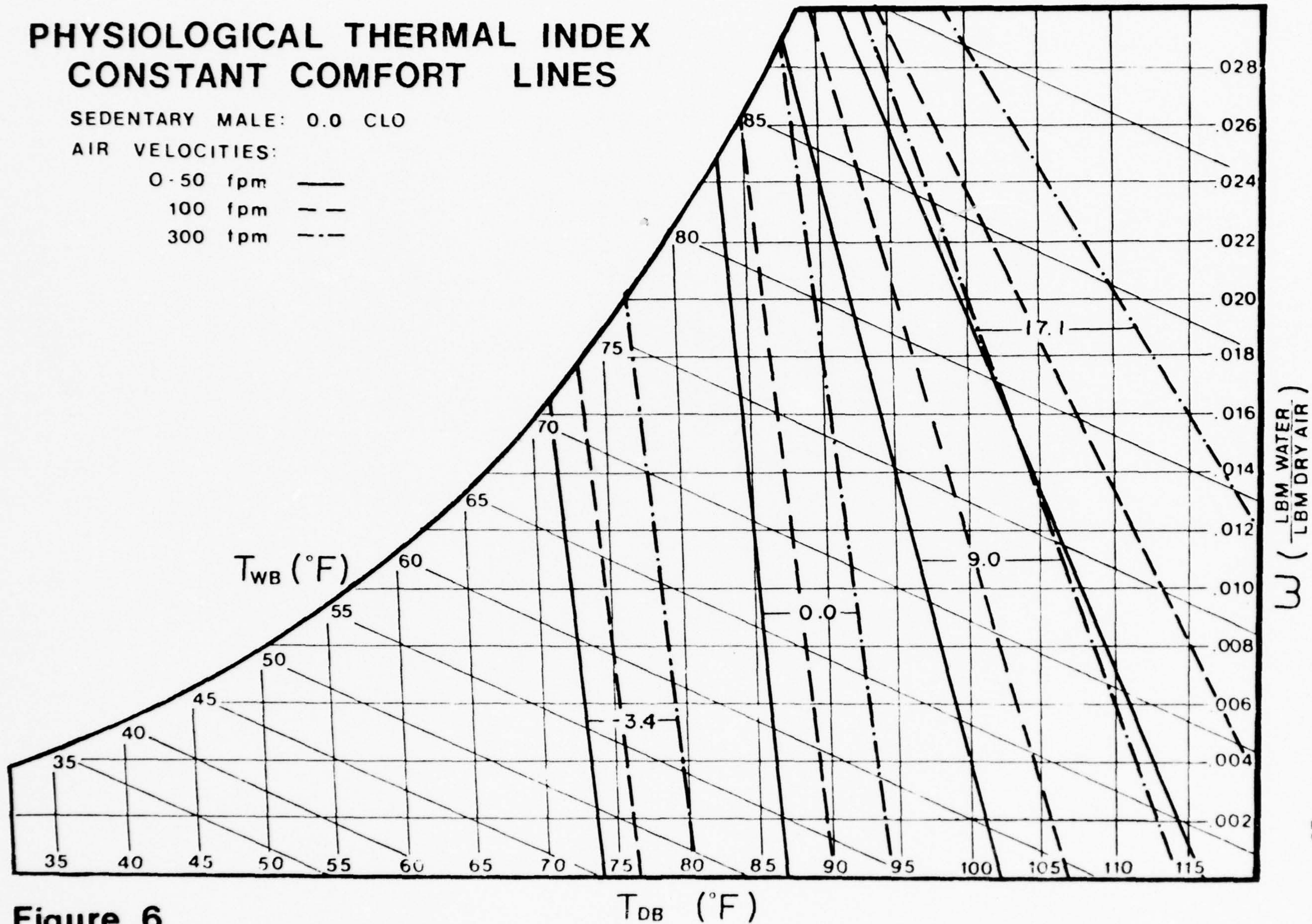
PHYSIOLOGICAL THERMAL INDEX
CONSTANT COMFORT LINES
SEDENTARY MALE: 0.0 CLO
AIR VELOCITIES:
0-50 fpm
100 fpm
300 fpm
T_{WB} (°F)
35
40
45
50
55
60
65
70
75
80
85
17.1
9.0
0.0
-3.4
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
ω (LBM WATER / LBM DRY AIR)
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115
T_{DB} (°F)

Figure 6

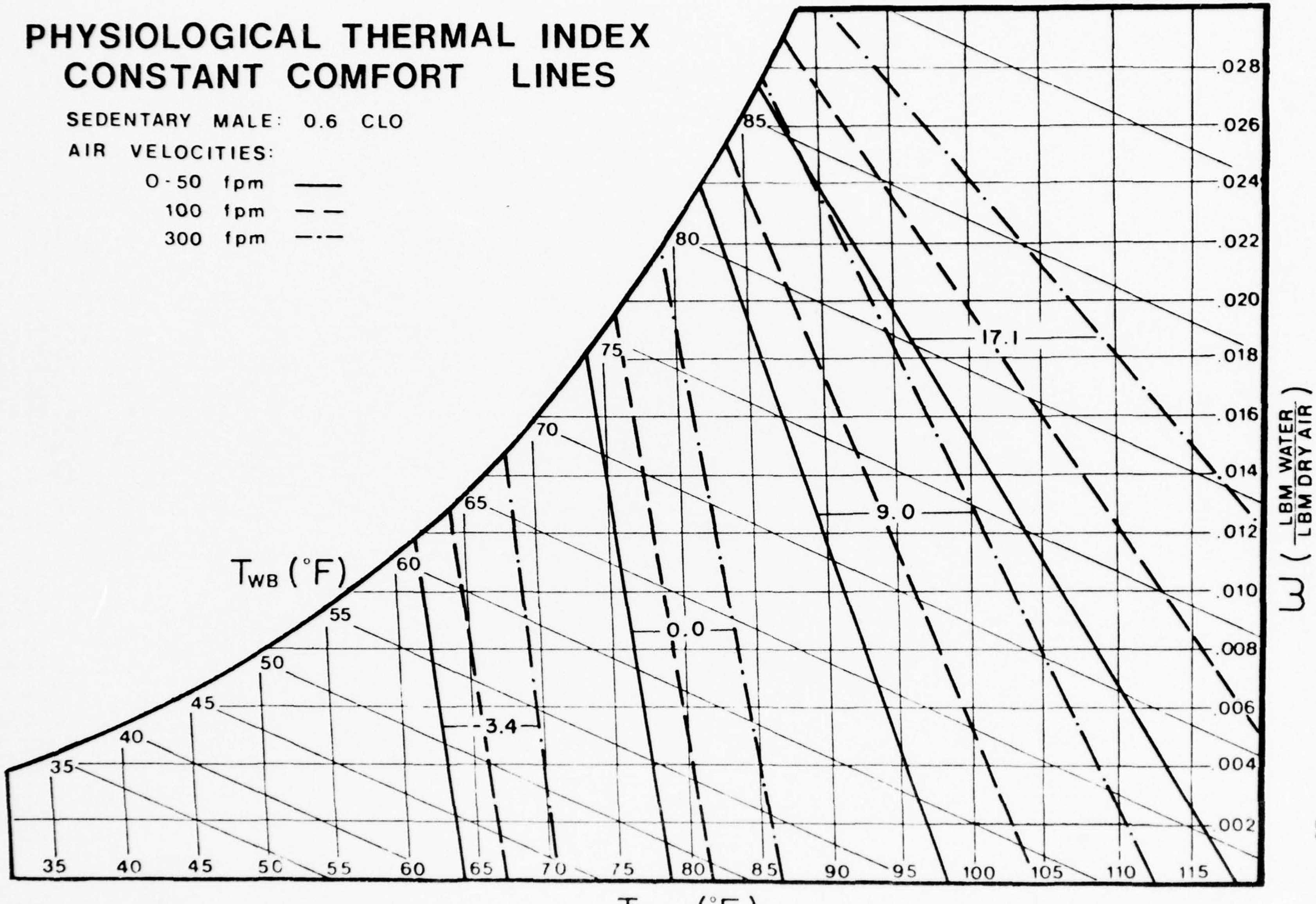
PHYSIOLOGICAL THERMAL INDEX
CONSTANT COMFORT LINES
SEDENTARY MALE: 0.6 CLO
AIR VELOCITIES:
0-50 fpm
100 fpm
300 fpm
T_{WB} (°F)
35
40
45
50
55
60
65
70
75
80
85
17.1
9.0
0.0
-3.4
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
ω (LBM WATER / LBM DRY AIR)
35 40 45 50 55 60 65 70 75 80 85 90 95 100 105 110 115
T_{DB} (°F)

Figure 7

wetness remains constant ($X = 1$) and the skin temperature rises rapidly. From this observation, it is concluded that the specific humidity ratio of the environment has slight effect on comfort at lower PTI values, while at the higher values it has a pronounced effect. This can be interpreted by noting that the skin temperature is the dominant index of comfort in the region of passive sweating (see Fig. 5), while the skin wetness is the dominant index of comfort in the active sweating region. It is further observed in comparing Fig. 6 to Fig. 7 that for any given comfort state of Fig. 6 the corresponding state of Fig. 7 is displaced to a cooler environment, indicating the insulating properties of the clothing. In a reciprocal manner, increasing the air velocity displaces the line to a warmer environment, indicating the increased cooling capacity that results.

A person interested in providing a microclimate suitable for a specified group of occupants can use this technique to determine such an environment. This has been done in previous studies relative to fall-out shelters and fire safe sanctuaries for specific circumstances.[1,13] Similarly, for a given environment, one can determine whether or not it is suitable for occupation and if it is not, the air velocity and clothing parameters can be adjusted (within reasonable limits) until it is. For the sake of not producing an overwhelming set of results, the parameters were confined as indicated, but by using the program of the Appendix any clothing value, air velocity or comfort state can be examined.

II. PREDICTIVE MEAN VOTE INDEX

In his book Thermal Comfort, P. O. Fanger synthesized the ongoing work at the Laboratory of Heating and Air Conditioning, Technical University of Denmark, and at the Institute for Environmental Research, Kansas State University. He recognized the fact that the existing knowledge of thermal comfort was quite inadequate and that many unresolved discrepancies exist among the researchers in this field. Although his model was developed for the purpose of resolving the issues of thermal comfort, he raises additional controversial topics that must be explored. His model,* in relation to the PTI model and other models of a similar nature, reveals several areas of agreement, but some issues warrant discussion as a result of their differences. His model is seen, therefore, as are all the models contained herein, as a means toward an end, but certainly not an end in itself. Fanger's work was primarily concerned with the conditions of neutral thermal comfort but, contrary to the PTI model considers various activity levels. Since the PTI model presently considers only one activity level (sedentary), but the entire comfort range, we are necessarily limited in the scope of our comparison. The common zones of comparison correspond to the neutral thermal comfort region of the PTI model and the sedentary activity level of Fanger's model. The results from Fanger's model in comparison with PTI evidence are shown in Figs. 8-13. These figures include the effects of air velocity and clothing.

In Fig. 5 of the previous section a comfort zone was specified. It was bounded by $-1 \leq \text{PTI} \leq 1$. This zone has been selected for comparison with Fanger's comfort model. As comfort is a subjective human response, it is more appropriately described by zone than by a single line.

It should be noted that the PTI lines do not change for air velocity values of 20 and 40 fpm as the natural motion limit heat and mass transfer coefficient values have been adopted for air velocities to 50 fpm. For the air velocities of 100 and 300 fpm, the Lewis relationship (LR) is used to determine the mass transfer coefficient from the heat transfer coefficient. The ratio of values is held equal to that

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* In this study called the PMV model for reasons explained subsequently.

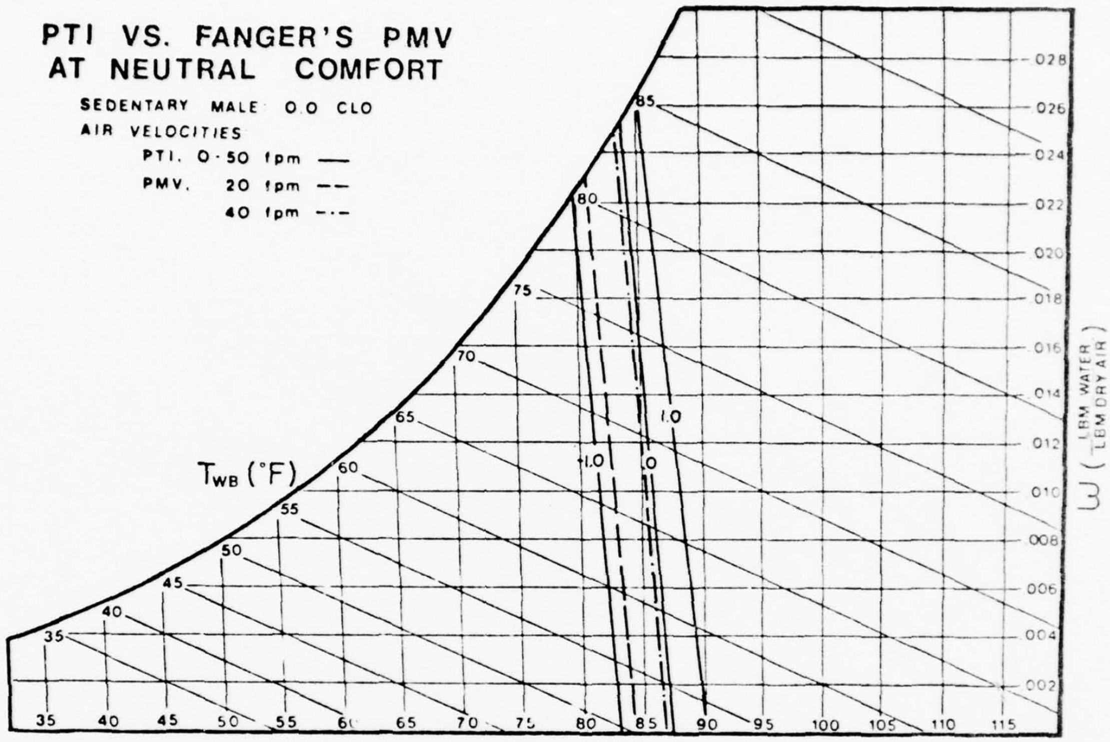


Figure 8

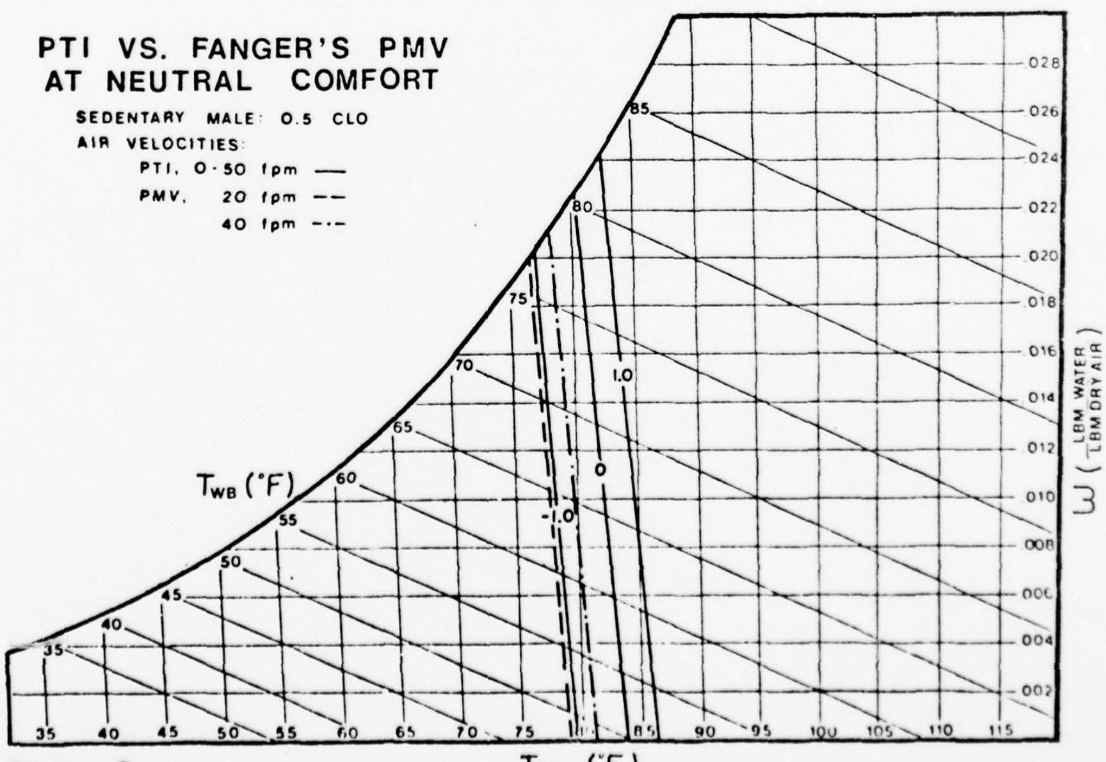


Figure 9

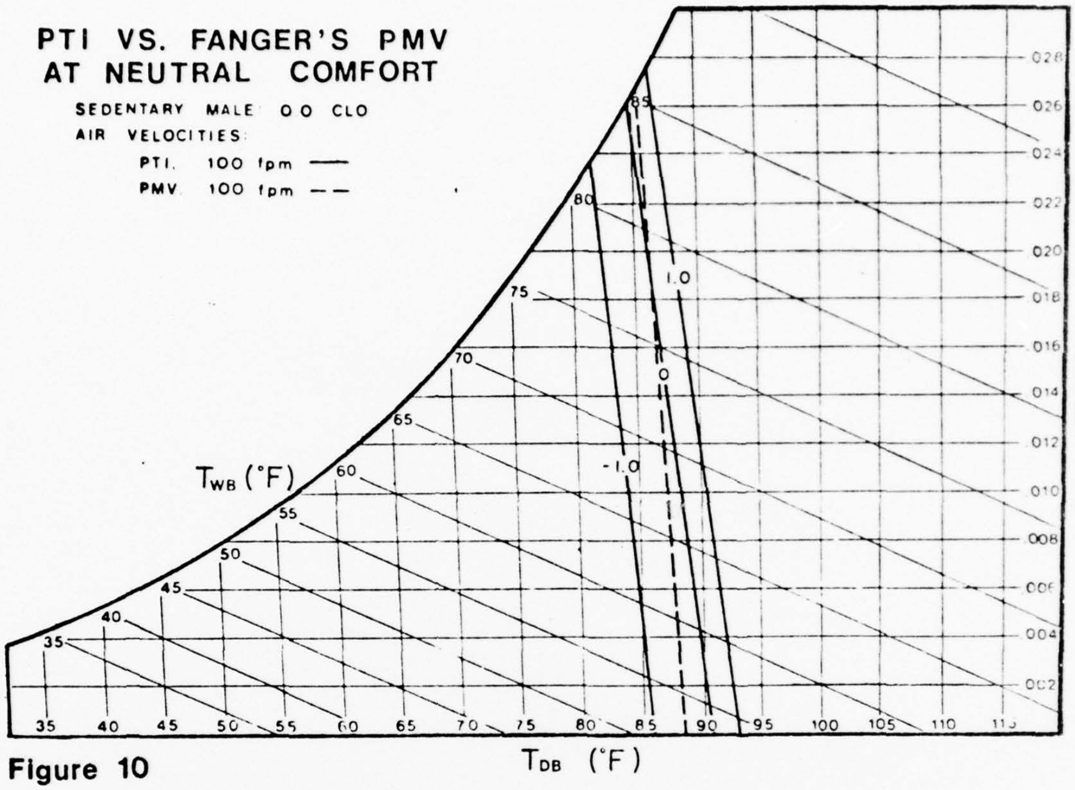


Figure 10

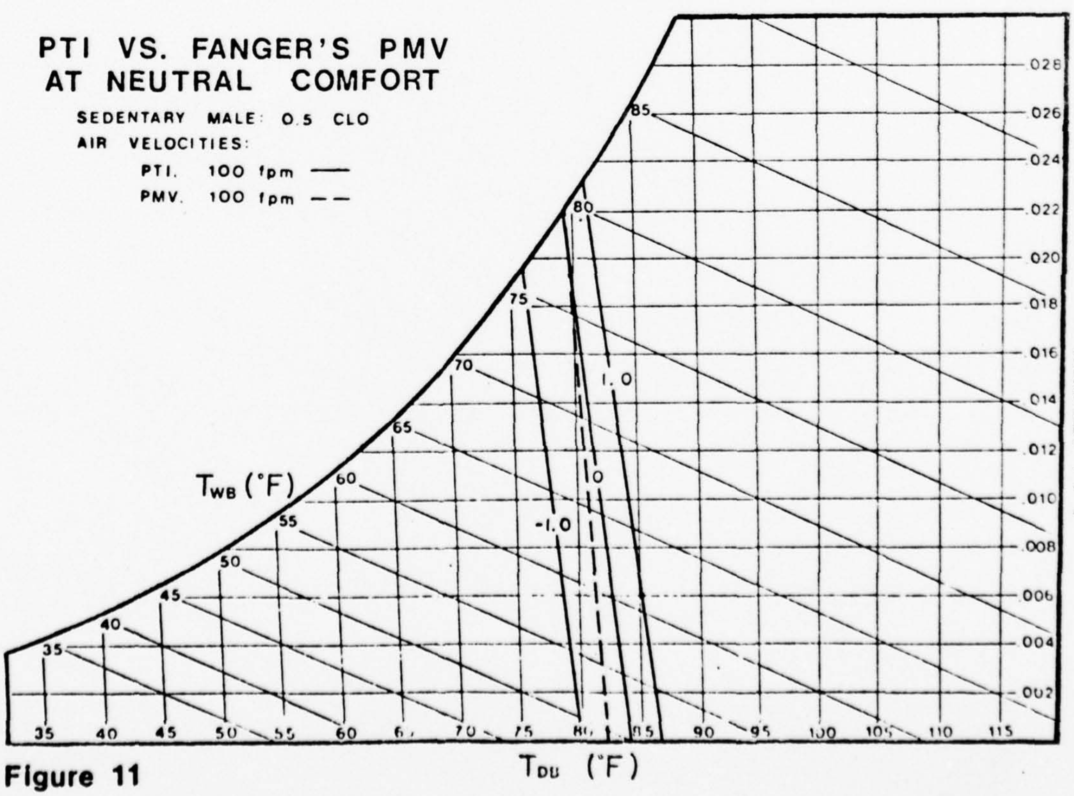


Figure 11

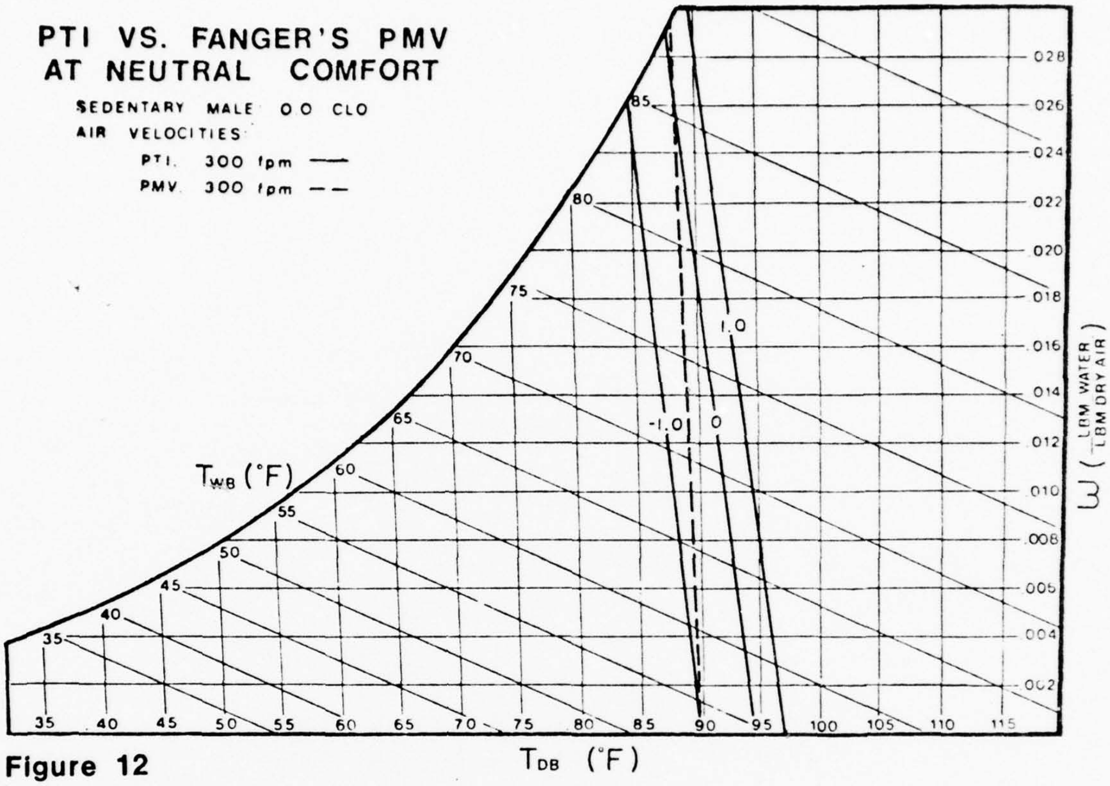


Figure 12

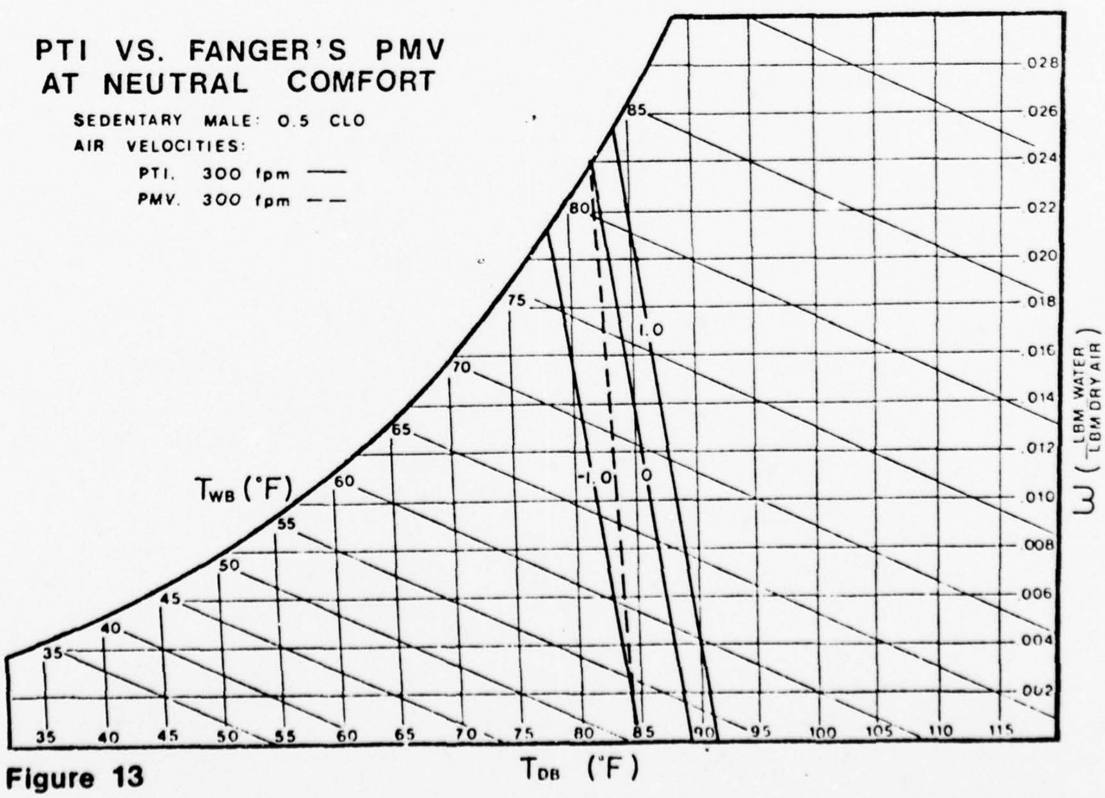


Figure 13

for natural motion limit:

$$LR = \frac{h_m}{h_c} = \frac{1.4}{0.765} \quad \frac{lbm - °F}{Btu}$$

This relationship is used in conjunction with h_c *which is related to* air velocity through Eq. 16 to provide values of h_m in Eq. 20.

In addition, the metabolic rate per unit of body surface area, MR/A_{du}, was taken as 18.5 Btu/hr - ft^2 for the PTI model to bring it into alignment with the Fanger model for this comparative study. As you may recall from previous discussions here and in other PTI model studies, 20 Btu/hr - ft^2 has been used for sedantary adult makes. This was based on literature evidence. Since that time, both Fanger[4] and Gagge, et. al.[5] have recommended the lower value.

The source of this discrepancy comes not in a dispute over the average metabolic rate of a sedentary male (agreed to be approximately 400 Btu/hr) but in the average DuBois skin surface area that should be used for a male. The PTI model has taken an average value of about 20 ft^2 whereas Fanger and Gagge have taken it to be 21.5 ft.^{2} In the ASHRAE literature[14], however, in which the ET* model is presented, a DuBois surface area was computed for an average adult male (170 lbs. and 5' 10"). The result was a DuBois surface area of 1.8 m^2 or 19.4 ft^2. Gagg has evaluated the DuBois area based on a subject of the same height but used an average weight of 180 lbs. Apparently Fanger used the same values for their DuBois area coincides. The 1976 Statistical Abstract, however, which is a collection of statistics gathered by the United States Government indicates that for an adult male of height 5' 10" his weight will be between 170 and 175 lbs. on the average. Even though this argument supports the use of the DuBois area recommended by the PTI model, Fanger's and Gagge's areas are used for the comparisons of the PTI model to their respective models. Again, this was done to eliminate an obvious discrepancy in the models for the sake of exposing those that are not so obvious nor so easily resolved. The effect of having used the PTI DuBois area would have been to displace the PTI lines of Figs. 8-13 slightly to the left. A cooler environment (< 2°F) would be required as a result of distributing the same metabolic rate over the smaller area.

Another point that should be noted is that Fanger recommends the use of the 20 fpm line for air velocities equal or lower than this value. He appears to recognize the natural motion limiting concept to the value of the heat transfer coefficient although he does not discuss it. Yet, in the text of his book, he deals with the concepts of free and forced convection for air velocities less than 20 fpm. This apparent contradiction may be indicative of Fanger's uncertainty on this particular issue. He also avoids the need for a mass transfer coefficient by restricting his model to passive sweat conditions.

In general the PTI model and Fanger's model correlate remarkably well. It is seen that at the lower air velocities, for both the nude and 0.5 clo cases, the lines are parallel, indicating that the relative weighing of the sensible heat transfer to evaporative heat transfer is the same. It is noted that both models incorporate radiant coupling effects but assume the mean radiant temperature equal to the ambient air temperature. There is particularly good agreement with the 40 fpm line of Fanger to the 0.0 PTI line for the 0.0 clo or nude condition. In all cases, moreover, with the exception of the 20 fpm, 0.5 clo line (Fig. 9), Fanger's constant neutral thermal comfort lines are well within the defined PTI comfort region.

The displacement of the lines relative to one another, while being parallel (0 - 50 fpm, 0.0 & 0.5 clo comparisons) is due to the models having somewhat different energy rate equations for the energy exchange modes, yet maintaining the same ratio of the modes. At the higher air velocities, however, the slope of the lines begin to deviate, growing worse for increasing air velocities. This is observed in the 100 to 300 fpm range. The reason for this deviation is that Fanger uses an evaporative energy exchange equation that is independent of air velocity. The equation is a function only of the humidity difference prescribed by the skin surface and the ambient air.

At conditions of thermal neutrality there is no active perspiration and, therefore, the only mass transfer that takes place is due to moisture diffusion through the skin and water vaproization in the respiratory tract. Since the diffused moisture is not visible liquid at the surface of the skin, it is agreed that the moisture is not subject to vaporization at the skin surface but at some distance beneath the surface. Fanger maintains that the main barrier to the rate of vapor diffusion results from the deep layers of the horny layers of the epidermis.[4] One would, therefore, be tempted to infer that Fanger regards the diffusion rate constant throughout the neutral comfort range, as it would be determined only by a constant internal diffusion resistance for a specified state of comfort. This temptation arises since Fanger implies that the resistance to vapor diffusion beneath the skin surface is large compared to the resistance generated by the boundary layer at the interface of the skin and ambient air. The mass transfer rate at the skin, air interface is a function of a mass transfer coefficient and humidity difference. However, as previously indicated, Fanger partially accounts for these factors by taking into account the humidity difference at the skin, air interface. This he combines with the diffusional resistance in the skin tissue. It is not clear why he omitted the mass transfer coefficient. In effect, he is treating it as a constant. The PTI model considers both the mass transfer coefficient and humidity factor and since the former of these is a function of air velocity, the PTI constant comfort lines exhibit a greater degree of humidity dependence than do Fanger's lines for increasing air velocity.

Furthermore, Fanger states explicitly that the sub-skin surface resistance to vapor diffusion is large compared to the resistance generated by clothing. Apparently for this reason Fanger chose not to incorporate a factor to account for the moisture permeation efficiency of different clothing ensembles. A. P. Gagge et. al.,[7] however, devoted a research project to quantifying the moisture permeation of clothing. To this end they developed the relationship given by Eq. 21 which is incorporated in the PTI model. Since the PTI model analysis used this factor and Fanger did not, we can account for the different relative displacements of the comfort lines that occur between the nude (0.0 clo) and clothed (0.05 clo) cases. This can be seen by observing the displacement between the PTI lines and Fanger's lines of Figs. 8, 10, and 12 (0.0 clo) and comparing them to the displacements of Figs. 9, 11, and 13 (0.5 clo), respectively. The reason we can infer that this factor is the cause for different relative displacement is due to the fact that the PTI model has implemented Fanger's clothing conduction concepts, thus, eliminating the possibility that this could have interjected any additional discrepancy. As a check the permeation efficiency factor, f_{pcl}, was taken out of the PTI model which resulted in the same relative positioning or displacement of the lines of Figures 9,11, and 13 when compared to Figs. 8, 10, and 12, respectively. This clearly proved this factor is indeed the cause of the shift.

The PMV notation used to describe Fanger's model in the figures refers to the index that was developed to be indicative of different states of comfort. The "Predicated Mean Vote" or PMV is based on a scale of -3 to 3 with divisions of one that represent comfort conditions ranging from cold to hot. By combining an empirically derived comfort response equation with his general heat balance equation, Fanger developed the Predicted Mean Vote Index.[4] The heat balance equation, which forms an integral part of this index, was developed for the conditions of neutral thermal comfort (PMV = 0). It was this equation alone that was used to generate the comfort lines of Figs. 8-13. The empirical comfort response relationship was combined with this equation to give a measure of the deviation from a zero PMV value for different environmental conditions. In developing the PMV Index, Fanger did not, therefore, consider active perspiration although the PMV range considers a "hot" (PMV = 3) thermal sensation. In this region of thermal comfort, perspiration or visible signs of liquid at the skin surface are present. This moisture is certainly influenced by a mass transfer coefficient, so it would appear that Fanger's model suffers from this oversight. It was for this reason that comparisons have been limited to the confines of neutral thermal comfort; a condition where active perspiration is not present. It appears that Fanger centered his attention on this comfort region and incorporated the "cold" to "hot" sensations as something of an add on to the neutral thermal comfort region.

In the overall comparisons of the PTI model to the PMV model we have so far noted two points of controversy.

1. Is the moisture diffusing through the skin under passive sweat conditions affected by the surface mass transfer coefficient?
2. Is free convection heat and mass transfer predicted for inanimate objects overridden by natural motions of the human?

An additional factor must be considered. Fanger utilizes empirical relationships to quantify the respiratory energy exchange with the environment. These equations include both sensible and latent heat exchange. In the PTI model both of these effects are considered minimal and are incorporated in the skin surface sensible and latent heat exchange.[8] The effect of increased air velocity will not affect the respired energy transport. Unless this is found to be a small portion of the total sedantary metabolic output, it may introduce a significant difference between the PMV model and the PTI model at the higher air velocities.

In Fig. 4a the h_m X product was plotted as a function of skin temperature. In generating the X - T_s relationship of Fig. 4b, it was assumed that the skin wetness fraction, X, could not exceed unity, as this term was considered to represent only the sweating skin area fraction. For the value X = 1, a corresponding value of h_m was computed. Since the skin wetness parameter, X, did not account for the respiration component, it is evident that the magnitude of the natural motion limit h_m must reflect its influence. The effect was to generate a mass transfer coefficient which is slightly larger than it would have been had the h_m X product contained only the skin evaporative losses.

The effect on h_m can be computed indirectly in terms of Fanger's respiration equation which is given by the following empirically derived relationship:

$$E_{res} = 2.67 \text{ MR } (0.038 - W_a) \text{ (Btu/hr)} \tag{31}$$

where MR and W_a are as defined previously.

Computation of a skin wetness, X, that will be made to exceed unity, for the sake of accounting for the respiratory exchange, is accomplished by combining Eqs. 20 and 31, simplifying and solving for an equivalent term,

$$X_{res} = \frac{2.67 \text{ MR}}{h_m h_{fg} A_{du} f_{pcl}} \tag{32}$$

The approximation made in combining Eqs. 20 and 31 is that the average lung temperature was equal to the average skin temperature. This is certainly reasonable for warm environments since the hypothetical respiratory component of X given by Eq. 20 is actually related to the difference between the skin and ambient humidity but in fact it should be related to the lung to ambient difference. The 0.038 lb_m W.V./lb_m d.a. specific humidity of Eq. 31 corresponds to an average lung temperature of 96°F at saturation. The skin temperature in the warm environments varies only a few degrees from this temperature. Thus, if the respired moisture loss to skin surface moisture loss is small, this approximation will produce very small error.

For the parameters of Eq. 32 defined by the PTI model for a nude subject (i.e., f_{pcl} = 1.0), X_{res} is 0.0367. Since the PTI model has not used an equivalent respiration component in X, we can use this result to adjust the mass transfer coefficient of the h_m X product correspondingly. If X is made to exceed unity it would do so by approximately the 0.0367 factor. Adjusting the PTI natural motion limit mass transfer coefficient by dividing by 1.0367 yields the result of 1.35 $lb_m/hr - ft^2$ (i.e., 1.4/1.0367). Using this value for h_m and the skin wetness fraction, X, as originally defined by the PTI model, yields an h_m X product that is independent of the latent respiratory heat loss.

Defining a new Lewis Relation by the ratio of this h_m to the PTI natural motion limit convective heat transfer coefficient and incorporating Fanger's respiratory equation (32) into the PTI model, gave an indication of the deviation that is involved by combining these subdivided evaporative losses into the h_m X product. The deviations from the previous results of the PTI model ranged from 2% - 5% for the respective air velocities ranging from natural motion limit to about 300 fpm. The reason the deviation increases for increasing air velocity is that when the fraction of moisture loss due to respiration is incorporated into h_m, as in the original PTI development, it is being affected by the air velocity (see Eq. 24). Thus, for increasing air velocity its effect is being over accentuated. It is obviously not as accurate to weigh the respiratory loss by the air velocity as has previously been done in the PTI model.

Fanger's latent respiratory exchange equation, for a given ambient humidity, is constant for all air velocities as it is independent of this parameter. Therefore, for increasing air velocity the magnitude of the deviation between the two models is influenced by the effect. Strictly speaking, it is invalid to weigh the magnitude of the respiratory heat exchange by air velocity, but it invokes very little relative displacement of the PTI line. Since the respiration component accounts for a relatively small amount of the deviation that was seen to occur in the PTI - PMV comparisons for increasing air velocity, it becomes evident that the main reason for deviation is the vapor diffusion issue that has already been discussed.

Furthermore, it was previously noted that Fanger also developed a dry respiratory heat loss expression but since this represents only about half the effect of the respired latent exchange, its effect on the PTI model will not be examined as it has been shown that this latent exchange has a negligible effect.

III. THE NEW EFFECTIVE TEMPERATURE

As early as 1923 the subject of human thermal comfort had become an area of reserach undertaken by the American Society of Heating and Ventilating Engineers (ASHVE; but presently known as the American Society of Heating, Refrigeration and Air Conditioning Engineers - ASHRAE). In that year, F.C. Houghton, et.al.[15], presented a paper at the Society's annual meeting on their determination of lines of equal comfort on a psychrometric chart. The results of their work were based solely upon empirical data. In order to determine the lines of equal comfort, they used two "psychrometric rooms" of varying dry bulb temperature and specific humidity. The atmospheric condition in one of the rooms was maintained at a relatively high dry bulb temperature and low relative humidity, such that judges passing from one room to the other came to the conclusion that the two rooms provided the same sense of comfort. This test procedure yielded a good spread between the two state points on the psychrometric chart and, thus, determined a line of constant comfort. This comfort index became known as the Effective Temperature (ET) Scale.

Previous studies at the University of Santa Clara showed good comparative evidence between the PTI and ET scale at very warm conditions corresponding to a PTI value of about twelve.[8] The overall conclusion, however, was that the ET lines overestimate the influence of humidity on comfort sensations at ordinary temperatures and underestimate the effect at very high temperatures. Yaglou came to the same conclusions twenty-five years after writing the paper. General experience had shown that the ET index was best suited to warm environments when radiation effects were not significant.[16] The PTI model, for those comparisons, had not incorporated radiation effects.

Since the ET index is of an entirely empirical nature as it utilizes the subjective response of human beings, analytical comparisons to the PTI model are not possible. One can merely observe the two indices plotted on a psychrometric chart and note the differences and similarities as has been done previously. Furthermore, since the PTI model requires consideration of radiation effects for the comparisons, any attempted analysis would be even less meaningful as radiation effects were not dealt with in the early work.

Realizing the inadequacy of their first adopted comfort index, ASHRAE sought to develop another more reliable index based in part on empirical work, but on analytical concepts as well. This task was undertaken in 1971 by A.P. Gagge, et.al.,[5] and came to be known as the New Effective Temperature Scale (ET*). This index, derived in much the same manner as the PTI model, can be segmented and analyzed rather carefully. It will be the purpose of this section to analyze the ET* model in both a cumulative and component manner with regard to the PTI model; noting the commonalities as well as the discrepancies.

Figure 14 was generate to indicate the relative agreement that exists between these models in their unaltered form. It is seen that there is good comparative evidence, especially in the neutral thermal comfort zone which is typically the region of greatest practical concern. However, the results deviate increasingly for warmer environments and disagree considerably as to what the location should be for the intolerably warm region.

It is important to observe that the 17.1 PTI line predicted by the PTI model is the upper limit of the environment that allows a human to maintain any thermoregulatory control. This has been established by testing many humans in such environments. It is also seen that the ET* uncomfortably warm region (above PTI=12 for the PTI model) extends to an environment even warmer than that defined by PTI-17.1. This is a region where the PTI model predicts thermoregulatory control will be lost. The evidence indicates that there is serious danger in using the new ET* scale for hot weather occupancy planning of survival shelters.

Contrary to this evidence, previous studies have shown good comparative results between the original effective temperature scale and the PTI lines. From this earlier comparison, effective temperatures were selected (82-85°F) as warm environment limits that were in agreement with the calorimeter studies.

It should be noted that the comparisons of Fig. 14 incorporate clothing whereas the earlier comparisons involved nude subjects. However, the clothing factors do not account for the disagreement between the ET* and PTI evidence.

The radiation effects on the PTI and ET* models of Fig. 14 have been included by assuming that the mean radiant temperature is equal to the ambient temperature. There is a difference, however, between the models with regard to this implementation. The PTI model has used two correction factors, namely the emissivity and effective radiation area factor, that the ET* model has not, even though the use of both paramters is justified. The elimination of these quantities from the PTI model does not, however, rectify the previously cited discrepancies. It merely makes the PTI lines more independent of the environmental humidity (i.e., more vertical). This phenomenon will be discussed more thoroughly in later paragraphs.

PTI VS. ASHRAE'S ET*

SEDENTARY MALE: 0.6 CLO
AIR VELOCITIES:

—— PTI, 0-50 fpm
- - - ET*, 30 fpm

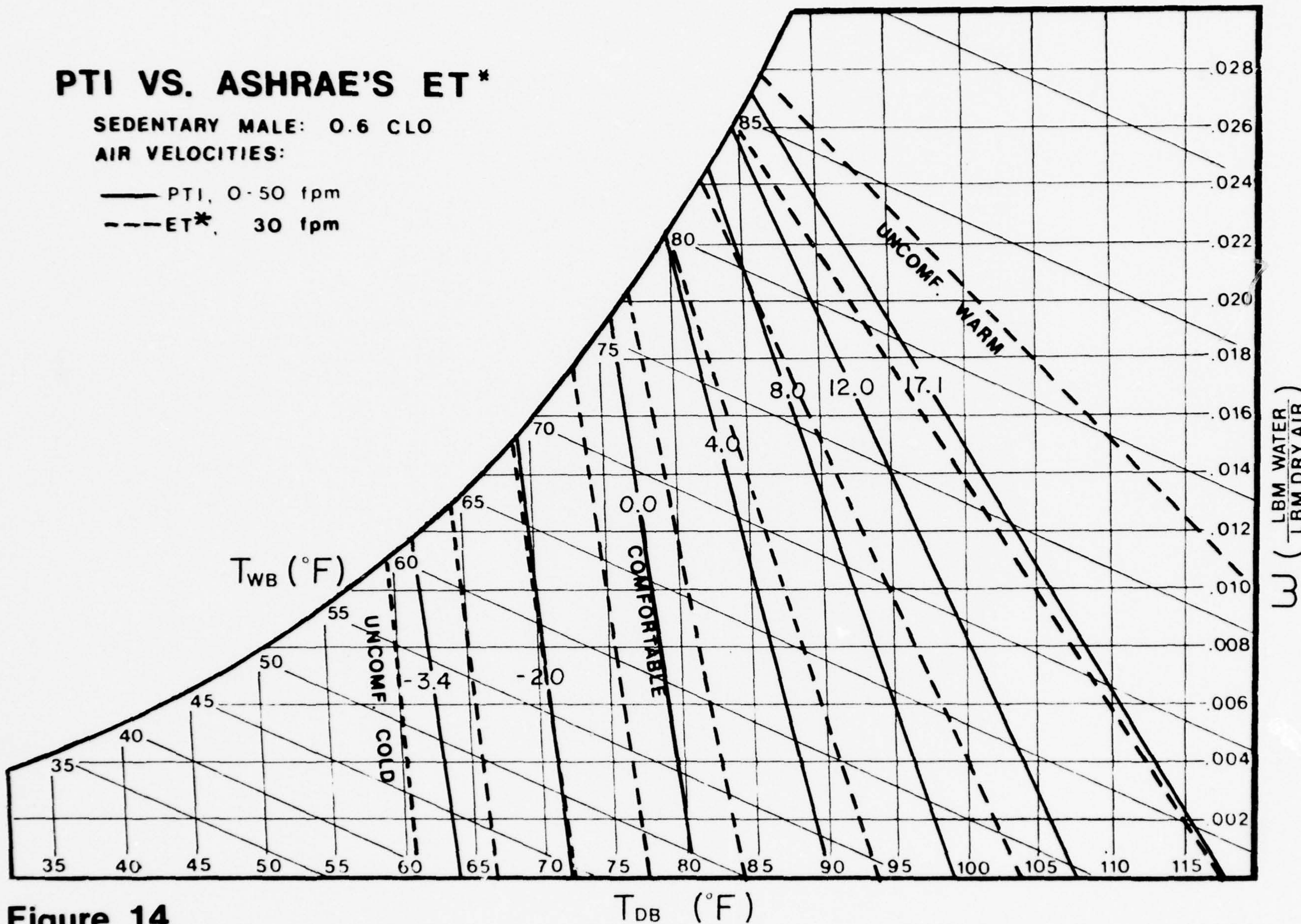


Figure 14

It appears, then, that since the earlier PTI model has given good comparative results to the old TI, effective temperature scale, in the warm-hot region and since these two scales have shown through experience to adequately define this zone, one must suspect that the ET* scale does not correctly define the warm and hot environments. This conclusion arises from the fact that the only consequential difference between the "earlier" PTI model and the PTI model discussed here are the two effects mentioned: clothing and radiation. Both have been compensated for in a justifiable manner. Even if Gagge were to argue the disuse of the additional factors discussed, this does not account for the extreme displacement of the warmer regions of Fig. 14.

In the event a person is interested in the neutral thermal comfort region either the PTI or ET* models can be used as the results are in good agreement. The subsequent paragraphs of this section will be devoted to reconciling the deviations that occur in the other comfort regions and in the two models in general.

III.1 Internal Body Resistance

The first analysis to be conducted on the ET* scale is to relate and compare the model's equivalent internal thermal body resistance to that given by Eq. 2 of the PTI model. The variable internal body resistance term, R", of the PTI model is not referred to nor computed as such in the ET(model but it can, however, be backed out equivalently using two of the energy balance equations. The first equation characterizes the net heat flow to and from the skin shell (the ET* skin shell being defined analogously to the one of the PTI model) is given by the following relation:

$$S_{sk} = K_{min} (T_{ce} - T_{sk}) + C_{bl} V_{bl} (T_{cr} - T_{sk}) - E_{sk} - (R + C) \qquad (33)$$

where S_{sk} is the rate of skin shell heat storage (Btu/hr - ft^2)

C_{bl} is the specific heat of blood (Btu/lbm - °F)

V_{bl} is the rate of skin blook flow (lbm/hr - ft^2)

K_{min} is the minimum heat conductance of skin tissue (Btu/hr - ft^2 - °F)

E_{sk} is the total skin evaporative loss (active sweating and moisture diffusion)

R + C is the sensible heat loss

The net heat flow to and from the core is given by:

$$S_{cr} = (M-E_{res}-W) - K_{min}(T_{cr}-T_{sk}) - C_{bl}V_{bl}(T_{cr}-T_{sk}) \qquad (34)$$

where S_{cr} is the rate of core heat storage (Btu/hr - ft^2)

M and W are analogous to the terms of the PTI model Eq. 7

E_{res} is the latent respiratory heat loss (Note: ET* uses Fanger's equation discussed in the previous section)

Considering only equilibrium conditions, which was the case in the derivation of the body resistance term of the PTI model given by Eq. 2, the terms S_{sk} and S_{cr} are equal to zero. Combining equations (33) and (34) of the ET* model and rearranging yields the following relationship:

$$\frac{2(T_{cr} - T_{sk})}{M - E_{res} + E_{sk} + (R + C)} = \frac{1}{K_{min} + C_{bl}\ V_{bl}} \qquad (35)$$

Since S = M - E - R - C - W of the ET* model (identical to Eq. 7 of the PTI model) where $E = E_{res} + E_{sk}$ and since the storage and work terms are equal to zero in this consideration, this expression simplifies to the following:

$$\frac{T_{cr} - T_{sk}}{M - E_{res}} = \frac{1}{K_{min}\ C_{bl}\ V_{bl}} \qquad (36)$$

Had the PTI model concerned itself with generating a relationship for the latent respiratory heat loss, E_{res}, (recall that this term is lumped into the h_m X product) the expression on the left hand side of Eq. 36 would have been identical to that given by Eq. 2 of the PTI model. Utilizing Eq. 6 of the PTI model and Fanger's respiratory loss expression given by Eq. 31, to determine the influence of the latter term on the PTI resistance given by Eq. 3, revealed a maximum deviation of about 8% over the entire relative humidity range. This deviation indicates the difference between using the PTI thermal body resistance with and without the E_{res} term.

In order to compare the magnitude of the ET* body resistance in terms of the PTI model, a relationship of T_{cr} to T_{sk} must be extracted from the ET* model. This will generate an equation in the same form as Eq. 3 of the PTI model which can then be plotted as a function of skin temperature, thus a comparison can be made with Fig. (2) of the PTI model.

To this end, Eq. 36 can be rearranged to give the desired relationship. From reference 5 the terms K_{min} and C_{bl} are constants equal to .93 Btu/hr - ft^2 - °F and 1.00 Btu/lb$_m$ - °F, respectively, and V_{bl} is given by the following relationship when T_{cr} is less than 97.9 F and T_{sk} is less than 93.4 F:

$$V_{bl} = \frac{1.3}{1 + 0.28\ (93.4 - T_{sk})} \quad (lb_m/hr - ft^2) \qquad (37)$$

When T_{cr} is greater than 97.9 F and T_{sk} is greater than 93.4 F, it is given by the following relationship:

$$V_{bl} = 1.3 + 8.53\ (T_{cr} - 97.9)\ (lb_m/hr - ft^2) \qquad (38)$$

In addition, two other expressions for V_{bl} are possible in the ET* model for the situations where T_{sk} is greater than 93.4 °F and T_{cr} is less than 97.9°F and where T_{sk} is less than 93.4 °F and T_{cr} is greater than 97.9 °F. However, these cases represent transient response conditions and are, therefore, not considered here.

Substituting Eq. 37 into Eq. 36 and solving for T_{cr} as a function of T_{sk}, generates the following relationship:

$$T_{cr} = T_{sk} + \frac{(M - E_{res})\ (1 + 0.28\ (93.4 - T_{sk}))}{(1 + 0.28\ (93.4 - T_{sk}))\ K_{min} + 1.3\ C_{bl}} \quad (°F) \qquad (39)$$

for T_{sk} less than 93.4 F.

Similarly, substituting equation (38) into equation (36), but solving for T_{sk} as a function of T_{cr}, yields:

$$T_{sk} = \frac{8.53\ C_{bl}\ T_{cr}^{\ 2} + (K_{min} - 833.8)\ T_{cr} + E_{res} - M}{8.53\ C_{bl}\ T_{cr} + K_{min} - 833.8\ C_{bl}} \quad (°F) \qquad (40)$$

for T_{cr} greater than 97.9 F.

Substituting the values for M equal to 18.5 Btu/ft^2 - hr given by the ET* model for a sedentary male, the constants K_{min} and C_{bl} and Eq. 32 for E_{res}, into equations (39) and (40) generates a relationship between T_s and T_{cr} with W_a as a parameter. Considering the entire humidity range specified by W_a, Eq. 40 predicts to within less than one half of one degree (°F) the relationship between T_{sk} and T_{cr} as compared to equation (4) which is recommended for use by the PTI model. This difference deviates from about a fifth of a degree to about two fifths of a degree over the entire humidity range, indicating once again the relatively insignificant effect of the latent respiratory exchange, E_{res}. The fact that Eq. 40 was derived independently of Eq. 4 and agrees with the results of the latter expression to within an acceptable tolearance, gives rise to the probability that these are indeed accurate expressions. This also tends to assist the probability that the skin-core temperature relation given by Eq. 6 of the PTI model is in error, as it deviates by as much as a degree (°F) from these expressions. It should be noted that the constant comfort lines predicted by the PTI model in Section 1 are in no way dependent on the skin-core temperature relationship; they only become dependent when analyzing transient response conditions.

Having stated that equation (40) gives reasonable results, we can now analyze equation (39). This equation corresponds to those conditions that are represented in the PTI model by values of the PTI Index less than 0 (i.e., neutral thermal comfort to cold conditions). Over the skin temperature range 90.0 °F to 93.4 °F the core temperature predicted by equation (39) ranges from 100 °F to 102 °F, again considering the entire humidity range. This result is in serious error. Over the same range of skin temperatures Eq. 4 predicts core temperatures ranging from 96.4 to 97.5 °F. When considering the precision with which the body can maintain its internal temperature, the five degree discrep ncy indicated here is enormous. It must be remembered,too, that the temperatures predicted here are core temperatures which are weighed by the lower skin temperature, therefore, Eq. 4 would predict a rectal temperature in the neighborhood of 105 °F based on the skin-core temperature relation of equation (39). The oral temperature is normally approximated as one degree less than the rectal temperature, thus, equation (39) of the ET* model predicts an oral temperature of roughly 104 °F for what are supposed to be conditions ranging from cold to those characterized as neutral thermal comfort.

When equations 39 and 40 are alternately substituted into Eq. 36, two expressions for the internal thermal body resistance are generated that, combined, cover the entire comfort spectrum. The body resistance can then be plotted as a function of the skin temperature as had been done for the PTI model, Fig. 2. This resistance is in error over the cold to neutral comfort range, reflecting the inaccurate relationship of Eq. 39. Nevertheless, the entire range of resistances were plotted against skin temperature for the sake of further indicating the erroneous result in this region. The results are shown in Figs. 15 and 16. Figure 15 uses metabolic rate per unit area equal to 20.0 Btu/ft^2 - hr corresponding to the value suggested by the PTI model. However, Fig. 16 uses a common metabolic rate per unit area for all three curves. This value corresponds to 18.5 Btu/ft^2 - hr which, as previously mentioned, corresponds to the value recommended by both Fanger and the ET* model. Again, as previously mentioned, the PTI curve was adjusted using this value (Fig. 16) to eliminate obvious sources of disagreement even though it appears that the former value is more accurate as indicated in the previously postulated argument.

Figure 15 was presented here to show that good comparative results do not necessarily mean that the parameters used to generate the curves are in agreement. It is seen in Fig. 16 that when a common metabolic rate per unit body area is used the correlation of the curves of the two models is not as good. The remaining deviation, again neglecting the region for T_s less than 93.4 °F, is indicative of the different results generated from Eq. 6 of the PTI model and Eq. 40 of the ET* model. These equations are the respective models' relationship between the skin and core temperature, which are the only remaining parameters of the internal thermal body resistance relation (left hand side of Eq. 36)to be contended with as the metabolic rate and respiratory term have already been discussed and reconciled. The thermal resistance generated from the relationship given by Eq. 4 is shown in Fig. 16. Taking the E_{res} term to be identically zero, the resulting variation in R" with Ts allows determination of T_{core} vs T_{skin}, see Fig. 17. The other T_{core}, T_{skin} relationships are directly compared in the figure. Equation 4 is the skin-core temperature relation originally recommended for use by the PTI model and Eq. 6 is the skin-core relation deduced from

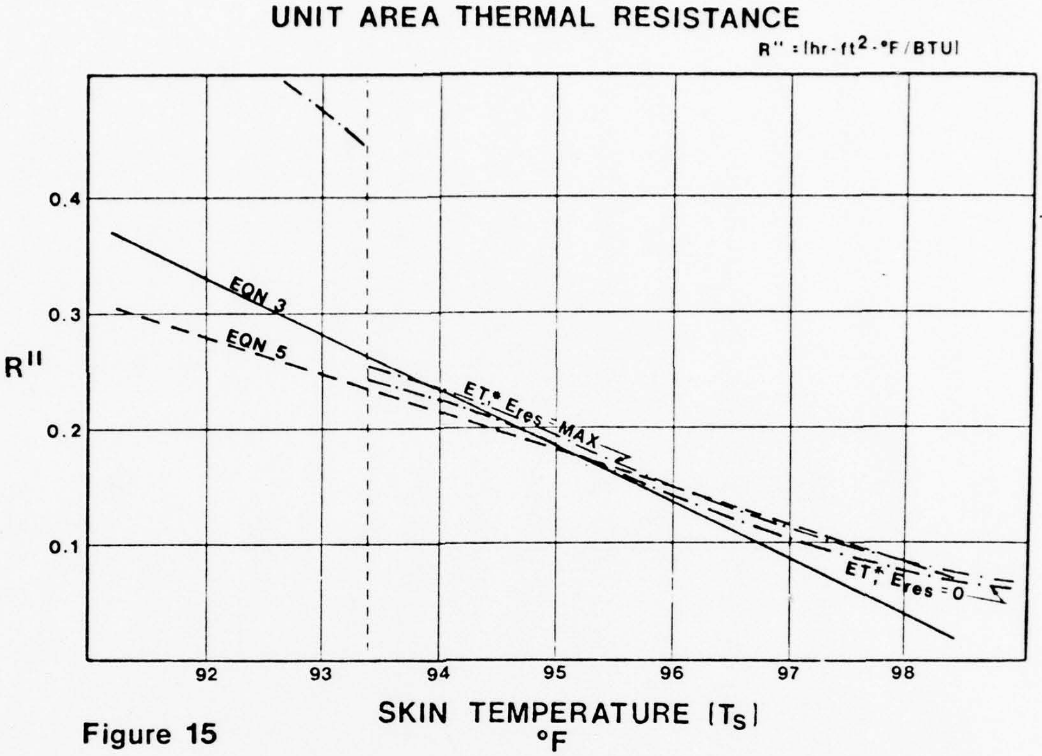


Figure 15

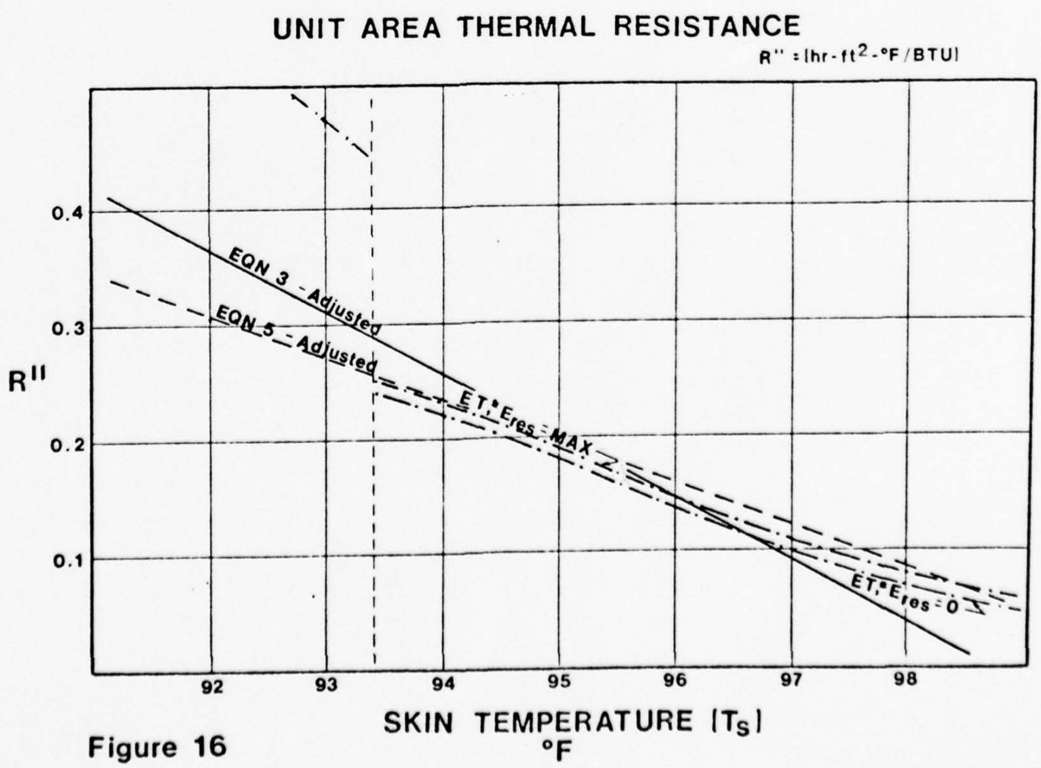


Figure 16

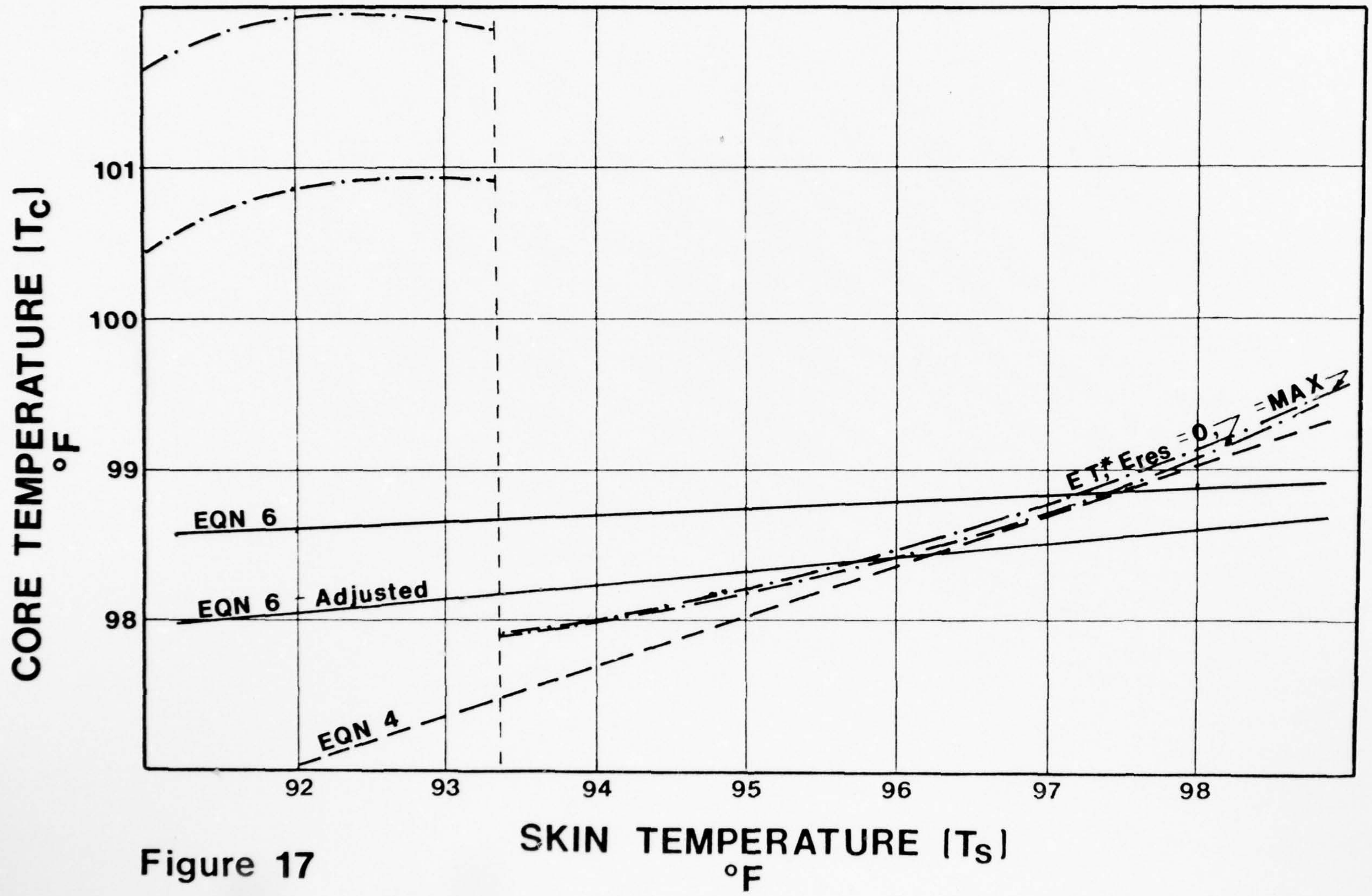

CORE TEMPERATURE [Tc]
°F
101
100
99
98
92
93
94
95
96
97
98
SKIN TEMPERATURE [Ts]
°F
EQN 6
EQN 6 - Adjusted
EQN 4
ET; Eres = 0, Z = MAX Z

Figure 17

the empirically derived PTI thermal body resistance expression given by Eq. 3.

It is seen in Fig. 17 that Eq. 6 of the PTI model does not correlate very well with the other relationships. This expression was derived from the thermal body resistance of the PTI model given by Eq. 3. It is again noted, as stated in Section I, that this equation may be inaccurate.

The left hand side of the neutral thermal comfort line (T_s less than 93.4°F) shows the absurd results of the ET* model predicted by Eq. 39. However, for T_s greater than 93.4°F it is observed that equations 4 and 40 correlate reasonably well. Of the relationships depicted here Eq. 40 is the only one to satisfy the neutral thernal comfort state point where T_s equals 93.4°F and T_{cr} equals 97.9°F.

This condition for neutral comfort was prescribed as neutral comfort in earlier PTI studies yet using the composite R" derived by least squares fit of a large number of test subjects does not allow the point to be reconstituted. This creates, as mentioned previously, uncertainty about R".

Since Eq. 40 satisfies this neutral comfort condition, is reasonably close to Eq. 4 (Fig. 17), it appears to represent the most accurate relationship between the skin and core temperture. For skin temperatures less than 93.4°F the ET* model involves Eq. 39 and produces completely erroneous evidence.

It is important to emphasize that these relations in no way affect the results for steady state conditions. Having specified an average body surface skin temperature, the core temperature need not be given nor predicted to generate the constant comfort lines under these conditions. This was pointed out earlier in the thermal circuit analysis.

It is again suggested, as in Section I, that additional research is needed to provide a more satisfactory skin-core temperature relation in the PTI model and in the ET* model.

III.2 External Heat Rejection

The method of handling external heat rejection at steady state conditions is essentially the same in the ET* model and the PTI model. Unlike Fanger, the ET* model uses a mass transfer coefficient to weigh the magnitude of the water vapor loss by diffusion. It does, however, use the respiratory exchange equation as derived by Fanger. Since this exchange mechanism has been thoroughly analyzed in the previous section, it will not be explicitly discussed any further.

In spite of the good comparative evidence of Fig. 14 for the neutral thermal comfort region, there are aspects of considerable disagreement between the two models. These appear primarily in the magnitudes of some of the parameters that are used in the modeling equations. Additional analysis has exposed underlying discrepancies that appear to have compensated for one another in the region of neutral thermal comfort thus, generating final results that are in relatively good agreement. The major discrepancies that will subsequently be discussed and analyzed concern the magnitudes of the heat and mass transfers coefficients that should be used and the shape of the effective $X\text{-}T_s$ relationship of the two models.

III.3 Heat and Mass Transfer Coefficients:

The ET* scale developed by Gagge only considered one air velocity; 30 fpm. In 1974, however, Nevins et. al.[17] incorporated the effects of other air velocities into the model originally developed. Since it will be seen that these coefficients are a major source of disagreement in the models, only the single air velocity of the earlier model will be analyzed to determine the effects of the discrepancy. Analysis of additional air velocities is less meaningful once this discrepancy has been exposed.

The convective heat transfer coefficient given by the ET* model, corresponding to the air velocity of 30 fpm, is 0.512 Btu/ft^2 -hr- ^{o}F. The recommended Lewis Relation is 4.43 lb$_m$ -^{o}F/Btu. These values are considerably different from those used in the PTI model for the same conditions; 0.765 and 1.83, respectively. To determine the effect of the ET* coefficients on the PTI results, Fig. 18 was generated. In this figure, the PTI model contains the ET* coefficients and has also been adjusted for the other parameters mentioned at the beginning of the section which have lesser effects. If all other factors in the models are the same, the two sets of curves should superimpose and comfort levels should agree.

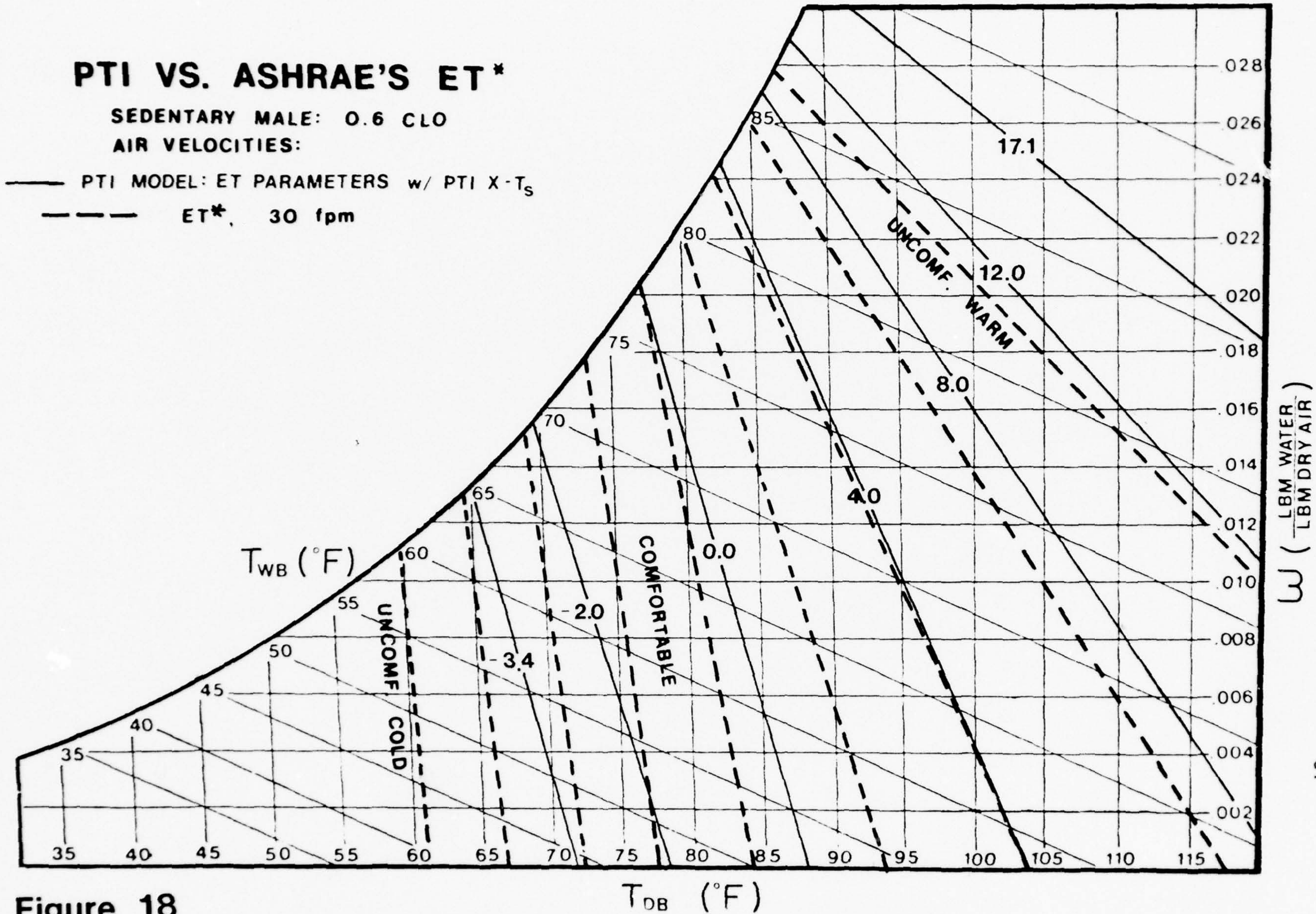

PTI VS. ASHRAE'S ET*
SEDENTARY MALE: 0.6 CLO
AIR VELOCITIES:
PTI MODEL: ET PARAMETERS w/ PTI X - T_S
ET*, 30 fpm
17.1
12.0
8.0
4.0
0.0
-2.0
-3.4
UNCOMF. WARM
COMFORTABLE
UNCOMF. COLD
T_{WB} (°F)
T_{DB} (°F)
ω (LBM WATER / LBM DRY AIR)
.028
.026
.024
.022
.020
.018
.016
.014
.012
.010
.008
.006
.004
.002
35
40
45
50
55
60
65
70
75
80
85
90
95
100
105
110
115

Figure 18

It was noted earlier that although good comparative evidence in the neutral comfort range exists in Fig. 14, there was poorer agreement near the intolerable limits; both hot and cold. It is now seen by comparing Fig. 18 with Fig. 7 that the effect of the ET* coefficients has moved the comfort lines of the PTI model to a considerably warmer environment for any given state of comfort and has also generated a much greater degree of humidity dependence. This shift in comfort is partially due to the large Lewis Relation of the ET* model. This factor is almost two and a half times that used in the PTI model and coupled with a slightly lower convective heat transfer coefficient generates a much greater degree of humidity dependence.

The Lewis Relationship used in the ET* calculations and carried into the PTI model forecasts that a human with 0.6 clo and in a still air environment will feel only uncomfortably warm at effective temperatures from the old ET scale of 85 - 90 °F. Experiments with the human calorimeter have shown that people in the semi-nude state are approaching loss of thermo regulatory control for these environments. Thus, it is believed that the ET* scale and its Lewis Relationship must be examined further before it is considered acceptable for use in the region of high temperature and humidity and low ventilation rates.

Another factor of disagreement between the ET* and PTI scales must be due to the X - T_S relationships of the respective models as all other parameters have been made identical in the comparison shown in Fig. 18. Note that the effective radiant area factor, the area ratio, and the emissivity, that were discussed previously, have been adjusted in the PTI model to be in accordance with the ET* model. This was done only to distinguish the effect of the remaining source of disagreement and not because their disuse is considered justified. In addition, the latent respiratory exchange equation of Fanger which was used in the ET* model was employed for the same reason. A full discussion of these parameters is delayed until later paragraphs.

The ET* model does not describe an $X - T_s$ relation as does the PTI model, but through manipulation of some of the exchange equations of this model it can be generated. The task of the next section will, therefore, be to produce this relationship. When this $X - T_s$ relationship is superimposed on the relationship used in the PTI model as given in Fig. 5, the results should be identical.

III.4 $X - T_s$ Relationship

In the text of Gagge's development of the ET* model, a graphical relationship of skin temperature and regulatory skin wetness was presented. This relationship provides the basis of the ET* equivalent $X - T_s$ curve shown in comparison to the PTI curve in Fig. 19. In addition to the regulatory skin wetness, which is characterized by sweat secretion, the effects of vapor diffusion has also been accounted for in Fig. 19. (The respiratory exchange equation has not been considered in developing this curve since, as shown in Section II, the PTI skin wetness, X, is independent of this quantity and it is the basis for comparison here.)

The ET* evaporative exchange equation, combining the effects of water vapor diffusion and actively secreted sweat, is given by the following relationship:

$$E = L.R.\ h_c\ h_{fg}\ (0.06 + 0.94\ W_{rsw})\ (W_s - W_a)\ f_{pcl}\ (\text{Btu/ft}^2 - \text{hr}) \tag{41}$$

where W_{rsw} is the ratio of the actual energy exchange due to evaporated sweat to the maximum possible. All other parameters are as defined previously.

The evaporative loss exchange equation of the PTI model is taken from Eq. 21:

$$E = L.R.\ h_c\ h_{fg}\ X\ (W_s - W_a)\ f_{pcl} \tag{42}$$

Note that $h_m = L.R.\ h_c$. Then by equating 41 and 42 and solving for the skin wetness, X:

$$X = 0.06 + 0.94\ W_{rsw} \tag{43}$$

where W_{rsw} is the term that is presented graphically in the ET* model.

The only parameter of this equation that is a function of the skin temperature is W_{rsw}. Thus, it is possible to generate the $X - T_s$ relationship of the ET* model.

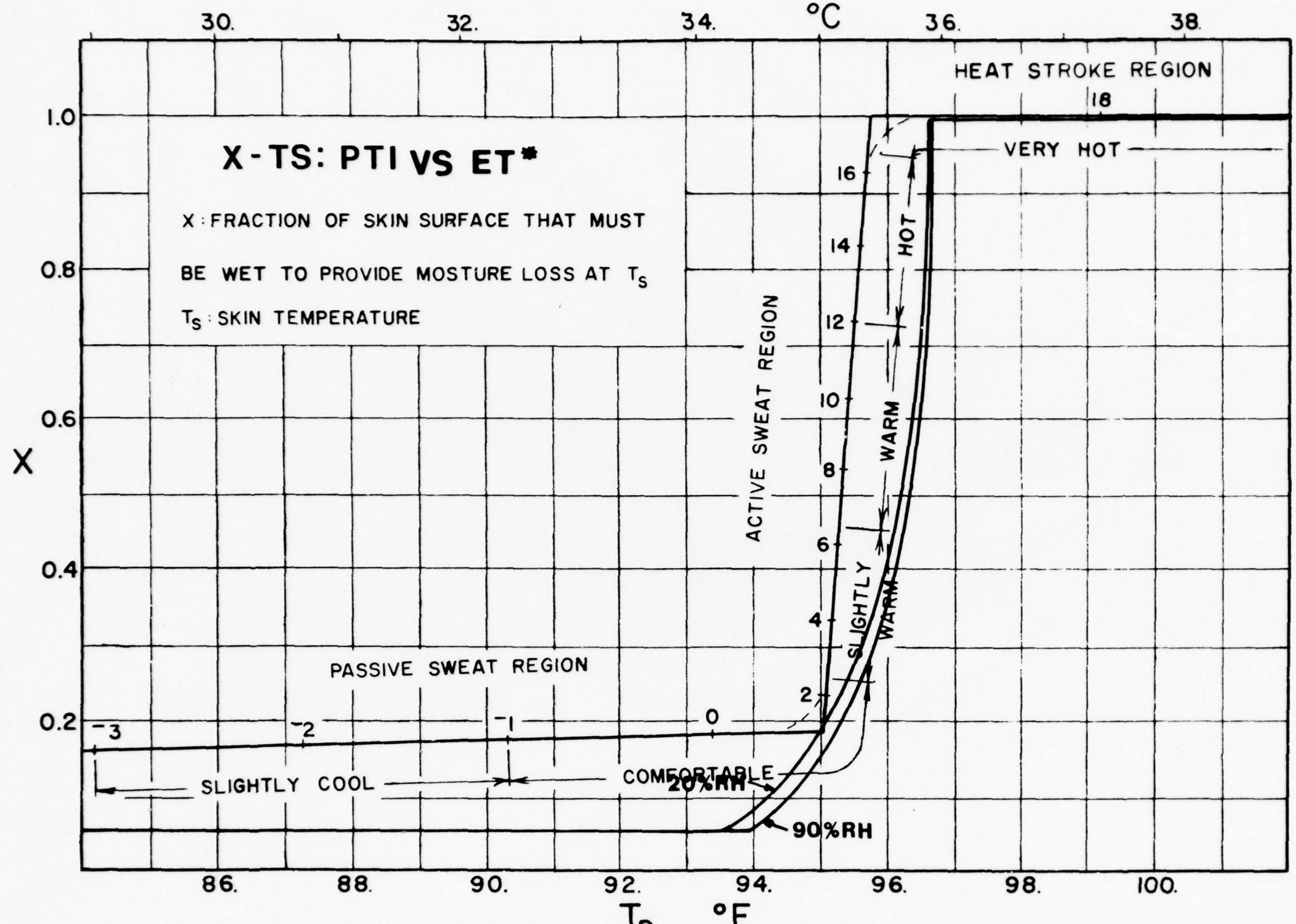


Figure 19 BODY EVAPORATIVE BEHAVIOR V.S. SKIN TEMPERATURE

The vapor diffusion component of X equal to 0.06 has been added to the W_{rsw} - T_s relationship of the ET* model to generate the comparative X - T_s curve shown in Fig. 19. It is noted that this relationship now contains the same two evaporative loss mechanisms as the X - T_s curve of the PTI model shown in the same figure.

It is further noted that the regulatory skin wetness, W_{rsw}, is a function of the humidity difference by virtue of its definition. For this reason the two curves, representing different humidities, are shown in the transition from the passive sweat region (where X of the ET* model is only a function of the constant diffusion term as W_{rsw} is zero in this region) to the active sweat region where X becomes a function of both terms in equation (43). This may be a justified relationship, but since the X - T_s relationship of the PTI model was based on a curve fit of experimental data, a normal data scatter would override the possibility of generating more than one relationship in such close proximity. In any event the two curves of the ET* model, which essentially represent the entire humidity range, are sufficiently close to one another such that one can use their average value (corresponding to RH = 50%) without inducing significant error. This will become evident when this curve is subsequently implemented in the PTI model for comparative purposes.

Comparing the curves of the two models in Fig. 19, reveals a striking similarity in their overall trend. The essential discrepancy arises only from a relatively constant displacement of one curve from the other. Moreover, the shape of the PTI curve has generally been accepted as valid and since the ET* curve also indicates the same relative trend, it cannot be refuted on this basis. The resolution of their differences will most likely come only through additional research.

The relative displacement of the curves in the passive sweat region tends to compensate for the disagreement in the mass transfer coefficients employed by the two models, thus generating $h_m X$ products for the two models in the neutral thermal comfort region that are very close in magnitude. The displacement in the active sweat region, however, compounds the effect of the disagreement in the mass transfer coefficients. As both curves approach unity it is inevitable that the larger mass transfer coefficient of the ET* model affects its results in such a way as to displace the predicted upper limit of survival environments to higher temperatures and humidities when compared to those predicted by the PTI model. In addition, the ET* skin wetness does not equal unity until almost a full degree (°F) higher in skin temperature as compared to the PTI model.

In order to more adequately visualize the effect of the $h_m X$ product, Fig. 20 was generated to indicate its trend as a function of skin temperature. The $h_m X$ product is the effective evaporative exchange coefficient and, therefore, gives more meaningful insight to a major source of discrepency in these models.

The $h_m X$ product of the PTI model incorporates a weighing factor for the latent respiratory heat loss (contrary to the ET* model) as shown in Section II. It was not adjusted to dissociate itself from this exchange mechanism for representation in Fig. 20. Since the $h_m X$ product used to generate the PTI results of Fig. 14 contained a quantity to reflect this mechanism, the comparison of the $h_m X$ products in Fig. 20 were chosen to be indicative of this quantity in its unaltered form. In doing so the deviations of Fig. 14 can be directly reconciled. The procedure used to generage Fig. 20 was to use the mass transfer coefficients and $X - T_s$ relationships of the respective models.

The largest value that the skin wetness fraction, X, can attain is unity, so that the $h_m X$ product under these circumstances is merely an indication of the discrepency in the mass transfer coefficients. It is clear from the figure that the deviation in $h_m X$ continually gets worse as T_s increases until X finally reaches unity. As stated previously, this indicates the

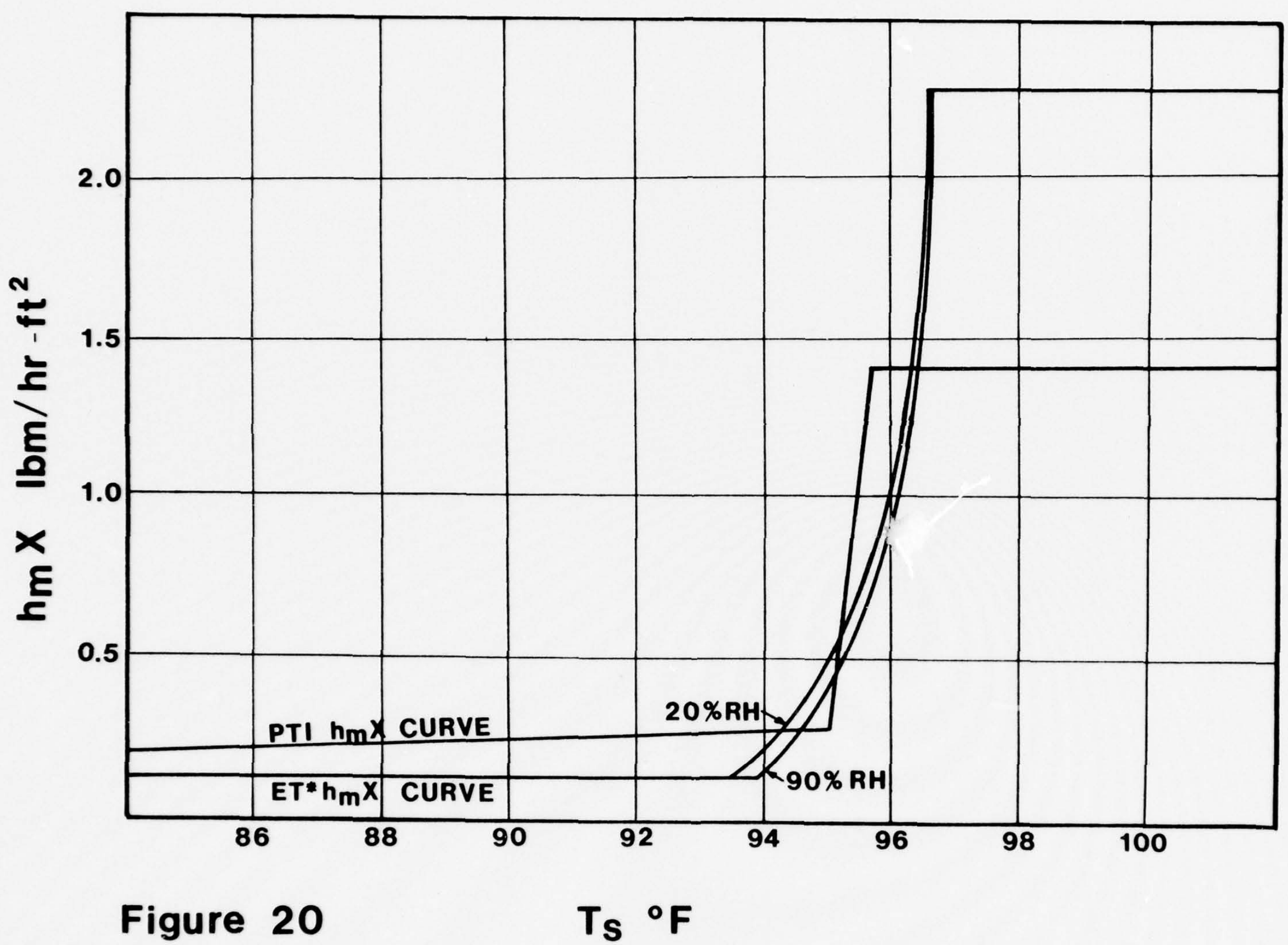

2.0
1.5
1.0
0.5
$h_m X$ lbm/hr-ft^2
PTI $h_m X$ CURVE
ET* $h_m X$ CURVE
20%RH
90%RH
86
88
90
92
94
96
98
100
T_S °F

Figure 20

reason that the upper limit environments of the ET* model differ radically from those of the PTI model. Again note that in the neutral thermal comfort region the h_m X products are virtually identical. Coupling this agreement with the fact that the discrepency in heat transfer coefficients is compensated primarily by the ET* model's disuse of the effective radiant area factor (two other factors were mentioned but they have lesser effects), results in the generation of the comparative results in the neutral thermal comfort region as shown in Fig. 14. It should be noted that these compensations are purely coincidental. The relative displacement in the cooler region of the lines of equal comfort is accounted for by the convective and radiative factor differences between the two models. As seen in Fig. 14 they do not diverge significantly as the evaporative exchange is less significant in this region.

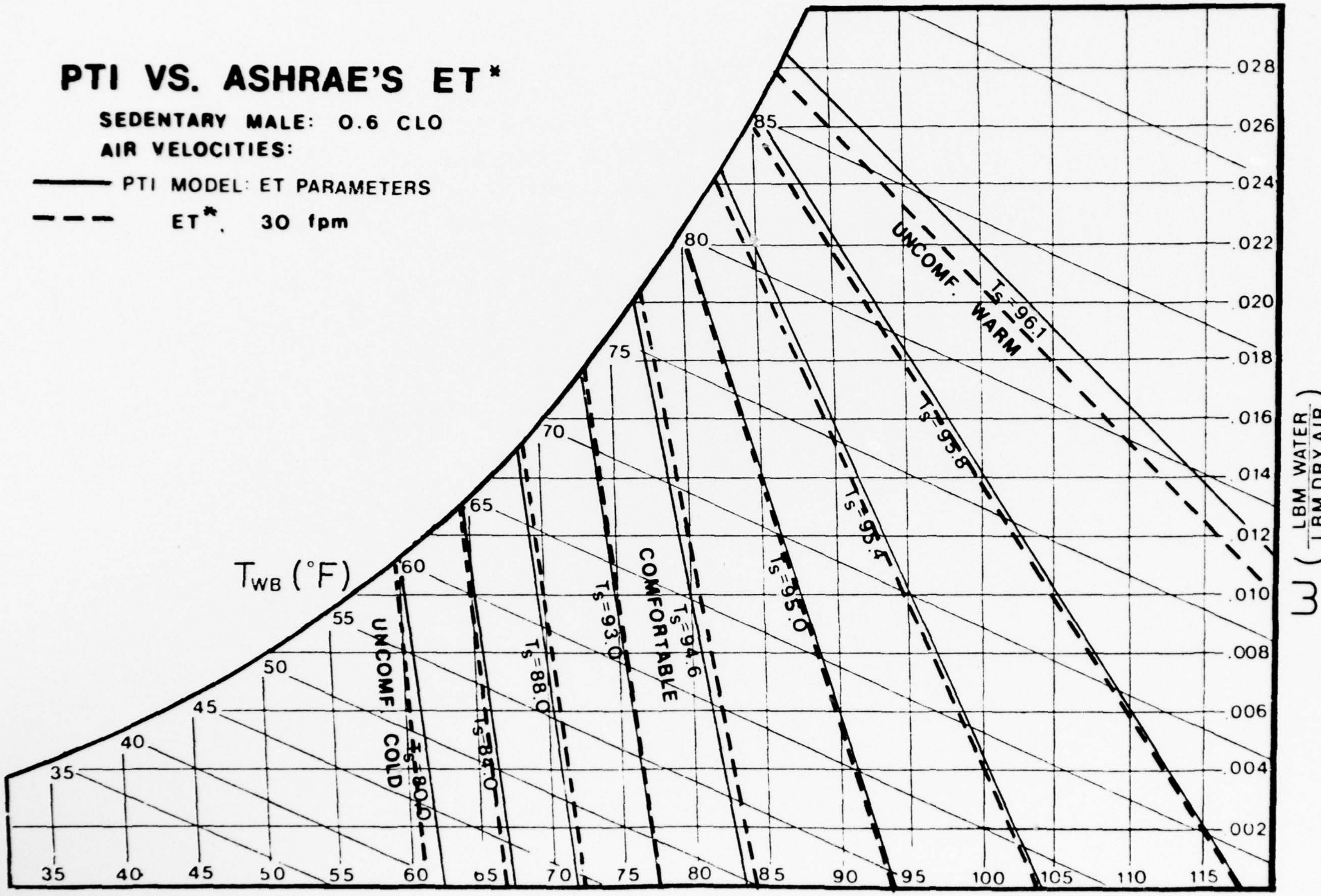
The remainder of this section will incorporate into the PTI model all of the ET* parameters that have been so far discussed. The resulting superimposition of the constant comfort lines thus generated, onto those of the ET* model, will give evidence to the fact that the only discrepencies in the models are those that have been mentioned.

III.5 <u>The Effect of the ET* Parameters on the PTI Model</u>: Toward the end of Section II a method was devised for reconciling the influence of Fanger's latent respiratory exchange equation on the h_m X product of the PTI model. However, since the ET* model has not developed the mass transfer coefficient such that it would reflect the respiratory loss, the technique employed there will not be needed in the present analysis. Implementing the respiratory effect was achieved by merely incorporating Eq. 31 directly into the PTI model. In addition to using the ET* heat and mass transfer coefficients and X - T_s relationship, there are other parameters of the PTI model that have to be adjusted to correspond to the values used in the ET* model. These will now be discussed briefly.

Of this list of parameters the clothing conduction concept of the ET* model which employed a thermal efficiency factor to account for clothing effects, has already been discussed. This parameter is given by Eq. 11. In addition, the ET* metabolic rate per unit body area has also been discussed. The other terms that need be adjusted are the linear radiant heat transfer coefficient, h_r, the effective radiation area factor, f_{eff}, and the emissivity of the outer surface of the body.

The radiant heat transfer coefficient was assumed constant in the ET* model whereas it was refined, as mentioned earlier, for every comfort line generated by the PTI model. Furthermore, the emissivity recommended for use by Fanger (0.97) was employed in the PTI model. In the ET* model this term was neglected and, therefore, effectively taken as unity. These two points are essentially of negligible concern, but are mentioned for the sake of completeness. In addition, Fanger has recommended an effective radiation area factor of 0.696 for the sedentary state which was used in the PTI model. The ET* model has also neglected the use of this parameter, as mentioned in the previous section. This factor does affect the results of PTI model significantly and since considerable evidence exists to justify its use, its omission from the ET* model is viewed as an oversight. However, since the ET* model has not considered its effect, it was taken as unity in the comparison to follow.

Figure 21 is the ultimate comparative result of having incorporated all of the parameters of the ET* model into the PTI model. The excellent comparative evidence of that figure indicates proper accounting for all of the ET* parameters. This being the case, one can use the previous analyses of the parameters as truly indicating the sources of discrepancies that exist between these models, and, therefore, take action toward resolving them.



PTI VS. ASHRAE'S ET*
SEDENTARY MALE: 0.6 CLO
AIR VELOCITIES:
PTI MODEL: ET PARAMETERS
ET*, 30 fpm
UNCOMF. WARM
COMFORTABLE
UNCOMF. COLD
Ts=96.1
Ts=95.8
Ts=95.4
Ts=95.0
Ts=94.6
Ts=93.0
Ts=88.0
Ts=84.0
Ts=80.0
TWB (°F)
TDB (°F)
ω (LBM WATER / LBM DRY AIR)
35
40
45
50
55
60
65
70
75
80
85
90
95
100
105
110
115
.002
.004
.006
.008
.010
.012
.014
.016
.018
.020
.022
.024
.026
.028

Figure 21

IV. CONCLUSIONS

Since each of the previous sections contains analyses and draws contrasts between the various models, it is the purpose of this section to summarize the noted differences in the order of their importance and indicate steps to resolve discrepancies.

1. The displacement of ET* warm-hot comfort lines to higher temperatures and humidities than recommended by PTI lines of equal comfort are cause for serious concern if the ET* model identified in the current ASHRAE literature is incorporated in the DCPA literature. There are also differences in the cool-cold comfort lines but these can be easily compensated by changes in clothing.
2. The skin-core temperature relation of the ET* model for skin temperatures less than 93.4 °F is obviously in error and, therefore, in need of additional analysis.
3. The clothing conduction concept employed in the ET* model should be reconsidered by ASHRAE relative to the effects of the area ratio factor, f_{cl}.
4. Sufficient conflicting evidence exists to justify additional research on the skin-core temperature relation of the PTI model as derived from the model's internal body resistance expression.
5. The PTI natural motion limit heat transfer coefficient may deserve further study as it corresponds to a relatively high air velocity according to Kerslake.
6. The skin vapor diffusion controversy should be resolved by proving that its effect should be either considered constant or weighed by a mass transfer coefficient which is sensitive to air motion. The latter method appears to be the correct handling procedure at present.
7. Since the models developed by the researchers in this field of study do not generate results that are sufficiently similar, one need not, as yet, be overly concerned with the added "precision" by the exchange mechanisms of the respiratory tract.
8. The implementation of the effective radiant area factor should be reconsidered by the authors of the ASHRAE model as it appears to be justified.

It appears evident from this study that DCPA literature should not be revised to embrace the ASHRAE-ET* scale until the aforementioned issues have been resolved.

APPENDIX

The following computer program predicts the set of average air states that will produce a given comfort sensation when specifying the air velocity and clothing. In addition, the sensible and evaporative heat outputs of the human are computed for each air state of the comfort line. For a specified air flow rate the inlet and exit air states can also be computed.

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SYMBOLS USED IN PROGRAM

ADU	body surface area (ft^2)
CP	specific heat at constant pressure for dry air (BTU/lbm-°F)
DA	average air stream density (lbm/ft^3)
DI	initial guess at air stream density (lbm/ft^3)
EPS	average emissivity of skin and clothing (N.D.)
FCL	ratio of the clothed body area to the nude body area (N.D.)
FEFF	effective ratiation area factor (N.D.)
FPCL	moisture permeation efficiency factor for clothing (N.D.)
G	mass flow rate of air (lbm/hr)
HC	convective heat transfer coefficient (BTU/ft^2-hr-°F)
HFG	latent heat of vaporization of water at 95 °F (BTU/lbm)
HI	initial guess at linear radiant heat transfer coefficient (BTU/ft^2-hr-°F)
HM	mass transfer coefficient ($lbm/hr\text{-}ft^2$)
HR	linear radiant heat transfer coefficient (BTU/ft^2-hr-°F)
ICLO	insulation factor for clothing (clo)
LH	latent heat transfer (BTU/hr)
MR	sedentary male metabolic rate (BTU/hr)
P	atmospheric pressure (lbf/ft^2)
PVS	vapor pressure of saturated water at the skin temperature (lbf/ft^2)
QI	volumetric air flow rate at inlet conditions (ft^3/hr)
R	radiant heat transfer (BTU/hr)
RA	gas constant for dry air (ft-lbf/lbm-°F)
RW	gas constant for water vapor (ft-lbf/lbm-°F)
SH	sensible heat transfer (BTU/hr)
TA	logarithmic mean air temperature (°F)
TCL	outer clothing surface temperature (°F)
TI	inlet air temperature (°F)
TO	exhaust air temperature (°F)
TS	average skin temperature (°F)

SIGMA	Stefan-Boltzmann constant: 0.1714×10^{-8} (BTU/ft^2-hr-°R^4)
V	air valocity (ft/min)
WA	logrithmic mean humidity ratio (lbm W.V./lbm D.A.)
WI	inlet humidity ratio (lbm W.V./lbm D.A.)
WO	exhaust humidity ratio (lbm W.V./lbm D.A.)
WS	humidity ratio corresponding to PVS (lbm W.V./lbm D.A.)
X	fraction of body area that is completely wet with perspiration (N.D.)

PROGRAM STATEMENTS

```
      PROGRAM PIT
      INTEGER V
      REAL ICLO, LH
C ******* READ DATA, V - AIR VELOCITY AS A PARAMETER
 12   READ (5, 15) V
 15   FORMAT (I3)
      IF (V) 3, 3, 4
C ******* READ DATA, PTI - PHYSIOLOGICAL THERMAL INDEX No.
C ******* AS A PARAMETER
      READ (5, 14) PTI
 14   FORMAT (F 3.1)
      IF (PTI - 0.5323) 30, 30, 21
 21   IF (PTI - 17.07) 31, 31, 32
C ******* STRAIGHT LINE APPROXIMATIONS OF THE X-TS CURVE
 30   TS = 3.08*PTI + 93.4
      X = (0.00234 + 0.00271*TS)/1.4
      GO TO 40
 31   TS = 0.0445*PTI + 95.0
      X = (1.552*TS-147.248)/1.4
      TO TO 40
 32   TS = 3.5*PTI + 36.2
      X = 1.0
 40   CONTINUE
      WRITE (6,160) PTI
160   FORMAT (1H1,41X, "PHYSIOLOGICAL THERMAL INDEX EQUALS",F5.1)
      WRITE (6,175)
175   FORMAT ("0", 18X, "FRACTION OF SKIN SURFACE AREA THAT IS
      -WET", 18X, "AVERAGE SKIN TEMPERATURE")
      WRITE (6,180) X, TS
180   FORMAT (1X, 34X, F5.4, 48X, F4.2)
      WRITE (6,185) V
185   FORMAT ("0", 44X, "AIR VELOCITY EQUALS", 1X, F5.1, 1X,"FT/MIN")
C ******* HEAT AND MASS TRANSFER COEFFICIENT AS FUNCTIONS OF
C       AIR VELOCITY, HC FROM KERSLAKE AND HM FROM LEWIS RELATION
C       DETERMINED BY RATIO OF NATURAL MOTION LIMIT COEFFICIENTS
C ***** AND HC
      HC = 8.3*0.176* SQRT (V*0.3048/60.)
      HM = 1.83*HC
C ******* TEST FOR NATURAL MOTION LIMIT
      IF (HC - 0.765) 55, 55, 110
 55   WRITE (6, 60)
 60   FORMAT ("0", 8X, "NATURAL MOTION LIMIT HEAT AND MASS TRANS-
      FER COEFFICIENTS IN FORCE FOR THE AIR VELOCITY LESS THAN
      -50 FT/MIN")
110   CONTINUE
C******** METABOLIC RATE AND CLOTHING PARAMETERS INITIALIZED
      MR = 400
      ICLO - 0.6
      WRITE (6.201) MR, ICLO
201   FORMAT ("0", 26X, "SUBJECT: SEDENTARY MALE, METABOLIC RATE
      -", 1X, 13, 1X, "BTU/HR, CLOTHING", 1X, F2.1, 1X, "CLO")
```

```
      WRITE 6, 205)
205   FORMAT ("0", 2X, "TI", 7X, "TO", 7X, "WI", 7X, "WO", 7X, "TA",
      -7X, "WA", 7X, "QI", 8X, "DA", 9X, "G", 10X, "HC", 7X, "HM",
      -8X," "SH, 8X, "LH")
C ******* DO-LOOP FOR CONTROLLING THE INLET SPECIFIC HUMIDITY
C ******* AS AN INDEPENDENT VARIABLE
      DO 100 1 - 1,301,25
      WI = FLOAT (I)/10000. - 0.0001
C ******* CONSTANTS ESTABLISHED
      HFG = 1039.
      ADU = 20.0
      SIGMA = 0.1714E-8
      FEFF = 0.696
      EPS = 0.97
      FCL = 1.1
      CP = 0.24
C ******* CALCULATIONS
      PVS = 3.61*TS-225.
      WS = (0.622*PVS)/(2116.-PVS)
C ******* TEST FOR NATURAL MOTION LIMIT
      IF (HC - 0.765) 65, 65, 125
 65   HC = 0.765
      HM = 1.4
125   CONTINUE
      DI = (14.7*144.)/(53.3*(460. + 50.))
275   G = DI*V* 9.0 * 60.
      FPCL - 1./(1. + 0.143*HC/0.176*ICLO*0.881)
      Z = HM*ADU*X*FPCL/G
      WO = WS - (WS - WI)*EXP(-Z)
      B = ALOG ((WS - WI)/ (WS -WO))
      LH = HM*ADU*X*HFG*FPCL*(WO - WI)/B
      SH = MR - LH
      TCL = TS - SH*ICLO/(1.135*ADU)
      HI = 0.70
236   TA = TCL - SH/((HC + HI*EPS*FEFF)*(FCL*ADU))
      R = ((460. + TCL)** 4 - (460. + TA)**4)*SIGMA
      HR = R/(TCL - TA)
      IF (ABS(HI - HR)) .GT. 0.01) 151, 251
151   HI = HR
      GO TO 236
251   DA   (14.7*144)/(53.3*(460. + TA))
C ******* TEST TO DETERMINE WHETHER CALCULATED AVERAGE AIR
C ******* STREAM DENSITY IS WITHIN ACCEPTED TOLERANCE
      IF (ABS(DI - DA) .GT. 0.00005) 150, 250
 150  DI = DA
      GO TO 275
 250  CONTINUE
      EXPO = SH/ (G*CP*(TCL - TA))
      FEXPO = 1. - EXP (EXPO)
      TO = (TCL*FEXPO + SH/(G*CP))/FEXPO
      TI = TO - SH/(G*CP)
      WA = WS - (WO - WI)/B
      RW - 85.778
      RA - 53.352
```

```
      P = 14.7*144.
      QI = ((TI + 460.)*G/P)*((WI*RW + RA)/(1. + WI))
C ******* WRITE RESULTS
      WRITE (6,190) TI, TO, WI, WO, TA, WA, QI, DI, G, HC,
     -HM, SH, LH
 190 FORMAT ("0", 2(F5.2, 4X), 2(F5.4, 4X), F5.2, 4X F5.4,
     -4X, F6.2, 4X, F6.5, 4X, F7.4, 4X, F6.4, 4X, F6.4,
     -4X, 2(F6.2, 4X))
C ******* TEMPERATURES TI, TO, TA ARE IN DEGREES F; SPECIFIC
C       HUMIDITIES WI, WO, WA ARE IN LBM W.V. / LBM D. A.:
C       QI IS IN CU. FT./hr; DA IS IN LBM/CU. FT. AND
C       IS THE DENSITY OF THE AIR STREAM AT THE AVERAGE CON-
C       DITIONS, TA AND WA; G IS THE MASS FLOW RATE IN LBM/
C       HR: HC IS THE CONVECTIVE HEAT TRANSFER COEFFICIENT IN
C       BTU / HR-SQFT-F; HM IS THE MASS TRANSFER COEFFICIENT
C       IN LBM W. V. / LBM D.A. HR-SQRT; SH IS THE SENSIBLE
C       HEAT TRANSFER IN BTU / HR; AND LH IS THE LATENT HEAT
C *******TRANSFER IN BTU / HR.
 100  CONTINUE
      GO TO 12
 3    END
```

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"THE USE OF THE PHYSIOLOGICAL THERMAL INDEX (PTI) AS AN ALTERNATE TO EFFECTIVE TEMPERATURE FOR CIVIL DEFENSE PLANNING"

UNCLASSIFIED — 70 Pages
UNIVERSITY OF SANTA CLARA — November 1977

CONTRACT # DAHC 20-69-C-0128 — WORK UNIT 1224D

ABSTRACT: Ventilation planning for survival shelters had been based on the effective temperature (ET) concept established in the literature of the American Society of Heating, Refrigeration and Air Conditioning Engineering (ASHRAE). The ET concept has been repeatedly subjected to criticism by several research workers and alternatives have been proposed. One, the new effective temperature (ET*) has replaced the ET concept in the current ASHRAE literature. Another, the Physiological Thermal Index (PTI) has been developed by University of Santa Clara researchers under DCPA sponsorship.

A comparison is presented here among three human thermal models that relate to these new indicies of comfort. The comparisons are in the context of suitability for predicting ventilation requirements for survival shelters. Some inadequacies are found in all three models. The principal conclusion is that the ET* index of comfort found in current ASHRAE literature predicts warm-hot comfort conditions at combinations of temperature and humidity that are greater than those recommended by the PTI and ET scales. Hence, caution is advised and more study recommended before the ET scale in the DCPA literature is supplanted by the ET*. Adoption of the PTI scale appears to be a more satisfactory alternative.

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