A DIRECT DETERMINATION OF THE WET BULB GLOBE TEMPERATURE IN TERMS OF ENVIRONMENTAL PROPERTIES

by

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B.S., Kansas State University, 1974

A MASTER'S THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1975

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CHAPTER I

INTRODUCTION

Since man's first days on Earth, he has probably recognized the influence which temperature, moisture, wind, and sunlight had on the way he felt and his ability to do work. As man became industrialized he was subject to increasingly severe thermal environments, and became aware that his health and lifespan were being influenced by the detrimental effects of this "thermal stress" imposed on him.

Early in the twentieth century, scientists, engineers, and physiologists began extensive investigations into the nature of thermal stresses and the resulting "thermal strain" exhibited by the human body. These studies lead to the development of many indices to evaluate thermal stress which were helpful in predicting conditions which would result in undue strain on the body. With today's ever increasing level of technology, workers are called on to perform more demanding tasks, and accurate evaluation of the working environment is becoming increasingly important to insure the safety and well-being of the worker, and to optimize worker output.

In 1970, the U.S. Congress passed the Occupational Safety and Health Act and created the National Institute for Occupational Safety and Health (NIOSH), under the Department of Health, Education, and Welfare as a research organization to provide information upon which standards are developed to protect the worker from potential hazards on the job. NIOSH developed the criteria for a recommended standard for exposure to hot environments in 1972 [29], naming the Wet Bulb Globe Temperature index (WBGT) as the index with which to evaluate the thermal environment.

The Wet Bulb Globe Temperature index was developed by C. P. Yaglou and D. Minard [39], and was originally used by them to limit the number of heat casualties in Marine Corps Training Centers. Minard [27] reported that, even with basic application of the WBGT index, heat casualties were significantly reduced in these centers. The WBGT index reflects the influence of the four thermal environment factors: dry bulb temperature, relative humidity, air movement, and radiant energy from surrounding surfaces. Two different relations are given for the WBGT index, one for application of the index for outdoor conditions, where the influence of sunlight must be considered, and one for indoor use.

Objectives of this Study

Since enforcement of the NIOSH standard for exposure to hot environments would require continuous compliance in all work situations with the stated guidelines, the need was felt for an interpretation of the WBGT index in terms of the variables which are controllable through the use of basic air conditioning equipment. The objective of this study was to provide the relationship between Wet Bulb Globe Temperature and the "controllable properties," dry bulb temperature, relative humidity, and air velocity, so that air conditioning systems can be designed with the WBGT index as the governing criterion. Included in this goal was the development of a concise form of presenting this information in such a way as to provide a tool which will allow the designer to determine an acceptable set of design conditions with reasonable accuracy and relative ease.

Basis and Scope of the Research

The derivation of the relationship between the environmental conditions and the WBGT index is based on the analysis of the heat transfer for a 6 inch diameter hollow copper sphere (painted black), and through the analysis of the heat and mass transfer for a 0.25 inch diameter natural (non-aspirated) wet bulb thermometer.

This investigation was concerned only with the interpretation of the Wet Bulb Globe Temperature index in terms of design parameters. No attempt was made to demonstrate the value of the index as a heat stress indicator. Also, only the indoor application of the index was considered, as this is the situation under which the index would most commonly be applied.

The analysis of the index encompasses values from WBGT = 70 °F to WBGT = 90 °F, a range which represents conditions from mild to severe heat stress. These values are the result of combinations of the following environmental factors in the ranges shown:

Air Velocity from 50 fpm to 200 fpm

Mean radiant temperature from 70 °F to 150 °F

Dry bulb temperature (t_d) and dew point temperature (t_{dew}) combinations within the region of the psycrometric chart described by:

t_d = 70 °F , t_{dew} = 25 °F

t_d = 70 °F , t_{dew} = 70 °F

t_d = 85 °F , t_{dew} = 85 °F

t_d = 105 °F , t_{dew} = 80 °F

t_d = 105 °F , t_{dew} = 25 °F

The values of environmental factors which lie within the above specified ranges are those which are commonly found in many industrial situations.

Definition of Terms

1. Air Velocity:

    The velocity (speed) of the air in the space or at the location under consideration.

2. Dry Bulb Temperature:

    The temperature indicated by a dry bulb thermometer in equilibrium with the environment.

3. Environmental Factors (or Climatic Factors):

    Those properties which describe the thermal environment - dry bulb temperature, dew point temperature (or other reference state, such as wet bulb temperature, humidity ratio, relative humidity, etc.), air velocity, and mean radiant temperature.

4. Globe Temperature:

    The temperature of a 6-inch diameter copper globe, painted flat black, at thermal equilibrium with the environment.

5. Mean Radiant Temperature:

    The uniform temperature of the surrounding surfaces which will yield the same exchange of radiant energy with a black body as do the actual surroundings.

6. Naturally Aspirated Wet Bulb Temperature (or Natural Wet Bulb Temperature):

    The temperature indicated by a wet bulb thermometer, at thermal equilibrium, which is influenced by the normal movement of the surrounding air.

7. Wet Bulb Temperature:

    The temperature indicated by a wet bulb thermometer, at thermal equilibrium, which is fully aspirated (velocity of air over thermometer > 900 fpm) by the surrounding air.

Nomenclature

Symbol	Significance	Units
A_g	Surface area of globe thermometer	FT^2
A_n	Surface area of natural wet bulb thermometer	FT^2
$C_{p,a}$	Specific heat of air	BTU/lb_m-°F
D	Diffusion coefficient	FT^2/hr
d_g	Diameter of Globe thermometer	FT
d_n	Diameter of natural wet bulb thermometer	FT
h_{fg}	Enthalpy of evaporation of water	BTU/lb_m
$h_{c,g}$	Convection heat transfer coefficient, globe thermometer	$BTU/hr\text{-}ft^2$-°F
$h_{c,n}$	Convection heat transfer coefficient, natural wet bulb thermometer	$BTU/hr\text{-}ft^2$-°F
h_D	Mass transfer coefficient	$\frac{lb_m H_2O}{hr\text{-}ft^2\text{-}lb_m H_2O/lb_m air}$
k_a	Thermal conductivity of air	BTU/hr-ft-°F
Le	Lewis number = $k_a/\rho_a C_{p,a} D$	dimensionless
Nu	Nusselt number = $h\ d/k_a$	dimensionless
Re	Reynolds number = $V_a \rho_a\ d/\mu_a$	dimensionless
MRT	Mean radiant temperature	°R
P_a	Partial pressure of water vapor contained in air	PSIA
P_n	Saturation pressure of water vapor at the natural wet bulb temperature	PSIA
t_{db}	Dry bulb temperature	°F
t_{dew}	Dew point temperature	°F
T_{dew}	Dew point temperature (abs.)	°R
t_g	Globe temperature	°F

Nomenclature (Continued)

Symbol	Significance	Units
T_g	Globe temperature (abs.)	°R
t_n	Natural wet bulb temperature	°F
T_n	Natural wet bulb temperature (abs.)	°R
t_w	Fully aspirated wet bulb temperature	°F
V_a	Air velocity	ft/hr.
W_a	Humidity ratio of air	$lb_m H_2O/lb_m air$
$W_{s,n}$	Humidity ratio at saturation at the natural wet bulb temperature	$lb_m H_2O/lb_m air$

GREEK SYMBOLS

Symbol	Significance	Units
ε_g	Emissivity of globe thermometer surface	dimensionless
ε_n	Emissivity of natural wet bulb thermometer surface	dimensionless
μ_a	Dynamic viscosity of air	$lb_m/hr\text{-}ft$
ρ_a	Density of air	lb_m/ft^3
σ	Stefan - Boltzman constant = 0.1714×10^{-8}	$BTU/hr\text{-}ft^2\text{-}°R^4$

CHAPTER II

REVIEW OF THE LITERATURE

Early Studies on Heat Stress

Modern studies of the effects of heat on workers began at the turn of the century with studies of Cornish miners exposed to the severe environments of tin mines. With the exception of the work done by Bedford and Warner [2] on the influence of heat on the performance of coal miners, little progress was made in the field for more than thirty years.

In the mid 1930's, renewed interest in heat stress led to the identification of chronic disorders caused by heat exposure. Many investigators, including J. H. Talbot [32], C. P. Yaglou [37], and D. B. Dill [9], realized the need to set forth standards to prevent exposure of workers to dangerous environments. The first of these standards was presented by the Committee on Atmospheric Comfort in 1947 [36]. The guidelines stated by this publication were found to be confusing, and the permissible limits were too vague for satisfactory application. In 1950, Yaglou [38] outlined more suitable limits for heat exposure. These limits are still considered applicable for use in industry today.

Studies to Derive a Heat Stress Index

During the past twenty-five years, much effort has been directed towards determining methods of predicting the physiological response of man to heat exposure through the measurement of environmental conditions. Primarily, these investigations centered around the industrial situation. The aim of these studies has been to model man's bodily response to the imposed environment. Very few epidemiological studies have been done during this period. Data from earlier studies on morbidity and mortality by Britten and Thomson [8] in 1926 and by Yaglou [35] in 1937 is too sparse to be accurately applied to today's industrial situations.

In order to determine an index which would provide a reliable and accurate indication of the effects of heat, NIOSH [29] established some basic objectives for a heat stress index:

a) The index should incorporate all four of the basic climatic factors -- dry bulb temperature, relative humidity, air velocity, and radiant heat

b) Consideration should be given to age, gender, fitness, body build, posture and the degree of acclimatization to heat.

c) The index should lend itself to the establishment of reliable limits to heat exposure, and should provide a predictable margin of safety.

In addition to these criteria, additional considerations should be made for an index designed for industrial use:

d) The index should have proven applicability in an industrial situation.

e) Any measurements and calculations required for proper use of the index should be simple.

f) The factors included in the index should be weighed to properly reflect their relation to the physiological strain of the industrial worker.

Applicable Indices

Past research has produced a large number of indices to relate imposed heat stress to resulting physiological strain, with many having characteristics making them favorable for use in industry. For the purposes of this study, only the most common of these indices will be discussed, these being:

1) The predicted 4-hour sweat rate

2) Heat Stress Index of Belding and Hatch

3) Effective Temperature (with modifications)

4) The Wet Bulb Globe Temperature Index

Information on other indices such as Givoni's Index of Thermal Stress, Operative Temperature, the Heat Strain Predictive System, and the Index of Physiological Effect is referenced in such sources as Fanger [12], the ASHRAE Handbook of Fundamentals [1], and Hardy, Gagge and Stolwijk [14].

The Relative Strain Index [20] is the only index that gives significant consideration to age, gender, fitness, acclimatization and other heat tolerance factors. These criteria, though, is oriented to fallout shelter environments, and is too specific to qualify as a suitable index for general industrial application. It is superior to the four indices reviewed here only in the sense that it does account for these factors.

The Predicted 4-Hour Sweat Rate (P4SR) was developed by McArdle, et al. [24] on behalf of the Royal Navy in 1947. Its basis of development was experimental data observed on the four hour naval watch. In addition to considering the four basic climatic factors, the P4SR also accounts for the rate of work output and two clothing ensembles (stripped to the waist in shorts, and overalls worn over shorts). Since the Wet Bulb Globe Temperature Index accounts for neither one of these factors, the P4SR has a distinct advantage over the WBGT Index, particularly when a task requires unique skills or involves unusual environmental situations.

The P4SR index number indicates the total sweat loss (in liters) during a four hour period. The index is intentionally overestimated for severe heat conditions (P4SR greater than 5 liters) to provide a built-in factor of safety.

Some modifications have been recommended by Leithead and Lind [22] to improve the P4SR's indication of physiological strain.

Many difficulties arise in the use of the P4SR. One of its primary shortcomings is the very complicated calculations required to utilize the index, where determination of several intermediate values is required for most applications. In addition, a nomogram must be employed, where interpolation is difficult and may not be justifiable due to its empirical nature. Although the most complicated of the indices considered here, the P4SR may well be the most precise. Investigations into its accuracy are referenced by Kerslake [19].

The P4SR loses sensitivity in severe thermal conditions, and even the overestimation in the upper values may not provide an accurate relationship to strain to make it completely adequate. Minard, et al. [28] observed that the correlation between the observed sweat rate and P4SR was not good, particularly for the upper and lower extremes of heat exposure.

The Wet Bulb Globe Temperature Index exibits its superiority over the P4SR not only in its greatly simplified calculation requirements, but also in the fact that no measurement of air velocity is required to determine the WBGT. NIOSH studies [11] have revealed that difficulties in establishing correct values for velocity over extended lengths of time posed a considerable problem in establishing stress index values to describe the environment over a full 8-hour work shift. The failure to properly evaluate the conditions over the entire work period may lead to misapplication of allowable heat stress limits.

In 1955, H.S. Belding and T. F. Hatch developed the Heat Stress Index (HSI) [5]. This index stemmed from the analysis of the heat exchange between man and his environment. The value given by the HSI is the percentage ratio of evaporation required for thermal balance to the maximum evaporation attainable in the given environment. An HSI value of 100 or less indicates thermal balance can be maintained by the body.

The heat exchange analysis is based on two assumptions: (1) A "standard man" (5'8", 154 lb. young man) is used, and (2) a constant skin temperature of 95 °F is assumed. Studies by Hutchinson and Baker [17] indicate that 95% of the population agrees with the "standard man" in terms of radiant heat exchange shape factor. No specific information was found on relationships for convective or evaporative heat exchange, but Belding and Hatch [5] indicate that the assumed constant 95 °F skin temperature provides an overestimation of stress in severe conditions. Modifications by Haines and Hatch [13] for industrial situations and by Hertig and Belding [15] for normally clothed workers gives improved applicability for the HSI, originally derived for nude subjects.

Like the P4SR, the Heat Stress Index accounts for the work rate -- a factor definitely favored in stress indices. The HSI also has a unique characteristic associated with it. By using the index to indicate heat loss of the body, minimum recovery time can be estimated. Also, as pointed out by Belding [4], when the HSI value exceeds 100, the difference may be used to predict tolerance time.

In order to determine the Heat Stress Index, the use of five separate charts is required. Although nomographs developed by McKarns and Brief [25] are available to determine HSI, they do not significantly simplify the procedure. Difficulty in evaluating HSI values, coupled with the need to directly measure air velocity, give favor to the WBGT index over the HSI. Even though the WBGT Index is superior to the HSI for general use in industry, the versatility of the HSI may make it a favorable companion of the WBGT index. Brief and Confer [7] have developed a direct correlation between the HSI and the WBGT index to aid in the interchanging of the indices (although NIOSH recognizes only the WBGT index in evaluating heat stress).

One of the oldest and probably the best known and most applied thermal index is the Effective Temperature scale (ET). Developed by F. G. Houghten and C. P. Yaglou [16] in 1923, E.T. is based on the concept of equal comfort, although not only the same sensory effect is experienced on initial exposure to equivalent environments, but also the same physiological strains result after longer exposures. In its original form, Effective Temperature does not account for radiation. For industrial application, a modification of ET must be used which does consider radiation.

Bedford and Warner [3] established the Corrected Effective Temperature (CET) scale based on the work of Vernon and Warner in 1932. Employing the Vernon Globe Thermometer, CET is determined by entering the ET scale with globe temperature instead of dry bulb temperature. In 1950, Yaglou [37] refuted the CET, proposing a modification of Bedford's procedure which he called Effective Temperature Including Radiation (ETR). Determination of the ETR is more difficult than finding CET, requiring adjustment of the wet bulb temperature to the actual vapor pressure. Investigations reported by Kerslake [19] and Minard [26] indicate that values for CET and ETR do not differ significantly in most applications to justify the added difficulty in determining ETR.

Other modifications of Effective Temperature scales include the basic scale (for subjects stripped to the waist) and the normal scale (for normally clothed subjects), plus the development of New Effective Temperature (ET*), which relates lines of constant physiological strain with the 50% relative humidity line, a situation found more commonly in actual environments than the 100% RH condition upon which ET is based. A good overview of ET* is given in the ASHRAE Handbook of Fundamentals [1].

Although still requiring a nomogram to determine the index value, ET is considerably easier to evaluate than P4SR or HSI. The major shortcoming of ET which makes it inferior to the WBGT index for industrial use is the fact that a separate measurement of air velocity must be made. Other criticisms of ET include its underestimation of the effects of humidity in the zone of evaporative regulation, falling short of the reliability exibited by P4SR or HSI in this region, and the over emphasis of humidity in the zone of vasomotor regulation. Minard [26] reports that investigations by Berber, Kerslake, and Waddell indicate that physiological responses are non-linear with respect to ET, but statistical analysis by Smith shows that the differences in predicting physiological responses by ET are not significant when compared to linear scales such as P4SR. Unlike P4SR and HSI, Effective Temperature does not account for work rate. The nature of its derivation restricts its use to sedentary conditions, although ET is not limited to the zone of evaporative thermoregulation, giving it the added versitility of application in all situations normally encountered in the home, office, and industrial environment.

In order to overcome the problem of incorporating radiant heat into Effective Temperature, Yaglou and Minard [39] developed the Wet Bulb Globe Temperature Index (WBGT). Their development procedures allowed them to eliminate the need to obtain a separate velocity measurement, an important factor as described earlier, and to reduce the technique of determining WBGT down to one very simple formula. In addition, the equipment upon which WBGT was based is simple to use and durable. These factors carried significant weight in NIOSH's decision to select the WBGT index as the recognized indicator of heat stress.

Many relationships have been proposed for determining the WBGT index. Yaglou and Minard [39] proposed the original formula for finding WBGT, where

$$WBGT = 0.7\ t_w + 0.3\ t_g \tag{1}$$

On further consideration, Minard [26] suggested that in addition to the above relationship, two others could be used:

$$WBGT = 0.7\ t_n + 0.3\ t_{OD}\ , \tag{2}$$

where t_{OD} is the temperature of a copper sphere covered with olive drab cloth to account for the effects of radiation for military fatigue uniforms, and

$$WBGT = 0.7\ t_n + 0.2\ t_g + 0.1\ t_{db} \tag{3}$$

for use in sunlit conditions. Lee [21] concurred with the use of Eqn. (1), but recommended the use of

$$WBGT = 0.7\ t_w + 0.2\ t_g + 0.1\ t_{db} \tag{4}$$

for circumstances where sunlight was a factor. Finally, Botsford [6] reports that a personal communication between C. P. Yaglou and R. S. Brief resulted in the introduction of

$$WBGT = 0.7\ t_n + 0.3\ t_g \tag{5}$$

as the relation for indoor use. The NIOSH recommended standard [29] is based on the use of Eqn. (5) (indoor use) and Eqn. (3) (for sunlit conditions). The distinguishing feature in Eqns. (3) and (5) is the use of the natural wet bulb temperature (t_n). Use of t_n reflects the effect of air motion on evaporative cooling even in the absence of radiation, a characteristic which Ramanathan and Belding [31] believe makes the WBGT a sufficient indicator of heat stress in the industrial environment.

Since the WBGT index is a derivative of Effective Temperature, its characteristics are similar to those of ET. Its greatest shortcoming is the underestimation of the effects of high humidity. Investigations by Ramanathan and Belding [31] have shown that young men worked successfully in moderately severe conditions described by the same WBGT index, except when those conditions were the result of high humidities, at which point the men were unable to accomplish their tasks. As a result of these studies, it was recommended that a two condition WBGT index (one for wet and one for dry environments) be established to achieve valid weighting of the index value to the total physiological strain.

In spite of its shortcomings, the WBGT index has been found to be the best indicator of heat stress in the industrial situation. The inexpensive and reliable instrumentation needed and simple calculations required for use of the index not only provide for quick and accurate evaluation of the environment, but may also prompt use of the index in situations where a more complicated one would go unused.

At this time, maximum exposure limits have been proposed by NIOSH, setting WBGT = 86 °F as the maximum allowable value for a continuous, 8-hour work period. Maximum values for shorter work periods are also given by the NIOSH proposed standard [29]. A more extensive study has been done by Dukes-Dobos and Henschel [10], establishing limits for both continuous and intermittent work periods, with consideration for work load. The maximum value given is WBGT = 90 °F for light work load with 25% work, 75% rest each hour.

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CHAPTER III

THEORETICAL DEVELOPMENT

In order to utilize the Wet Bulb Globe Temperature Index as a basis for design, the index must be related to the individual environmental constituents. This was accomplished through a heat and mass transfer balance for the natural wet bulb thermometer and a heat transfer balance on the globe thermometer. The equations were derived through the use of empirical relationships for heat and mass transfer.

Naturally Aspirated Wet Bulb Thermometer

Figure 1 gives the schematic representation of heat and mass transfer for the natural wet bulb thermometer.

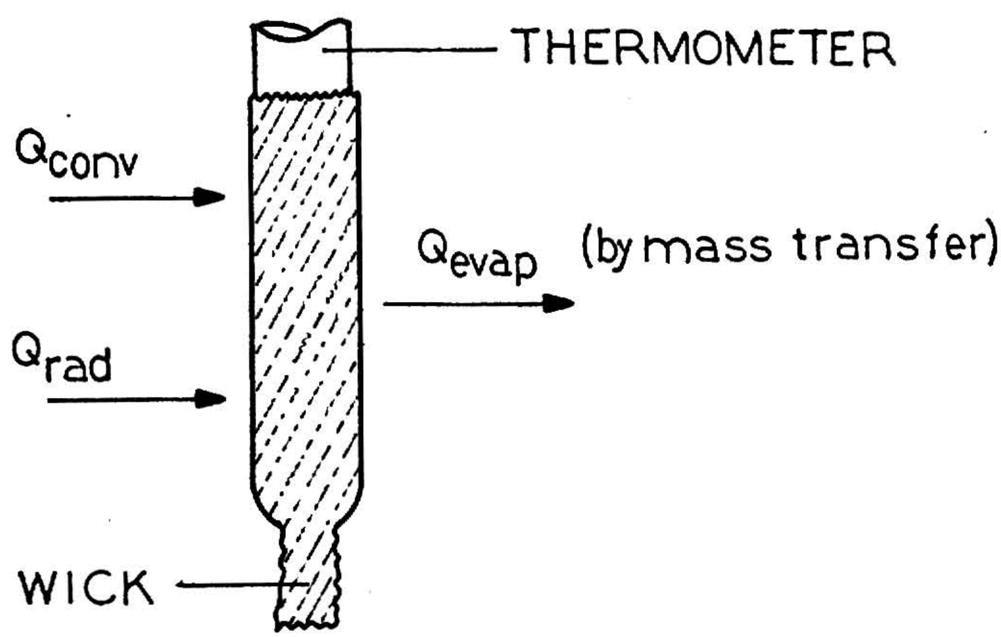

Figure 1

Heat and Mass Transfer: Naturally Aspirated Wet Bulb Thermometer

The steady state heat transfer may be expressed as follows:

Heat Gain:

$$\text{Heat Transfer by convection} = h_{c,n}A_n(t_{db} - t_n) \tag{6}$$

$$\text{Heat Transfer by radiation} = \sigma A_n \varepsilon_n(MRT^4 - T_n^4) \tag{7}$$

Heat Loss:

$$\text{Heat Transfer by evaporation} = h_D A_n(W_{s,n} - W_a)h_{fg} \tag{8}$$

Equating the heat gains and heat loss, where the area A_n cancels:

$$h_{c,n}(t_{db} - t_n) + \sigma\varepsilon_n(MRT^4 - T_n^4) = h_D(W_{s,n} - W_a)h_{fg} \tag{9}$$

Globe Thermometer

Figure 2 shows the heat transfer of the globe thermometer.

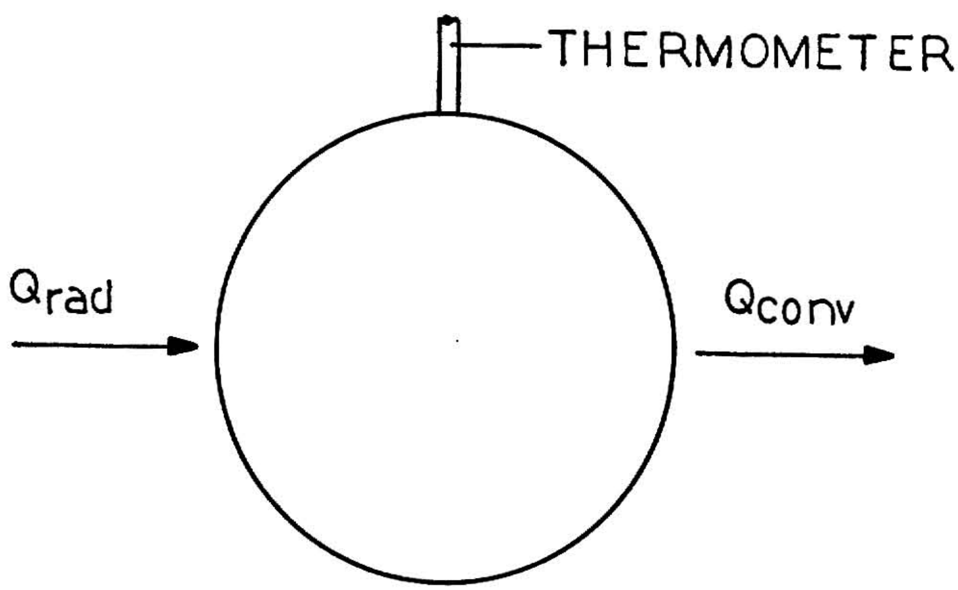

Figure 2

Heat Transfer: Globe Thermometer

For the globe thermometer, the steady state heat transfer is given by:

Heat Gain:

$$\text{Heat transfer by radiation} = \sigma A_g \varepsilon_g (MRT^4 - T_g^4) \tag{10}$$

Heat Loss:

$$\text{Heat Transfer by convection} = h_{c,g} A_g (t_g - t_{db}) \tag{11}$$

Setting heat loss = heat gain, where the area of the globe is equal on both sides of the equation:

$$\sigma \varepsilon_g (MRT^4 - T_g^4) = h_{c,g}(t_g - t_{db}) \tag{12}$$

Evaluation of Terms

Natural Wet Bulb Thermometer

To determine the convective heat transfer coefficient in Eqn. (6), an empirical relation developed for heat transfer from a cylinder was used. McAdams [23] recommends the following relation for the flow of air normal to single wires and cylinders:

$$Nu = \frac{h_{c,n}\, d_n}{k_a} = 0.615(Re)^{0.466} \qquad 40 < Re < 4000 \tag{13}$$

Solving for $h_{c,n}$ and expanding the Reynolds number, Eqn. (13) becomes

$$h_{c,n} = 0.615\left(\frac{V_a \rho_a d_n}{\mu_a}\right)^{0.466} \left(\frac{k_a}{d_n}\right) \tag{14}$$

Evaluating the properties of moist air, ρ_a, μ_a, and k_a, over the range of film temperatures from 50 °F to 110 °F, and taking the diameter of the natural wet bulb thermometer to be 0.25 inch, the mean value of the convection heat transfer coefficient was found to be:

$$h_{c,n} = 0.09106\ V_a^{0.466}\ , \tag{15}$$

with a standard deviation from the mean of 0.54% in the temperature range specified. (Data for this calculation may be found in Appendix A).

The emissivity of the natural wet bulb thermometer wick was taken as:

$$\varepsilon_n = 0.95$$

The mass transfer coefficient in Eqn. (8) was evaluated through the Lewis number relation:

$$h_D = \frac{h_{c,n}}{C_{p,a}\ Le} \tag{16}$$

The work of Arnold, as reported by Threlkeld [33], gives values for the Lewis number for evaporation of water from a wet bulb thermometer wick. From these data:

$$Le = 0.9$$

(This value is accurate to $\pm 2.2\%$ for velocities from 10 to 1000 fpm, according to Arnold's data). Taking $C_{p,a} = 0.24$ over the entire range, and using the value for $h_{c,n}$ from Eqn. (15), the mass transfer coefficient was found to be:

$$h_D = 0.42157\ V_a^{0.466} \tag{17}$$

The value for the enthalpy of evaporation, h_{fg}, was taken at 75 °F, where:

$$h_{fg} = 1051.5\ \mathrm{BTU/lb_m}$$

Substituting the values for $h_{c,n}$, h_D, h_{fg}, ε_n, and the Stefan-Boltzman constant, σ, into Eqn. (9) yields:

$$0.09106\ V_a^{0.466}(t_{db} - t_n) + 1.7882 \times 10^{-9}(MRT^4 - T_n^4)$$

$$= 433.28\ V_a^{0.466}(W_{s,n} - W_a) \tag{18}$$

Globe Thermometer

The convection heat transfer coefficient in Eqn. (11) was evaluated through the use of the relation recommended by McAdams [23] for heat transfer from spheres to a flowing gas:

$$Nu = \frac{h_{c,g}\, d_g}{k_a} = 0.37(Re)^{0.6} \qquad 17 < Re < 70{,}000 \tag{19}$$

Solving for the coefficient $h_{c,g}$, Eqn. (16) becomes:

$$h_{c,g} = 0.37\left(\frac{V_a \rho_a d_g}{\mu_a}\right)^{0.6} \left(\frac{k_a}{d_g}\right) \tag{20}$$

Evaluation of the properties of moist air over the range of film temperatures from 60 °F to 200 °F, and using the prescribed diameter of 6 inches for the globe, the convection heat transfer coefficient was found to be:

$$h_{c,g} = 0.009413\ V_a^{0.6} \tag{21}$$

with a standard deviation from the mean of 3.56% in the temperature range specified. (The data for this calculation may also be found in Appendix A). Taking the value for the emissivity of the black globe to be

$$\varepsilon_g = 0.95\ ,$$

and substituing for the Stefan-Boltzman constant, σ, Eqn. (12) becomes:

$$1.6283 \times 10^{-9}(MRT^4 - T_g^4) = 0.009413\ V_a^{0.6}(t_g - t_{db}) \tag{22}$$

Solution of the Governing Equations

Equations (18) and (22) are the equations which describe the heat and mass transfer in terms of the properties of the environment for the WBGT index. Extracting those environmental properties which give a particular WBGT index requires the solution of the two equations by a trial and adjustment technique, described below.

Beginning with the relationship for the WBGT Index:

$$WBGT = 0.7\ t_n + 0.3\ t_g\ , \tag{5}$$

The globe temperature may be found by assuming a value for WBGT and t_n.

$$t_g = 3.3333\ WBGT - 2.3333\ t_n \tag{23}$$

Solving Eqn. (22) for dry bulb temperature gives

$$t_{db} = t_g - 1.7298 \times 10^{-7}(MRT^4 - T_g^4)/V_a^{0.6} \tag{24}$$

By choosing values for the mean radiant temperature (MRT) and velocity, and substituting the value of t_g found from Eqn. (23), the corresponding dry bulb temperature can be found.

Solving Eqn. (18) for dry bulb temperature yields

$$t_{db} = t_n + 4758.18(W_{s,n} - W_a)$$

$$- 1.7882 \times 10^{-8}(MRT^4 - T_n^4)/V_a^{0.466} \tag{25}$$

The solution of Eqn. (25) requires the evaluation of the humidity ratio at the natural wet bulb temperature and at the dew point temperature of the air. Since the natural wet bulb temperature has already been assumed for the solution of Eqn. (23), the humidity ratio at this temperature can be calculated directly.

First, the vapor pressure at t_n is found by the following relationship given by Keenan and Keyes [18]:

$$P_n = 218.167(P_{bar})10^{\left(1 - \frac{1165.07}{T_n}\right)\left(\frac{a' + b'x + c'x^3}{1 + d'x}\right)} \tag{26}$$

where

$a' = 3.243781$

$b' = 3.260144 \times 10^{-3}$

$c' = 2.006581 \times 10^{-9}$

$d' = 1.215470 \times 10^{-3}$

$x = 1165.07 - T_n$

and the barometric pressure P_{bar} can be assumed to be 14.696 psia for the general case. The humidity ratio at the natural wet bulb temperature is then given by

$$W_{s,n} = 0.62198 \frac{P_n}{P_{bar} - P_n} \tag{27}$$

The dew point temperature may now be found by trial and error. By assuming values of t_{dew} and inserting them in Eqn. (26) (where the value of T_{dew} is used in the place of T_n, and the resulting vapor pressure is now P_a), and solving for the humidity ratio of the air by Eqn. (27) (P_n is replaced by P_a, the equation yields W_a), Eqn. (25) can be solved for t_{db}. The process of assuming t_{dew} is continued until Eqn. (25) yields the same value of dry bulb temperature as found in Eqn. (24) (within reasonable limits). Appendix B contains a computer program which performs this operation for values of the WBGT index from 70 °F to 90 °F in 1 °F increments, for MRT values from 60 °F to 200 °F, and for a selected air velocity. (This program solves for a dry bulb temperature from Eqn. (25) which is within ± 0.002 °F of the value of t_{db} found from Eqn. (24)).

Results of the Evaluation of the Governing Equations

From values obtained through use of the computer program in Appendix B, lines of constant Wet Bulb Globe Temperature Index were found to be straight lines (for any given value of velocity and MRT) when plotted on a psychrometric chart, with slopes of greater (negative) value than the slopes of lines of constant enthalpy. This is in agreement with the results reported by Kerslake [19]. From this, determination of the locus of values of dry bulb temperatures and dew point temperatures for a given WBGT index is greatly simplified. Finding values of dry bulb and wet bulb temperature (at some selected velocity and mean radiant temperature) which satisfy Eqns. (24) and (25) at two extremes of the psychrometric chart and drawing a straight line through them will yield the entire set of combinations of dry bulb and wet bulb temperatures for the desired WBGT index. This plot is valid only for the chosen air velocity and MRT. If either of these properties is changed, a new set of dry bulb and wet bulb temperatures must be obtained. This subject is pursued further in Chapter V, "Application of Results."

CHAPTER IV

EXPERIMENTAL INVESTIGATION

In order to verify that the empirical relations used to interpret the WBGT index were properly sensitive to the four basic environmental factors, the theoretical correlations were checked against measured values obtained by placing the WBGT apparatus in a chamber where the environmental factors could be controlled and measured. A description of the equipment and procedures for this investigation, as well as the results of the experiment, is presented in the following text.

Apparatus

A schematic representation of the test apparatus is given in Figure 3. The fan contained in the plenum was a Dayton 15-inch centrifugal fan driven at 700 rpm by a 1-1/2 horsepower electric motor. The air was carried through an 18-1/2 inch by 15-3/4 inch duct, with a two-way transition having an 18° vertical and 15° horizontal divergence angle connecting the duct with the chamber. A single thickness of burlap backed by a piece of 1/2 inch by 1/4 inch rolled expanded metal was used to provide uniform flow. The flow was adjusted to a uniformity of $\pm 10\%$ at 200 fpm, this being sufficient, considering the insensitivity of the equations to velocity. The air was returned through a 35 inch by x 35 inch duct which turned 90° into a 35 inch by 12 inch throat. In the upper section of the throat, a 35 inch by 12 inch damper was installed to control air velocity. Below the damper, two 12 inch by 10 inch evaporator coils were placed on either side of the throat for cooling, with an 11 inch by 12 inch adjustable bypass damper between them. Refrigeration was supplied by a conventional 3/4 ton Freon unit. Evaporator coil temperature was controlled by manually adjustable expansion values on each coil. The air was then returned to the plenum through a 35 inch by 25 inch section. A 1/2" steam pipe entered at the top of this section for humidification, and two 660-watt electric cone heaters were installed at the center of the rear of the section. Humidification was controlled by a valving arrangement, and air heating was controlled by adjusting the voltage to the two cone heaters with variacs. A dehumidifier was connected across the upper and lower sections of the return duct, but was never needed, since the ambient air was at a very low humidity ratio.

The inside walls of the test chamber were constructed from 1/32 inch stainless steel sheet painted flat black. On the back of each panel, 1/4 inch copper tubing on 2-3/4 inch centers as was placed in a counter-flow arrangement shown in Figure 3. The tubes were coated with high thermal conductivity paste to provide good thermal contact between the tubes and the panels. Hot water was pumped through the tubes at approximately 5 gpm to provide panel heating. The water was supplied from a 10 gallon reservoir, heated by two 2000-watt immersible electric heaters controlled by variacs. Steam was used to obtain high panel temperatures. Copper-constantan thermocouples were welded to the back surfaces of the panels -- six on the top and bottom and four on each side panel -- to measure panel temperatures. The back of each panel was covered with 5/8 inch foam insulation and 1/4 inch composition board. The entire chamber was assembled with nylon bolts to reduce heat transfer.

Instrumentation

Conditions of the air within the chamber were evaluated by measuring the dry bulb and wet bulb temperatures. Dry bulb temperature was measured

LEGEND for FIGURE 3

A - Test Chamber

B - Copper Tubing (for wall heating)

C - Air Control Damper

D - Evaporator Coils (2)

E - Steam Inlet (for humidification)

F - Cone Heaters (2)

G - Return Duct

H - Fan Plenum

I - Supply Duct

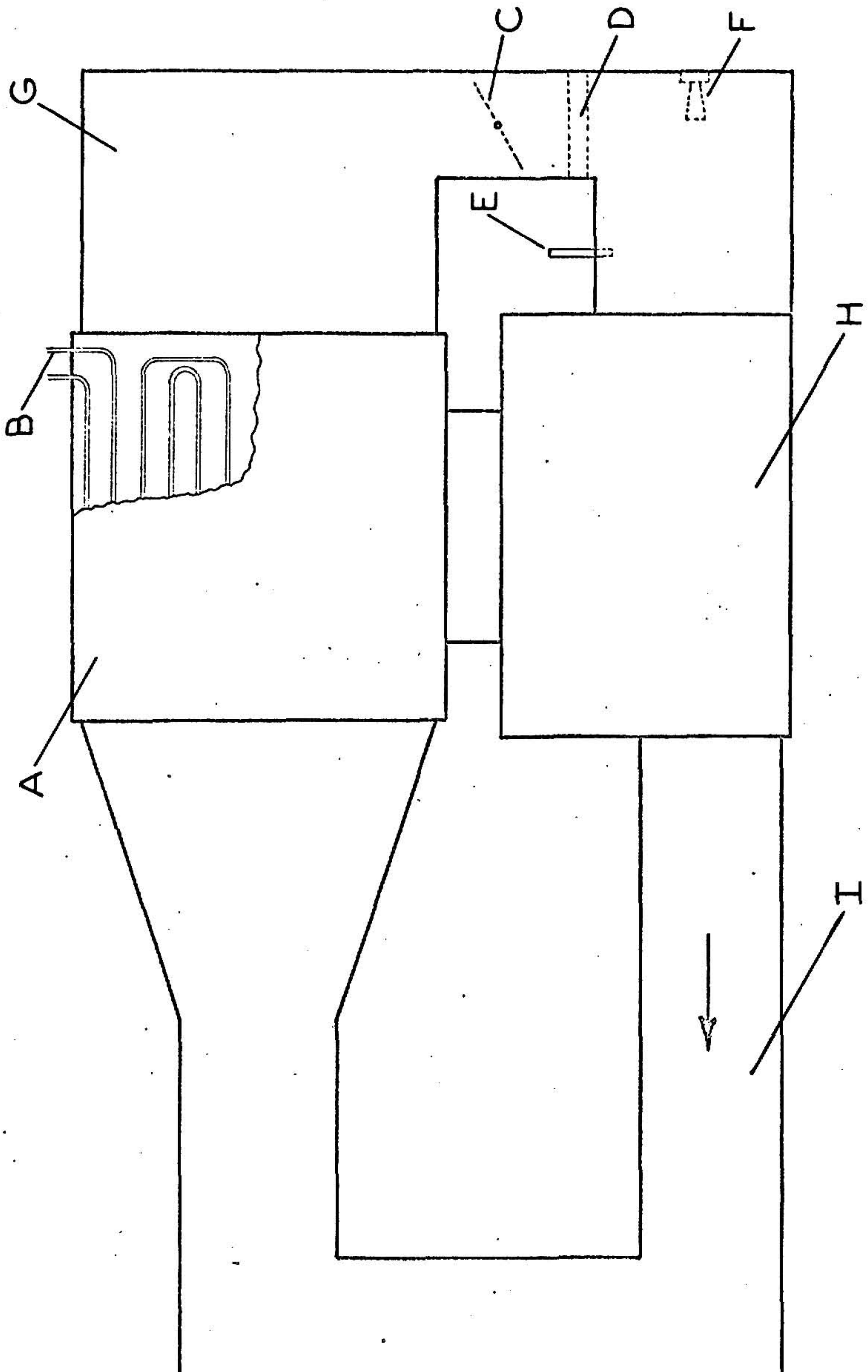

FIGURE 3

by a 400 series thermistor in a shielded tube. This, and other instruments, is shown in Figure 4. Wet bulb temperature was measured with a psychrometer, mounted on the outside of the return duct, using a thermistor to measure temperature. Air was brought to the psycrometer from the chamber through a 3/4 inch tube. Air velocity was measured with an Anemotherm Model 60 Air Meter. The anemometer probe was mounted on a wood dowel, and could traverse the width of the chamber immediately in front of the globe thermometer.

The globe thermometer and natural wet bulb thermometer were mounted as shown. With the return duct in place, the globe was located in the center of the chamber. Both temperatures were measured with 400 series thermistors.

Output of the panel thermocouples was measured and recorded with a Beckman digital voltmeter-scanner-recorder. Temperatures of the thermistor probes were indicated by a Digitec digital temperature indicator. A more detailed description of the instruments, along with their accuracies, is given in the first part of Appendix E.

Procedures

Figure 5 shows the bounds of the area of interest in this study, plotted on a psychrometric chart. Tests were run near the extremes and center of this area at approximately the same air velocity and mean radiant temperature, to insure that the theoretical relationships were properly responsive over the desired range of dry bulb and dew point temperatures. Further runs were made near the center of the region, first with different velocities, then with different mean radiant temperatures, to test the sensitivity of

LEGEND for FIGURE 4

A - Globe Thermometer

B - Natural Wet Bulb Thermometer

C - Dry Bulb Thermometer (with shield)

D - Anemotherm Probe

E - Psychrometer

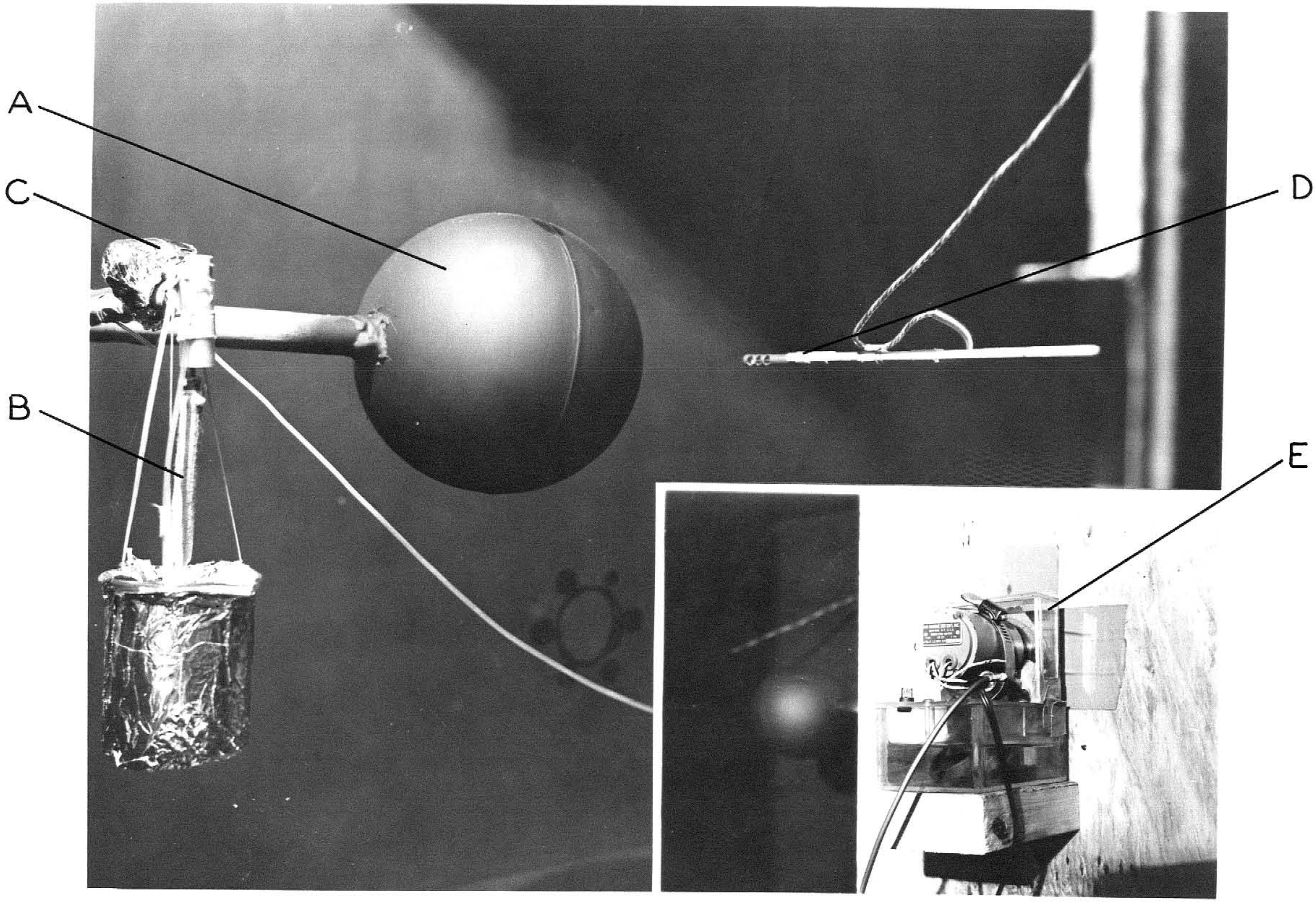

FIGURE 4

ILLEGIBLE DOCUMENT

THE FOLLOWING DOCUMENT(S) IS OF POOR LEGIBILITY IN THE ORIGINAL

THIS IS THE BEST COPY AVAILABLE

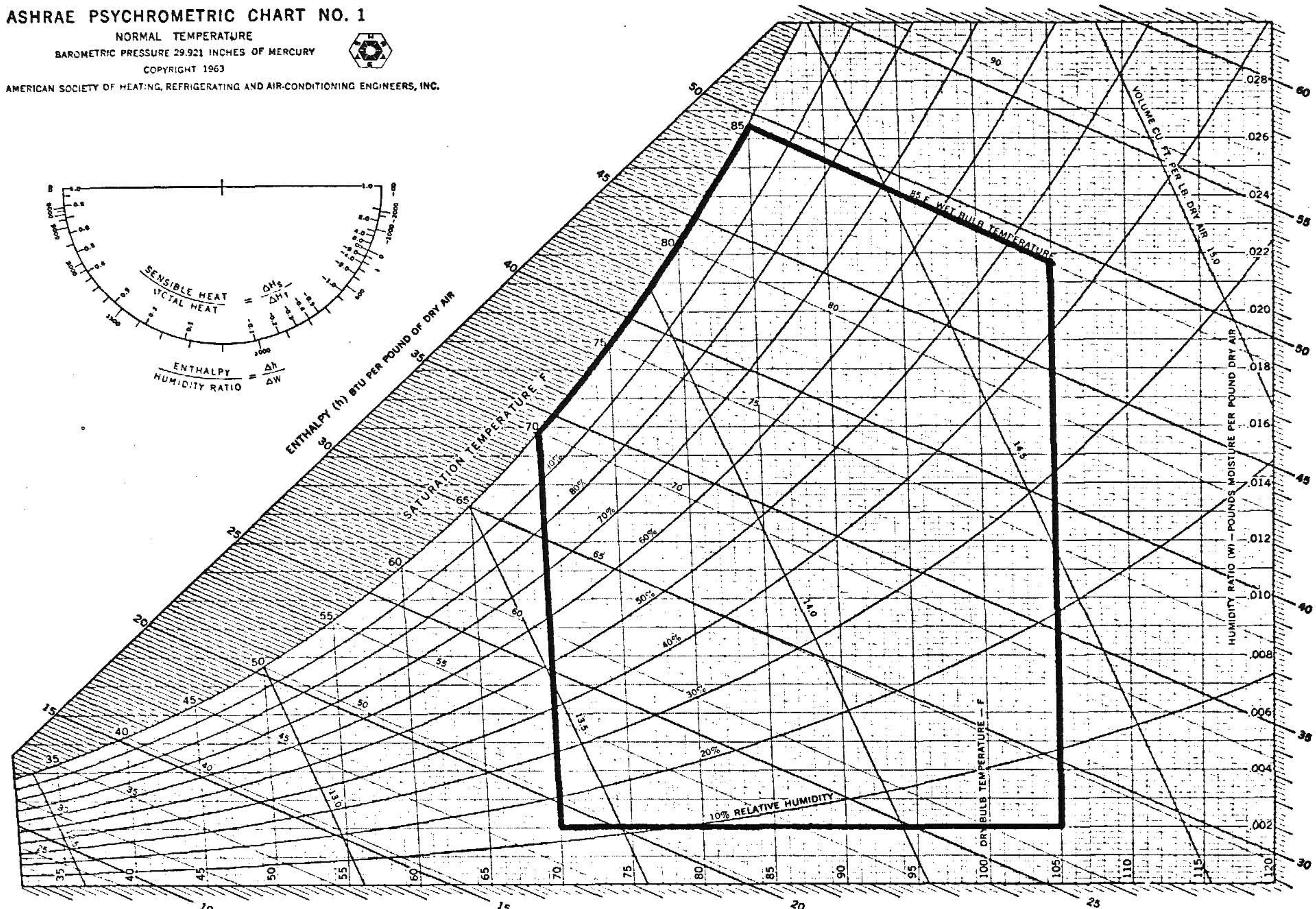

FIGURE 5

chart courtesy of ASHRAE

the relations to these factors. On each run, data not recorded by an automatic printing device was registered on a data sheet such as the one shown in Figure 6.

Although the test chamber responded rapidly to changes in test conditions, approximately 3 hours were allowed on each run for the systems and instruments to come to steady state after the desired conditions were selected. The only exception to this was in the operation of the psychrometer. With the exception of the first two runs, this instrument remained off until approximately 15 minutes prior to the taking of data. This was to minimize any effects of heating of the thermistor sensor by the motor used to aspirate the psychrometer.

Analysis of Data

Mean radiant temperatures were found by the equation given by Fanger [12]:

$$MRT = \left[\sum_{i=1}^{22} f_i (T_i)^4 \right]^{1/4} \tag{28}$$

T_1 through T_{20} are the temperatures of each panel section as measured by the thermocouples. T_{21} and T_{22} are the temperatures of the air inlet and outlet, and are assumed equal to the measured dry bulb temperature. The shape factors f_i were found through the method described by Tripp, Hwang, and Crank [34].

Globe temperature and natural wet bulb temperature were calculated by trial and error from Eqns. (24), (25), (26), and (27), using measured dry bulb temperature, air velocity, and mean radiant temperature, and with the dew point temperature found from the psycrometric chart using values of the measured dry bulb and wet bulb temperatures. The computer program used to perform this operation is shown in Appendix C. Comparison of the calculated values of globe and natural wet bulb temperatures with their respective measured values is presented in the following section.

Test Number ____________________

Date ____________________ Time ____________________ Hrs.

Location ____

Ambient Temperature ____________________ °F

Barometric Pressure ____________________ inches of mercury

Chamber Conditions:

Dry Bulb Temperature ____________________ °F

Wet Bulb Temperature ____________________ °F

Dew Point Temperature ____________________ °F

Globe Temperature ____________________ °F

Natural Wet Bulb Temp. ____________________ °F

Air Velocity ____________________ ffm

Air Conditioning:

Compressor ____________________

High Side ____________________ psi

Suction Side ____________________ psi

Air Heaters ____________________

Heater #1 ____________________ (100 full scale)

Heater #2 ____________________ (140 full scale)

Humidifier ____________________

Dehumidifier ____________________

Water Panels:

Pump ____________________

Heater #1 ____________________ (280 full scale)

Heater #2 ____________________ (280 full scale)

Comments:

____

____

____

Figure 6

Sample Data Sheet

Experimental Results

The measured values of globe temperature and natural wet bulb temperature are plotted against their respective calculated values in Figures 7 and 8. Applying the Wet Bulb Globe Temperature index relation (Eqn. (5)), WBGT index values were found from both the measured and calculated values of Globe and Natural Wet Bulb Temperature. The measured and calculated values of the index are compared in Figure 9. Original data from which Figures 7-9 were constructed may be found in Appendix D.

As Figures 7-9 show, correlation of the values calculated from the governing heat and mass transfer equations with the experimentally determined values is quite good. The accuracy of the calculated and measured values, subject to assumed normally distributed instrumentation errors, is evaluated in Appendix E.

Discussion of Experimental Results

The data presented in Figure 7 shows a mean value of the natural wet bulb temperature differences (measured value-calculated value) of +0.37°F. The maximum difference was 2.7 °F. The largest deviations were the result of values obtained from the first two experimental runs. The major source of error causing these discrepancies was probably due to the fact that the water in the reservoir of the psychrometer was allowed to reach the ambient dry bulb temperature (typically about 88°F). Since no controllable mechanism for maintaining reservoir temperature was available (such as a constant temperature bath), no attempt was made to reduce the water temperature, due to the possibility of introducing an error by subcooling the liquid. Another possible point of error was heating of the wick and thermistor probe by the psychrometer motor. After the first two tests, the

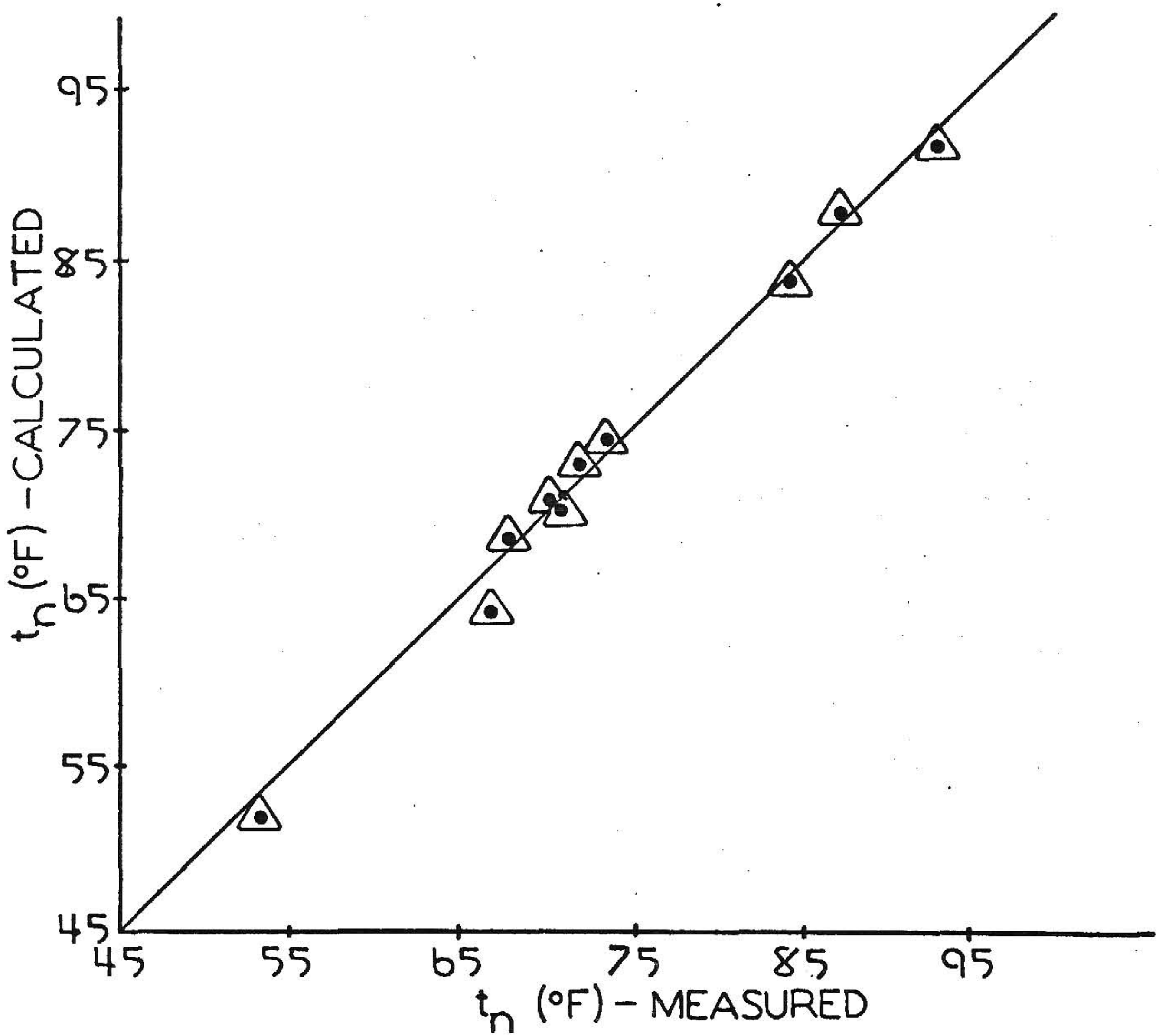

FIGURE 7

psychrometer motor was kept off until the final 15 minutes of the test to help eliminate this source of error. Measured values of natural wet bulb temperature showed good agreement with the calculated values on subsequent runs. The uncertainty due to instrument error in the measured and calculated values of natural wet bulb temperature is evaluated in Appendix E.

The mean value of the globe temperature differences was -0.11 °F, with the maximum difference being 0.93 °F. In general, the measured and calculated globe temperatures, shown in Figure 8, were in good agreement. Uncertainty due to instrument error for both the measured and calculated values of globe temperature is determined in Appendix E.

The mean difference of Wet Bulb Globe Temperature Index values, plotted in Figure 9, was 0.21 °F, with the greatest difference being 1.69 °F. Discrepancies in the measured and calculated values of the natural wet bulb and globe temperatures give rise to the differences in WBGT Index values, although the index does not reflect the full difference in each constituent due to its sensitivity to the individual errors. Evaluation of the uncertainty in the WBGT Index due to errors in the natural wet bulb and globe temperatures is found in Appendix E.

In conclusion, the experimental data shows that the relations derived for the Wet Bulb Globe Temperature Index provide a good prediction of the environmental conditions which combine to give a particular index value. Since many of the environmental conditions found in industrial situations fall within the ranges of the variables considered, the theoretical relations presented here would be suitable for use in most cases. Use of values beyond the range of those tested may be appropriate, since the theory showed good consistency over the range of conditions considered. It should be recalled here that the convection heat transfer coefficients were evaluated over a

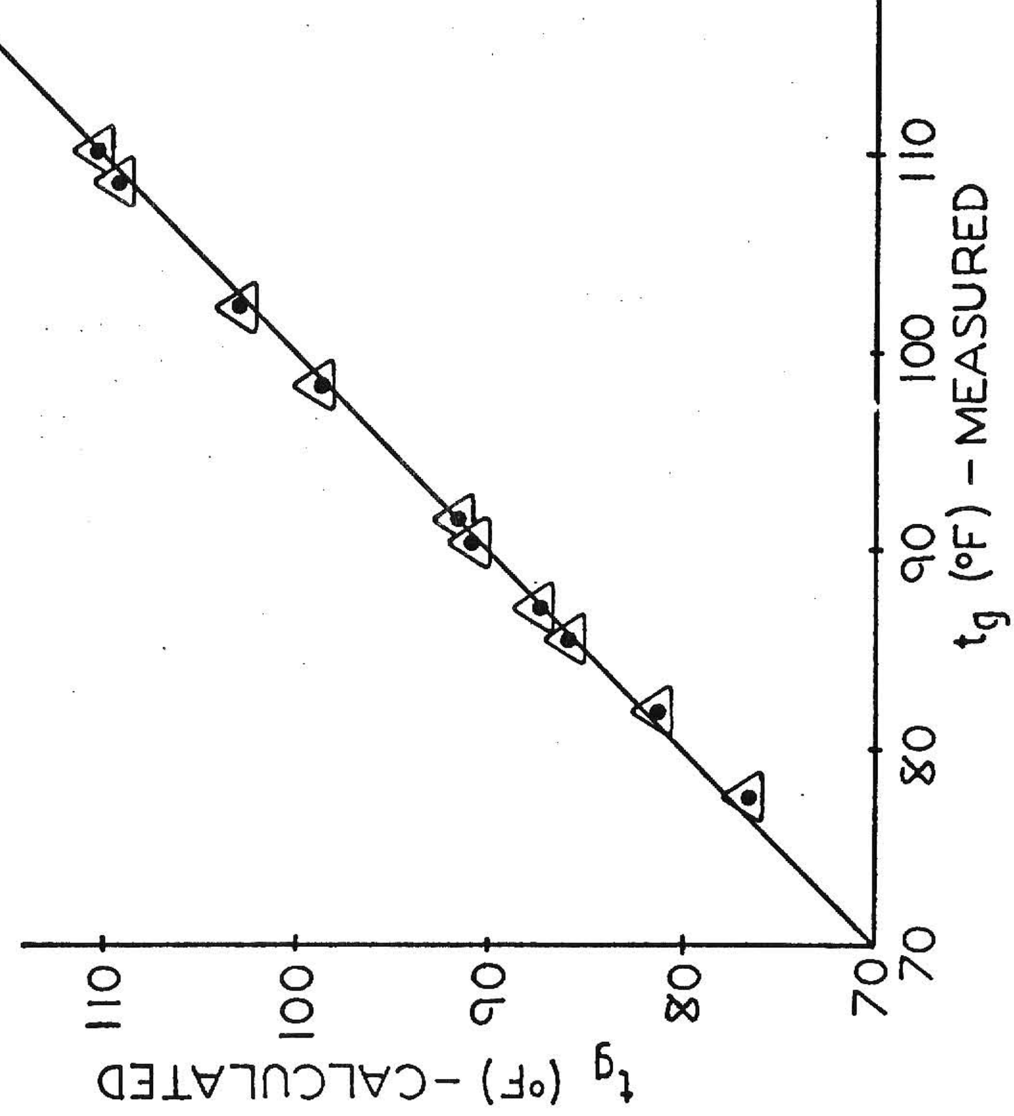

FIGURE 8

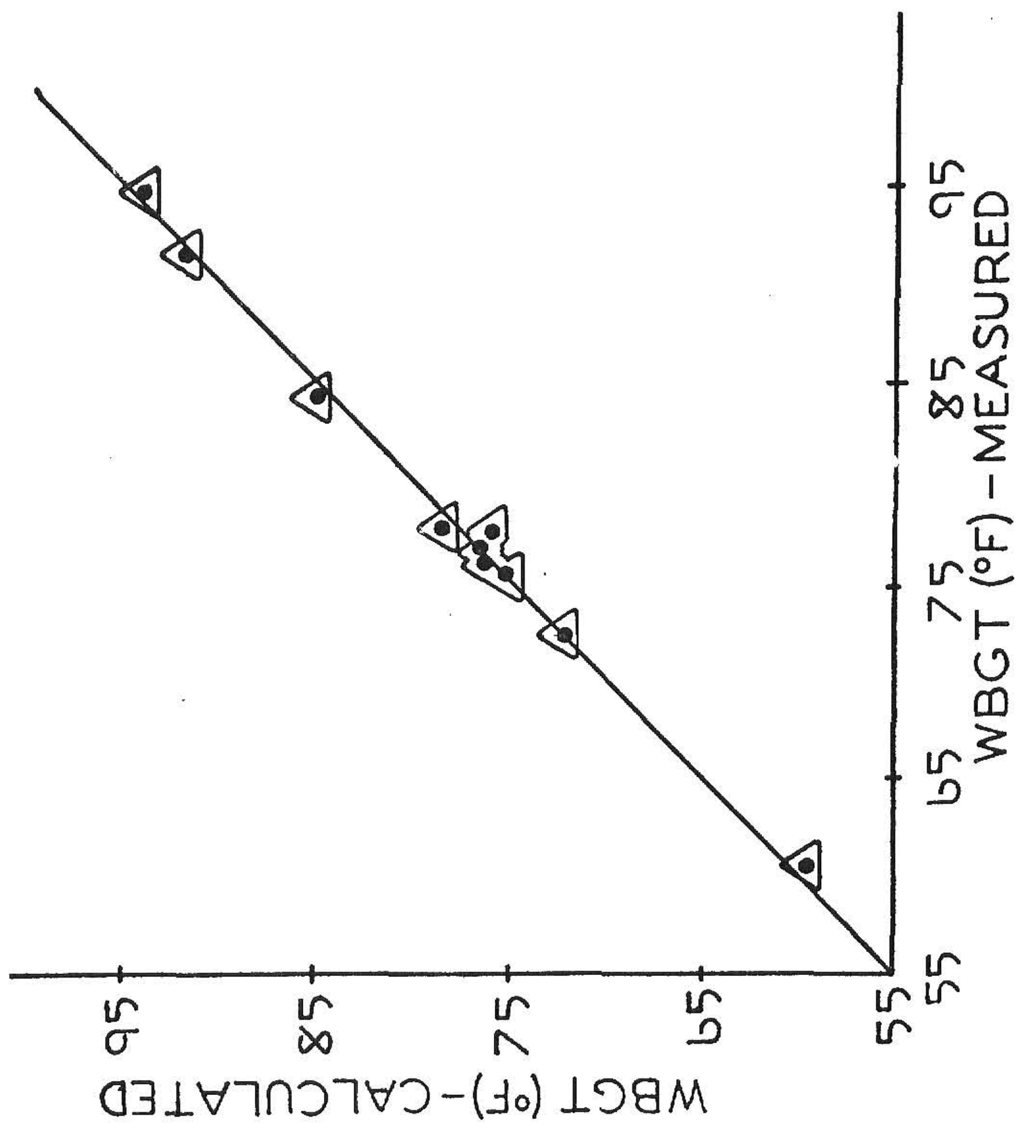

FIGURE 9

Note: WBGT values denoted "measured" and "calculated" were found using the respective values of measured and calculated natural wet bulb temperature and globe temperature and the relation

$$WBGT = 0.7t_n + 0.3t_g \quad .$$

limited range of film temperatures, and deviating from this range may result in erroneous values. The variable allowing the greatest freedom is the air velocity, but even here large deviations from the range of values tested should be avoided.

CHAPTER V

APPLICATION OF RESULTS

In order to make use of the Wet Bulb Globe Temperature Index as a design parameter, a quick and accurate method of determining the set of environmental conditions which yield the desired WBGT Index value is needed. A computer program similar to the one shown in Appendix B could be used to generate a complete set of tabulated values. This method, however, is undesirable from the standpoint that it restricts the usage of the index to those who have access to large computing machines, or those willing to pay for a large set of published tables. Even a program which produces a restricted number of values would not be desirable, since this still requires some familiarity with, and access to, computers.

The method of evaluation presented here was developed to eliminate the need for sophisticated methods of calculation. Although the assumption of one value is required, a little experience with the technique will enable a person to quickly and accurately evaluate the relations. The only tools required for the calculation are a psychrometric chart and a table of logarithms (although a slide rule or hand-held calculator would be highly desirable).

Since Eqns. (24) and (25) are both equal to t_{db}, the right hand sides can be equated.

$$t_g - 1.7298 \times 10^{-7}(MRT^4 - T_g^4)/V_a^{0.6}$$

$$= t_n - 1.7882 \times 10^{-8}(MRT^4 - T_n^4)/V_a^{0.466} + 4758.18(W_{s,n} - W_a) \qquad (29)$$

Solving the above equation for humidity ratio of the air yields:

$$W_a = W_{s,n} + 2.10164 \times 10^{-4}(t_n - t_g) + 3.6354 \times 10^{-11}(MRT^4 - T_g^4)/V_a^{0.6}$$
$$- 3.7581 \times 10^{-12}(MRT^4 - T_n^4)/V_a^{0.466} \quad (30)$$

Substituting the relationship for globe temperature in terms of WBGT and natural wet bulb temperature,

$$t_g = 3.3333\ WBGT - 2.3333\ t_n\ , \quad (31)$$

and writing T_n and V_a as

$$T_n = (460 + t_n) \quad (32)$$

and

$$V_a = (60\ U)\ ,$$

$$W_a = W_{s,n} - 7.0055 \times 10^{-4}(WBGT - t_n)$$
$$+ 3.6354 \times 10^{-11}[MRT^4 - (3.3333\ WBGT - 2.3333\ t_n + 460)^4]/(60\ U)^{0.6}$$
$$- 3.7581 \times 10^{-12}[MRT^4 - (t_n + 460)^4]/(60\ U)^{0.466} \quad (33)$$

For a given case, the design WBGT index value would be known, and the mean radiant temperature could be estimated. The design velocity would also be chosen. Since this value is the velocity of the air in the work space, it may well be only an estimate, however the insensitivity of the relation to velocity allows for some deviation in the true value without appreciable effect on the final design conditions.

By assuming a value for t_n, and reading $W_{s,n}$ from the psychrometric chart, Eqn. (33) can be solved for W_a. Then, using the assumed value of t_n and the desired value of the WBGT index, Eqn. (31) can be solved for t_g. Substituting this value and the estimated values of MRT and velocity, the dry bulb temperature may be found by Eqn. (24) (modified for conventional units of t_g in F degrees and V_a in ft/min).

$$t_{db} = t_g - 1.7298 \times 10^{-7}[MRT^4 - (t_g + 460)^4]/(60\ U)^{0.6} \tag{34}$$

Using this value of dry bulb temperature and the humidity ratio found from Eqn. (33), the first point can be located on the psychrometric chart. The process is then repeated, using a different value of t_n. Since lines of constant Wet Bulb Globe Temperature Index are straight lines when plotted on a psychrometric chart, the line passing through these two points will be the locus of dry bulb and wet bulb temperatures which yield the design WBGT index.

For greatest accuracy, t_n should be chosen to yield points at the extremes of the psychrometric chart. If Eqn. (33) yields a negative humidity ratio, the assumed value of the natural wet bult temperature should be increased. If the point located by W_a and t_{db} lies to the left of the saturation line, the value of t_n should be decreased. If the dry bulb temperature found from Eqn. (34) is above the maximum dry bulb temperature on the chart, the value of t_n should be decreased.

Once this line has been established, the most desirable dry bulb and wet bulb temperature can be selected as a basis for design. Then, equipment can be sized by an acceptable method. Since all the possible pairs of dry and wet bulb temperature are available, several choices could be made, from which the optimum design could be selected. Changes in velocity may also be considered. Although not very sensitive to this variable, the properties may be changed enough to enable the designer to choose smaller capacity equipment, or provide a margin of safety in "borderline" cases.

To illustrate the process of determining the sets of dry bulb and wet bulb temperatures, at a selected air velocity and MRT, for a particular value of WBGT, an example problem is shown in Appendix F. In addition to the solution at the specified design conditions, solutions for conditions which vary from those chosen are given to illustrate the effect of altering the values of air velocity and mean radiant temperature.

CHAPTER VI

SUMMARY

Summary of this Study

By the application of empirical relations for heat and mass transfer, the Wet Bulb Globe Temperature Index was described in terms of the four basic properties of the thermal environment. Checking these relations against experimentally determined values showed that the theoretical derivation provided a good prediction of the actual environment. The range of variables considered encompassed those conditions most commonly found in the industrial environment. The consistency of prediction over the range gives promise to the equation's ability to predict values beyond the range considered.

Rearrangement of the governing equations allowed for the application of a simple trial and error technique to determine the environmental conditions which give a particular WBGT index. Through this technique, the index can be easily used as a design parameter, allowing determination of the environmental properties which relate to selection and sizing of equipment.

Suggestions for Future Study

Investigation into actual conditions experienced in industry is necessary to determine if situations arise where the values of the environmental conditions lie far beyond the ranges of those studied here. If so, experimental confirmation of the reliability of the equations over an extended range is necessary. Also, the assumption of mean radiant temperature may prove to be a weakness. If so, establishment of standardized values of MRT through evaluation of existing industrial conditions may be needed.

Other questions arise regarding the response of the instruments which measure Wet Bulb Globe Temperature, and the correlation between the instrument's response and the response of the human body. In a personal communication between Dr. R. L. Gorton and Dr. F. N. Dukes-Dobos, it was revealed that, above 200 fpm, increased velocities had little effect on the value of natural wet bulb temperature. This is suspected to be contradictory to the response of the human body, which apparently continues to show increased cooling with increased velocities in this range. Investigation of this phenomenon is included in one phase of the research specified in a "Request for Proposal"[1] from NIOSH. If a discrepancy is shown to exist at these high velocities, re-evaluation of the governing equations, or possible restructuring of the WBGT index may be warranted.

Problems arising from asymmetric radiation of the Wet Bulb Globe Temperature measurement are discussed in a work presented by F. J. Patoile [30]. This point may also require examination.

1. NIOSH, Request for Proposal No. CDC-99-OSH-62(5). "Cooling Efficiency of Different Air Velocities in Hot Environments," Feb. 14, 1975.

REFERENCES

1. ASHRAE Handbook of Fundamentals. Published by the American Society of Heating, Refrigerating, and Air Conditioning Engineers, New York, New York, 1974.

2. Bedford, T. and C. G. Warner, "Observations of the Working Capacity of Coal Miners in Relation to Atmospheric Conditions," J. of Industrial Hygiene and Toxicology, Vol. 13, pp. 252, 1931.

3. Bedford, T. and C. G. Warner, "The Globe Thermometer in Studies of Heating and Ventilating," J. of Hygiene, Cambridge, Vol. 34, p. 458, 1934.

4. Belding, H. S., "The Search for a Universal Heat Stress Index," from Behavioral Temperature Regulation, Hardy, Gagge, and Stolwijk. Thomas, Springfield, Ill., 1970.

5. Belding, H. S. and T. F. Hatch, "Index for Evaluating Heat Stress in Terms of Resulting Physiological Strains," Heating, Piping, and Air Conditioning, Vol. 8, p. 129, Aug., 1955.

6. Botsford, J. H., "A Wet Globe Thermometer for Environmental Heat Measurement," Am. Industrial Hygiene Assoc. Journal, Vol. 32, p. 1, Jan., 1971.

7. Brief, R. S. and R. G. Confer, "Comparison of Heat Stress Indices," American Industrial Hygiene Assoc., Journal, Vol. 32, No. 1, Jan., 1971.

8. Britten, R. H. and L. R. Thompson, "A Health Survey of 10,000 Industrial Workers," U.S. Public Health Bulletin, No. 163, 1926.

9. Dill, D. B., Life, Heat, and Altitude, Harvard University Press, Cambridge, Mass., 1938.

10. Dukes-Dobos, F. N. and A. Henschel, "The Modification of the WBGT Limit for Establishing Permissible Heat Exposure Limits for Occupational Work," Health, Education, and Welfare, U.S. Public Health Service, ROSH Transaction No. 69, 1971.

11. Dukes-Dobos, F. N., A. Henschel, C. Humphres, K. Kronoveter, M. Brenner, and W. Carlson, Industrial Heat Stress -- Southern Phase, Health, Education, and Welfare, U.S. Public Health Service, Research Report No. 5, 1966.

12. Fanger, P. O., Thermal Comfort, McGraw-Hill, New York, New York, 1970.

13. Haines, G. F. and T. Hatch, "Industrial Heat Exposures -- Evaluation and Control," Heating and Ventilating, Vol. 49, p. 93, 1952.

14. Hardy, J. D., A. P. Gagge, and J. Stolwijk, Physiological and Behavioral Temperature Regulation, Thomas, Springfield, Ill., 1970.

15. Hertig, B. A. and H. S. Belding, "Evaluation of Health Hazards," from Temperature--Its Measurement and Control in Science and Industry, Vol. 3, Pt. 3, p. 347, J. D. Hardy, Ed., Reinhold, New York, N.Y., 1963.

16. Houghten, F. C. and C. P. Yaglou, "Determination of the Comfort Zone," Am. Society of Heating and Ventilating Engineers Research Report No. 673, ASHVE Transactions, Vol. 29, p. 361, 1923.

17. Hutchinson, F. W. and M. Baker, Independent Studies on Human Shape Factors, reported by Heating and Ventilating, Vol. 49, Feb., 1952.

18. Keenan, J. H. and F. G. Keyes, Thermodynamic Properties of Steam, John Wiley & Sons, New York, N.Y., 1936.

19. Kerslake, D., The Stress of Hot Environments, University Press, Cambridge, England, 1972.

20. Lee, D. H. K. and A. Henschel, "Evaluation of Thermal Environments in Shelters," U.S. Dept. of Health, Education, and Welfare, PHS Transaction 8, 1963.

21. Lee, D. H. K., "Heat and Cold Effects and Their Control," U.S. Public Health Service Monogram No. 72, Washington, D.C., 1964.

22. Leithead, C. S. and A. R. Lind, Heat Stress and Heat Disorders, Churchill, London, England, 1964.

23. McAdams, W. H., Heat Transmission, 3rd Edition, McGraw-Hill Book Co., New York, N.Y., 1954.

24. McArdle, B., W. Durham, H. Hollong, W. Ladell, W. Scott, M. Thomson, and J. Weiner, "The Prediction of the Physiological Effects of Warm and Hot Environments," Medical Research Council, London, Report No. RNP 47/391, 1947.

25. McKarns, J. S. and R. S. Brief, "Nomographs Give Refined Estimate of Heat Stress Index," Heating, Piping, and Air Conditioning, Vol. 38, No. 1, p. 113, Jan., 1966.

26. Minard, D., "Effective Temperature Scale and Its Modification," Naval Med. Res. Inst., Bethesda, Res. Report No. 6, 1964.

27. Minard, D., "Heat Casualties in the Navy and Marine Corps, 1959-1962, with Appendices on the Field Use of the Wet Bulb Globe Temperature Index," Naval Med. Res. Inst., Bethesda, Res. Report No. 7, 1964.

28. Minard, D., J. Kingston, and H. VanLiew, "Heat Stress in Working Spaces of an Aircraft Carrier," from the proceedings of the 13th International Congress on Occupational Health, p. 911, 1960.

29. NIOSH, Criteria for a Recommended Standard, Occupational Exposure to Hot Environments, U.S. Dept. of Health, Education, and Welfare, National Institute for Occupational Safety and Health, 1972.

30. Patoile, F. J., "Errors Arising from Design Limitations in Wet Bulb Globe Temperature Measuring Devices," Presented at the American Industrial Hygiene Conference, May 12-17, 1974.

31. Ramanathan, N. L. and H. S. Belding, "Physiological Evaluation of the WBGT Index for Occupational Heat Stress," Am. Industrial Hygiene Assoc. Journal, Vol. 34, No. 9, p. 375, 1973.

32. Talbott, J. H., "Heat Cramps," Medicine, Vol. 14, p. 323, 1935.

33. Threlkeld, J. L., Thermal Environmental Engineering, Prentice-Hall, Inc., Englewood Cliffs, N.J., 1962.

34. Tripp, W., C. Hwang, and R. Crank, "Radiation Shape Factors for Plane Surfaces and Spheres, Circles, or Cylinders," Kansas State University Bulletin, Vol. 46, No. 4, April, 1962.

35. Yaglou, C. P., "Abnormal Air Conditions in Industry: Their Effect on Workers and Methods of Control," J. of Industrial Hygiene and Toxicology, Vol. 19, p. 12, 1937.

36. Yaglou, C. P., "Thermal Standards in Industry; Report of the Committee on Atmospheric Comfort," Am. Journal of Public Health, Yearbook Publications, 1947.

37. Yaglou, C. P., "Estimation of Radiant Heat, Equivalent Temperature, and Effective Temperature Corrected for Radiation," Am. Journal of Public Health, Yearbook Publication, 1950.

38. Yaglou, C. P., "Thermal Standards in Industry; Report on the Committee on Atmospheric Comfort," Am. Journal of Public Health, Vol. 40, Pt. 2, 1950.

39. Yaglou, C. P. and D. Minard, "Control of Heat Casulaties," AMA Archives of Industrial Health, Vol. 16, p. 302, 1957.

APPENDIX A

Convection Heat Transfer Coefficients

Natural Wet Bulb Thermometer

$$h_{c,n} = 0.615\left(\frac{\rho_a V_a d_n}{\mu_a}\right)^{0.466}\left(\frac{k_a}{d_n}\right) \qquad (1\text{-A})$$

where $d_n = 2.083 \times 10^{-2}$ ft

TABLE 1

Properties of Moist Air and the Solution for the Convection Heat Transfer Coefficient of the Natural Wet Bulb Thermometer

Film Temp °F	ρ_a lb_m/ft^3	μ_a lb_m//hr-ft	k_a BTU/hr-ft-°F	$h_{c,n}$ (Eqn. (1-A)) BTU/hr-°F-fph
60	0.0761	0.0431	0.0143	$0.09058\ V_a^{0.466}$
70	0.0746	0.0438	0.0146	$0.09090\ V_a^{0.466}$
80	0.0731	0.0445	0.0148	$0.09065\ V_a^{0.466}$
90	0.0712	0.0450	0.0151	$0.09090\ V_a^{0.466}$
100	0.0704	0.0455	0.0154	$0.09176\ V_a^{0.466}$
110	0.0690	0.0460	0.0156	$0.09157\ V_a^{0.466}$

Mean value of $h_{c,n}$:

$$h_{c,n} = 0.09106\ V_a^{0.466}$$

Standard deviation from mean:

$$\sigma_{c,n} = \pm\ 0.54\%$$

Globe Thermometer

$$h_{c,g} = 0.37\left(\frac{\rho_a V_a d_g}{\mu_a}\right)^{0.6}\left(\frac{k_a}{d_g}\right)$$

where $d_g = 0.5$ ft

TABLE 2

Properties of Moist Air and the Solution for the Convection Heat Transfer Coefficient of the Globe Thermometer

Film Temp °F	ρ_a lb_m/ft^3	μ_a lb_m/hr-ft	k_a BTU/hr-ft-°F	$h_{c,g}$ (Eqn. (2-A)) BTU/hr-°F-fph
60	0.0761	0.0431	0.0143	$0.009819\ V_a^{0.6}$
70	0.0746	0.0438	0.0146	$0.009812\ V_a^{0.6}$
80	0.0731	0.0445	0.0148	$0.009732\ V_a^{0.6}$
90	0.0712	0.0450	0.0151	$0.009709\ V_a^{0.6}$
100	0.0704	0.0455	0.0154	$0.009770\ V_a^{0.6}$
110	0.0690	0.0460	0.0156	$0.009714\ V_a^{0.6}$
120	0.0661	0.0465	0.0158	$0.009526\ V_a^{0.6}$
130	0.0642	0.0469	0.0160	$0.009431\ V_a^{0.6}$
140	0.0622	0.0474	0.0162	$0.009310\ V_a^{0.6}$
150	0.0603	0.0482	0.0164	$0.009159\ V_a^{0.6}$
160	0.0599	0.0488	0.0167	$0.009220\ V_a^{0.6}$
170	0.0579	0.0496	0.0170	$0.009109\ V_a^{0.6}$
180	0.0557	0.0501	0.0172	$0.008949\ V_a^{0.6}$
190	0.0553	0.0504	0.0175	$0.009033\ V_a^{0.6}$
200	0.0531	0.0510	0.0178	$0.008903\ V_a^{0.6}$

Mean Value of $h_{c,g}$:

$$h_{c,g} = 0.009413\ V_a^{0.6}$$

Standard deviation from the mean:

$$\sigma_{c,g} = \pm 3.56\%$$

APPENDIX B

```
      INTEGER P,Q,R
      REAL MRT,MRTF
      READ(5,300) PBAR,VFPM
  300 FORMAT(2F10.2)
      VAIR=VFPM*60.
      N=1
      CYCLE=1
      WRITE(6,400)
      WBGT=69.
C     ***********************************************************
C     SELECTION OF WBGT INDEX VALUE
C     ***********************************************************
      DO 10 I=1,21
      WBGT=WBGT+1.
      MRT=509.67
C     ***********************************************************
C     SELECTION OF THE MEAN RADIANT TEMPERATURE
C     ***********************************************************
      DO 20 J=1,15
      MRT=MRT+10.
      TNW=49
C     ***********************************************************
C     SELECTION OF THE NATURAL WET BULB TEMPERATURE
C     ***********************************************************
      DO 30 K=1,61
      TNW=TNW+1
      TNWR=TNW+459.67
C     ***********************************************************
C     GLOBE TEMPERATURE EVALUATION BY THE WBGT INDEX RELATION
C     ***********************************************************
      TG=3.333333*WBGT-2.333333*TNW
      IF(TG.LT.59.) GO TO 20
      TGR=TG+459.67
C     ***********************************************************
C     SOLVING FOR DRY BULB TEMPERATURE
C     ***********************************************************
      TDA=TG-1.72984E-7*(MRT**4-TGR**4)/VAIR**0.6
      IF(TDA.GT.110) GO TO 30
      TDAR=TDA+459.67
C     ***********************************************************
C     SATURATION PRESSURE OF WATER VAPOR IN AIR (PSAIR),AND
C     THE SATURATION PRESSURE OF WATER VAPOR AT THE NATURAL
C     WET BULB TEMPERATURE (PSNWB),BY USE OF THE FORMULA GIVEN
C     BY SMITH,KEYES,AND GERRY
C     ***********************************************************
      PSAIR=218.167*PBAR*10**((1.-1165.07/TDAR)*(3.243718
     X+3.260144E-3*(1165.07-TDAR)+2.006581E-9
     X*(1165.07-TDAR)**3)/(1.+1.215470E-3*(1165.07-TDAR)))
      PSNWB=218.167*PBAR*10**((1.-1165.07/TNWR)*(3.243718
```

```
     X+3.260144E-3*(1165.07-TNWR)+2.006581E-9
     X*(1165.07-TNWR)**3)/(1.+1.215470E-3*(1165.07-TNWR)))
C     ****************************************************
C     HUMIDITY RATIO AT SATURATICN,NATURAL WET BULB THERMCMETER
C     ****************************************************
      WSNWB=.62198*(PSNWB)/(PBAR-PSNWB)
      TWBA=30
C     ****************************************************
C     TRIAL AND ERRCR SCLUTICN FCR CRY BULB TEMPERATURE BY
C     SUCCESSIVE REFINEMENT CF THE WET BULB TEMPERATURE
C     ESTIMATICN
C     ****************************************************
      DO 40 L=1,18
      TWBA=TWBA+10
      TWBAR=TWBA+459.67
C     ****************************************************
C     SATURATION PRESSURE CF WATER VAPOR IN AIR AT THE WET
C     BULB TEMPERATURE
C     ****************************************************
      PSAIRW=218.167*PBAR*10**((1.-1165.07/TWBAR)*(3.243781
     X+3.260144E-3*(1165.C7-TWBAR)+2.006581E-9
     X*(1165.07-TWBAR)**3)/(1.+1.215470E-3*(1165.07-TWBAR)))
C     ****************************************************
C     ACTUAL VAPOR PRESSURE,FOUND WITH CARRIER'S EQUATION
C     ****************************************************
      PWAIR=PSAIRW-((PBAR-PSAIRW)*(TCA-TWBA))/(2831-1.43*TWBA)
C     ****************************************************
C     HUMIDITY RATIO OF AIR
C     ****************************************************
      WAIR=.62198*PWAIR/(PBAR-PWAIR)
      TDANEW=4868.06*(WSNWB-WAIR)+TNW-1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466
      IF((TCA-TCANEW).GT.0.0) GC TC 5
   40 CONTINUE
    5 TWBA=TWBA-11.
C     ****************************************************
C     SUCCESSIVE REFINEMENTS
C     ****************************************************
      CO 41 M=1,11
      TWBA=TWBA+1.
      TWBAR=TWBA+459.67
      PSAIRW=218.167*PBAR*10**((1.-1165.07/TWBAR)*(3.243781
     X+3.260144E-3*(1165.07-TWBAR)+2.006581E-9
     X*(1165.07-TWBAR)**3)/(1.+1.215470E-3*(1165.07-TWBAR)))
      PWAIR=PSAIRW-((PBAR-PSAIRW)*(TCA-TWBA))/(2831-1.43*TWBA)
      WAIR=.62198*PWAIR/(PBAR-PWAIR)
      TDANEW=4868.06*(WSNWB-WAIR)+TNW-1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466
      IF((TCA-TCANEW).GT.0.0) GC TC 6
```

```
   41 CONTINUE
    6 TWBA=TWBA-1.1
      DO 42 P=1,11
      TWBA=TWBA+0.1
      TWBAR=TWBA+459.67
      PSAIRW=218.167*PBAR*10**((1.-1165.07/TWBAR)*(3.243781
     X+3.260144E-3*(1165.07-TWBAR)+2.006581E-9
     X*(1165.07-TWBAR)**3)/(1.+1.215470E-3*(1165.07-TWBAR)))
      PWAIR=PSAIRW-((PBAR-PSAIRW)*(TDA-TWBA))/(2831-1.43*TWBA)
      IF(PWAIR.LT.0.0) GO TO 30
      WAIR=.62198*PWAIR/(PBAR-PWAIR)
      TDANEW=4868.06*(WSNWB-WAIR)+TNW-1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466
      IF((TDA-TDANEW).GT.0.0) GO TO 7
   42 CONTINUE
    7 TWBA=TWBA-0.11
      DO 43 Q=1,11
      TWBA=TWBA+0.01
      TWBAR=TWBA+459.67
      PSAIRW=218.167*PBAR*10**((1.-1165.07/TWBAR)*(3.243781
     X+3.260144E-3*(1165.07-TWBAR)+2.006581E-9
     X*(1165.07-TWBAR)**3)/(1.+1.215470E-3*(1165.07-TWBAR)))
      PWAIR=PSAIRW-((PBAR-PSAIRW)*(TDA-TWBA))/(2831-1.43*TWBA)
      WAIR=.62198*PWAIR/(PBAR-PWAIR)
      TDANEW=4868.06*(WSNWB-WAIR)+TNW-1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466
      IF((TDA-TDANEW).GT.0.0) GO TO 8
   43 CONTINUE
    8 TWBA=TWBA-0.011
      DO 44 R=1,11
      TWBA=TWBA+0.001
      TWBAR=TWBA+459.67
      IF(TWBA.GT.TDA)GO TO 30
      PSAIRW=218.167*PBAR*10**((1.-1165.07/TWBAR)*(3.243781
     X+3.260144E-3*(1165.07-TWBAR)+2.006581E-9
     X*(1165.07-TWBAR)**3)/(1.+1.215470E-3*(1165.07-TWBAR)))
      PWAIR=PSAIRW-((PBAR-PSAIRW)*(TDA-TWBA))/(2831-1.43*TWBA)
      WAIR=.62198*PWAIR/(PBAR-PWAIR)
      TDANEW=4868.06*(WSNWB-WAIR)+TNW-1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466
C     ************************************************************
C     FINAL TEST OF CONVERGENCE
C     ************************************************************
      IF(ABS(TDA-TDANEW).LT.0.002) GO TO 101
   44 CONTINUE
  101 PHI=(PWAIR/PSAIR)*100
      MRTF=MRT-459.67
      WETAIR=WAIR*7000
      WETNWB=WSNWB*7000
```

```
      CYCLE=CYCLE+1
      N=N+CYCLE/30
      GO TO(1,2),N
    2 WRITE(6,400)
  400 FORMAT('1',8X,'WBGT',9X,'MRT',8X,'DRY BULB',5X,
     X'WET BULB',5X,'AIR VEL',9X,'%RH',8X,'H2O(AIR)',5X,
     X'H2O(NWB)',6X,'T-NWB',7X,'T-GLOBE')
    1 WRITE(6,200) WBGT,MRTF,TDA,TWBA,VFPM,PHI,WETAIR,
     XWETNWB,TNW,TG
  200 FORMAT('0',5F13.3,5F13.2)
      IF(N.LT.2) GO TO 30
      N=1
      CYCLE=1
   30 CONTINUE
   20 CONTINUE
   10 CONTINUE
      STOP
      END
```

APPENDIX C

```
      DIMENSION T(20)
      REAL MRT,MRTF
      INTEGER P,Q,R,S,Z
      READ(5,100) N
  100 FORMAT(I2)
      WRITE(6,1)
    1 FORMAT('1')
      DO 10 I=1,N
      READ(5,200) TDA,TDP,VFPM,TG,TNWB,PBAR
  200 FORMAT(6F10.3)
      READ(5,201)(T(Z),Z=1,20)
  201 FORMAT(10F8.3)
      TDAR=TDA+459.67
C     ****************************************************************
C     CALCULATION OF MEAN RADIANT TEMPERATURE FROM INDIVIDUAL
C     PANEL TEMPERATURES
C     ****************************************************************
      MRT=(0.0235704*(T(1)**4+T(3)**4+T(4)**4+T(6)**4+T(15)**4
     X+T(17)**4+T(18)**4+T(20)**4)+0.040526*(T(7)**4+T(8)**4
     X+T(9)**4+T(10)**4+T(11)**4+T(12)**4+T(13)**4+T(14)**4)
     X+0.0348603*(T(2)**4+T(5)**4+T(16)**4+T(19)**4)
     X+0.3477818*TDAR**4)**0.25
      VAIR=VFPM*60.
      TGCAL=50.
C     ****************************************************************
C     TRIAL AND ERROR DETERMINATION OF GLOBE TEMPERATURE
C     ****************************************************************
      DO 1000 J=1,20
      TGCAL=TGCAL+10.
      TGCALR=TGCAL+459.67
      TDANEW=TGCAL-(1.72984E-7)*(MRT**4-TGCALR**4)/VAIR**0.6
      IF((TDA-TDANEW).LT.0.0) GO TO 20
 1000 CONTINUE
   20 TGCAL=TGCAL-11.
      DO 1001 K=1,11
      TGCAL = TGCAL+1.
      TGCALR=TGCAL+459.67
      TDANEW=TGCAL-(1.72984E-7)*(MRT**4-TGCALR**4)/VAIR**0.6
      IF((TDA-TDANEW).LT.0.0) GO TO 30
 1001 CONTINUE
   30 TGCAL=TGCAL-1.1
      DO 1002 L=1,11
      TGCAL=TGCAL+0.1
      TGCALR=TGCAL+459.67
      TDANEW=TGCAL-(1.72984E-7)*(MRT**4-TGCALR**4)/VAIR**0.6
      IF((TDA-TDANEW).LT.0.0) GO TO 40
 1002 CONTINUE
   40 TGCAL=TGCAL-0.11
      DO 1003 M=1,11
```

```
      TGCAL=TGCAL+0.01
      TGCALR=TGCAL+459.67
      TDANEW=TGCAL-(1.72984E-7)*(MRT**4-TGCALR**4)/VAIR**0.6
      IF(ABS(TDA-TDANEW).LT.0.02) GO TO 50
 1003 CONTINUE
C     **********************************************************
C     TRIAL AND ERROR DETERMINATION OF DEW POINT TEMPERATURE
C     **********************************************************
   50 TDPR=TDP+459.67
      PAIR =218.167*PBAR*10**((1.-1165.07/TDPR)*(3.243781
     X+3.260144E-3*(1165.07-TDPR)+2.006581E-9*(1165.07-TDPR)**3)
     X/(1.+1.215470E-3*(1165.07-TDPR)))
      WAIR=0.62198*PAIR/(PBAR-PAIR)
      TNWCAL=40.
      DO 2000 P=1,20
      TNWCAL=TNWCAL+10.
      TNWR=TNWCAL+459.67
      PSNWB=218.167*PBAR*10**((1.-1165.07/TNWR)*(3.243781
     X+3.260144E-3*(1165.07-TNWR)+2.006581E-9*(1165.07-TNWR)**3)
     X/(1.+1.215470E-3*(1165.07-TNWR)))
      WSNWB=0.62198*PSNWB/(PBAR-PSNWB)
      TDANEW=4868.06*(WSNWB-WAIR)-(1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466)+TNWCAL
      IF((TDA-TDANEW).LT.0.0) GO TO 60
 2000 CONTINUE
   60 TNWCAL=TNWCAL-11.
      DO 2001 Q=1,11
      TNWCAL=TNWCAL+1.0
      TNWR=TNWCAL+459.67
      PSNWB=218.167*PBAR*10**((1.-1165.07/TNWR)*(3.243781
     X+3.260144E-3*(1165.07-TNWR)+2.006581E-9*(1165.07-TNWR)**3)
     X/(1.+1.215470E-3*(1165.07-TNWR)))
      WSNWB=0.62198*PSNWB/(PBAR-PSNWB)
      TDANEW=4868.06*(WSNWB-WAIR)-(1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466)+TNWCAL
      IF((TDA-TDANEW).LT.0.0) GO TO 70
 2001 CONTINUE
   70 TNWCAL=TNWCAL-1.1
      DO 2002 R=1,11
      TNWCAL=TNWCAL+0.1
      TNWR=TNWCAL+459.67
      PSNWB=218.167*PBAR*10**((1.-1165.07/TNWR)*(3.243781
     X+3.260144E-3*(1165.07-TNWR)+2.006581E-9*(1165.07-TNWR)**3)
     X/(1.+1.215470E-3*(1165.07-TNWR)))
      WSNWB=0.62198*PSNWB/(PBAR-PSNWB)
      TDANEW=4868.06*(WSNWB-WAIR)-(1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466)+TNWCAL
      IF((TDA-TDANEW).LT.0.0) GO TO 80
 2002 CONTINUE
```

```
   80 TNWCAL=TNWCAL-0.11
      DO 2003 S=1,11
      TNWCAL=TNWCAL+0.01
      TNWR=TNWCAL+459.67
      PSNWB=218.167*PBAR*1C**((1.-1165.07/TNWR)*(3.243781
     X+3.260144E-3*(1165.07-TNWR)+2.006581E-9*(1165.07-TNWR)**3)
     X/(1.+1.215470E-3*(1165.07-TNWR)))
      TDANEW=4868.06*(WSNWB-WAIR)-(1.78816E-8*(MRT**4-TNWR**4)
     X/VAIR**0.466)+TNWCAL
      IF(ABS(TCA-TDANEW).LT.0.02) GC TO 90
 2003 CCNTINUE
   90 WBGTCL=C.7*TNWCAL+0.3*TGCAL
      WBGT=C.7*TNWB+0.3*TG
      MRTF=MRT-459.67
      WRITE(6,300)I,TCA,TDP,VFPM,MRTF,PBAR,TNWCAL,TNWB
     X,TGCAL,TG,WBGTCL,WBGT
  300 FORMAT(/////'0',15X,'TEST NUMBER',I3
     X/' ',15X,'CHAMBER CONDITIONS:'
     X/' ',25X,'DRY BULB TEMPERATURE',2X,F6.2,' DEG.F'/' ',
     X25X,'DEW POINT TEMPERATURE',2X,F6.2,' DEG.F'/' ',25X,
     X'AIR VELOCITY',2X,F5.1,' FPM'/' ',25X,
     X'MEAN RADIANT TEMPERATURE',2X,F5.1,' DEG.F'/' ',25X,
     X'BAROMETRIC PRESSURE',2X,F6.3,' PSIA'/' ',15X,
     X'NATURAL WET BULB TEMPERATURE:'/' ',25X,
     X'CALCULATED',2X,F6.2,' DEG.F'/' ',25X,
     X'MEASURED',2X,F6.2,' DEG.F'/' ',15X,'GLOBE TEMPERATURE:'
     X/' ',25X,'CALCULATED',2X,F6.2,' DEG.F'/' ',25X,
     X'MEASURED',2X,F6.2,' DEG.F'/' ',15X,
     X'WET BULB GLOBE TEMPERATURE:'/' ',25X,
     X'CALCULATED',2X,F6.2,' DEG.F'/' ',25X,'MEASURED',
     XF6.2,2X,' DEG.F'//)
   10 CCNTINUE
      STCP
      END
```

APPENDIX D

```
TEST NUMBER  1
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE   70.25 DEG.F
          DEW POINT TEMPERATURE   26.70 DEG.F
          AIR VELOCITY   95.0 FPM
          MEAN RADIANT TEMPERATURE   86.4 DEG.F
          BAROMETRIC PRESSURE  14.376 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   51.90 DEG.F
          MEASURED   53.24 DEG.F
GLOBE TEMPERATURE:
          CALCULATED   76.37 DEG.F
          MEASURED   77.30 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   59.24 DEG.F
          MEASURED 60.46   DEG.F
```

```
TEST NUMBER  2
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE  103.54 DEG.F
          DEW POINT TEMPERATURE   27.50 DEG.F
          AIR VELOCITY   95.0 FPM
          MEAN RADIANT TEMPERATURE  102.0 DEG.F
          BAROMETRIC PRESSURE  14.376 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   64.20 DEG.F
          MEASURED   66.85 DEG.F
GLOBE TEMPERATURE:
          CALCULATED  102.91 DEG.F
          MEASURED  102.20 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   75.81 DEG.F
          MEASURED 77.45   DEG.F
```

```
TEST NUMBER  3
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE   86.96 DEG.F
          DEW POINT TEMPERATURE   65.80 DEG.F
          AIR VELOCITY   95.0 FPM
          MEAN RADIANT TEMPERATURE   97.0 DEG.F
          BAROMETRIC PRESSURE  14.200 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   72.92 DEG.F
          MEASURED   72.04 DEG.F
GLOBE TEMPERATURE:
          CALCULATED   90.93 DEG.F
          MEASURED   90.62 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   78.32 DEG.F
          MEASURED 77.61   DEG.F
```

```
TEST NUMBER  4
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE   75.83 DEG.F
          DEW POINT TEMPERATURE   63.20 DEG.F
          AIR VELOCITY  120.0 FPM
          MEAN RADIANT TEMPERATURE   91.5 DEG.F
          BAROMETRIC PRESSURE  14.200 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   68.30 DEG.F
          MEASURED   68.04 DEG.F
GLOBE TEMPERATURE:
          CALCULATED   81.36 DEG.F
          MEASURED   81.75 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   72.22 DEG.F
          MEASURED 72.15   DEG.F
```

```
TEST NUMBER  5
CHAMBER CCNCITIONS:
          DRY BULB TEMPERATURE   94.72 DEG.F
          DEW POINT TEMPERATURE   9C.70 DEG.F
          AIR VELCCITY   85.0 FPM
          MEAN RACIANT TEMPERATLRE  103.9 DEG.F
          BARCMETRIC PRESSLRE  14.175 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   91.90 DEG.F
          MEASUREC   92.97 DEG.F
GLCBE TEMPERATURE:
          CALCULATED   98.59 DEG.F
          MEASUREC   98.10 CEG.F
WET BULB GLCBE TEMPERATURE:
          CALCULATED   93.91 DEG.F
          MEASURED 94.51   CEG.F
```

```
TEST NUMBER  6
CHAMBER CCNDITIONS:
          DRY BULB TEMPERATURE  106.00 DEG.F
          DEW POINT TEMPERATURE   75.30 CEG.F
          AIR VELCCITY   75.0 FPM
          MEAN RACIANT TEMPERATLRE  113.0 DEG.F
          BAROMETRIC PRESSLRE  14.185 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   83.60 CEG.F
          MEASURED   83.80 DEG.F
GLCBE TEMPERATURE:
          CALCULATED  109.17 DEG.F
          MEASUREC  108.56 DEG.F
WET BULB GLCBE TEMPERATURE:
          CALCULATED   91.27 DEG.F
          MEASUREC 91.23   CEG.F
```

```
TEST NUMBER  7
CHAMBER CCNCITIONS:
          DRY BULB TEMPERATURE   83.67 DEG.F
          DEW POINT TEMPERATURE   65.50 DEG.F
          AIR VELCCITY  190.0 FPM
          MEAN RACIANT TEMPERATLRE   95.4 DEG.F
          BARCMETRIC PRESSURE  14.264 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   71.53 DEG.F
          MEASURED   71.46 DEG.F
GLCBE TEMPERATURE:
          CALCULATEC   87.15 DEG.F
          MEASUREC   87.07 DEG.F
WET BULB GLCBE TEMPERATURE:
          CALCULATED   76.22 DEG.F
          MEASURED 76.14   DEG.F
```

```
TEST NUMBER  8
CHAMBER CCNCITIONS:
          DRY BULB TEMPERATURE   86.19 DEG.F
          DEW POINT TEMPERATURE   60.80 DEG.F
          AIR VELCCITY   70.0 FPM
          MEAN RADIANT TEMPERATLRE   98.4 DEG.F
          BARCMETRIC PRESSURE  14.298 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   70.34 DEG.F
          MEASUREC   70.31 DEG.F
GLCBE TEMPERATURE:
          CALCULATED   91.56 DEG.F
          MEASUREC   91.52 DEG.F
WET BULB GLCBE TEMPERATURE:
          CALCULATED   76.71 DEG.F
          MEASUREC 76.67   DEG.F
```

```
TEST NUMBER  9
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE   85.26 DEG.F
          DEW POINT TEMPERATURE   63.30 DEG.F
          AIR VELOCITY  115.0 FPM
          MEAN RADIANT TEMPERATURE   86.4 DEG.F
          BAROMETRIC PRESSURE  14.298 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   70.60 DEG.F
          MEASURED   70.76 DEG.F
GLOBE TEMPERATURE:
          CALCULATED   85.67 DEG.F
          MEASURED   85.61 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   75.12 DEG.F
          MEASURED 75.21   DEG.F
```

```
TEST NUMBER 10
CHAMBER CONDITIONS:
          DRY BULB TEMPERATURE   86.05 DEG.F
          DEW POINT TEMPERATURE   63.60 DEG.F
          AIR VELOCITY  115.0 FPM
          MEAN RADIANT TEMPERATURE  144.5 DEG.F
          BAROMETRIC PRESSURE  14.278 PSIA
NATURAL WET BULB TEMPERATURE:
          CALCULATED   74.10 DEG.F
          MEASURED   73.52 DEG.F
GLOBE TEMPERATURE:
          CALCULATED  110.06 DEG.F
          MEASURED  109.98 DEG.F
WET BULB GLOBE TEMPERATURE:
          CALCULATED   84.89 DEG.F
          MEASURED 84.46   DEG.F
```

APPENDIX E

Instrument Specifications and Error Analysis

Since the magnitude of the errors in the measurements differed across the range of each variable, the maximum error within the range of each value was used in this analysis. The same is true for the sensitivities. Values specified for error are limits of error (95% confidence interval).

Instrument Specifications

Thermocouples:

Copper/Constantan (ANSI Type T)

Error; $\pm$ 1-1/2 °F (-75 °F to 200 °F range)

Thermistors:

YSI 400 series thermistors

Error: $\pm$ 0.18 °F (32 °F to 175 °F range)

Digital Thermister Thermometer

United Systems Corporation

Digitec Model 500-1

Error: Instrument $\pm$ 0.15 °F

Resolution $\pm$ 0.01 °F

Total $\pm$ 0.15 °F

Digital Thermocouple Thermometer

Beckman 4011 RVP with scanner and printer

Error: Negligable

Air Velocity:

Anemotherm Air Meter

Error: $\pm$ 10 fpm

Error Analysis

The sensitivities to error of the natural wet bulb temperature were determined by a numerical analysis with the aid of a digital computer. These values are summarized below. (The subscript indicates the variable under consideration.)

$$S_{t_{db}} = 0.499 \text{ °F/°F}$$

$$S_{MRT} = -0.123 \text{ °F/°R}$$

$$S_{V_a} = 0.0268 \text{ °F/fpm}$$

$$S_{W_a} = 2.357 \text{ °F/0.001 } \frac{lb_m H_2O}{lb_m air}$$

The error in the value of natural wet bulb temperature calculated from the measured environmental conditions was determined by the following relation (where λ indicates the limit of error of the value)

$$\lambda_{t_n} = [(S_{t_{db}} \lambda_{t_{db}})^2 + (S_{MRT}\lambda_{MRT})^2 + (S_{V_a} \lambda_{V_a})^2 + (S_{W_a} \lambda_{W_a})^2]^{1/2} \qquad \text{(E-1)}$$

The values for $\lambda_{t_{db}}$ and λ_{V_a} were taken directly from the thermistor and digital thermometer specifications (rms value). The value for λ_{W_a} was determined from the psychrometric chart using the errors specified for the dry bulb and wet bulb temperatures (obtained with thermistor probes).

$$\lambda_{W_a} = \pm\ 0.0003 \frac{lb_m\ H_2O}{lb_m\ air}$$

The value for the error in the mean radiant temperature was determined by

$$\lambda_{MRT} = \sum_{i=1}^{22} (S_{T,i}\lambda_i)^{2\ 1/2} \qquad \text{(E-2)}$$

where the sensitivities $S_{T,i}$ were found through:

$$S_{T,i} = \frac{\frac{\partial f(MRT)}{\partial T_i}(T_i)}{f(MRT)} \tag{E-3}$$

The function f(MRT) is given by Eqn. (28).

The limit of error in the MRT measurement was found to be:

$$\lambda_{MRT} = \pm\ 0.70\ °R$$

Substituting the sensitivities and errors into Eqn. (E-1) yielded

$$\lambda_{t_n} = \pm\ 0.77\ °F$$

The limit of error for the measured value of natural wet bulb temperature, λ'_{t_n}, was due to the error in the thermistor probe and digital thermometer:

$$\lambda'_{t_n} = 0.23\ °F$$

The sensitivities to error of the globe temperature were found by the same method as for the natural wet bulb temperature.

$$S_{t_{db}} = 0.966\ °F/°F$$

$$S_{MRT} = 0.461\ °F/°R$$

$$S_{V_a} = -0.0356\ °F/fpm$$

The error in the calculated globe temperature due to error in the measured variables was found by

$$\lambda_{t_g} = [(S_{t_{db}}\lambda_{t_{db}})^2 + (S_{MRT}\lambda_{MRT})^2 + (S_{V_a}\lambda_{V_a})^2]^{1/2} \tag{E-4}$$

The values of $\lambda_{t_{db}}$, λ_{MRT}, and λ_{V_a} being the same as those used in determining λ_{t_n}, Eqn. (E-4) yielded

$$\lambda_{t_g} = \pm\ 0.53\ °F$$

The error arising in the measured globe temperature, resulting from thermistor and digital thermometer error, was

$$\lambda'_{t_g} = \pm\ 0.23\ °F$$

The limit of error of the calculated value of the Wet Bulb Globe Temperature Index was found by

$$\lambda_{WBGT} = [(S_{t_n} \lambda_{t_n})^2 + (S_{t_g} \lambda_{t_g})^2]^{1/2} \qquad (E\text{-}5)$$

The sensitivities S_{t_n} and S_{t_g} are given by:

$$S_{t_n} = \frac{\frac{\partial f(WBGT)}{\partial t_n} t_n}{f(WBGT)} \qquad (E\text{-}6)$$

and

$$S_{t_g} = \frac{\frac{\partial f(WBGT)}{\partial t_g} t_g}{f(WBGT)} \qquad (E\text{-}7)$$

where

$$f(WBGT) = 0.7\ t_n + 0.3\ t_g \qquad (E\text{-}8)$$

Using t_n = 70°F and t_g = 100 °F, the sensitivities were found to be

$$S_{t_n} = 0.620$$

$$S_{t_g} = 0.380$$

The resulting error in the calculated value of WBGT was:

$$\lambda_{WBGT} = \pm\ 0.52\ °F$$

Similarly, the error in the measured value of the Wet Bulb Globe Temperature Index was found to be:

$$\lambda'_{WBGT} = \pm\ 0.17\ °F$$

APPENDIX F

Example Problem

Situation:

Light assembly area, power hand tools used.

Limited movement of workers.

No heavy equipment in area.

Design Criteria:

Light assembly work requires a moderate work level (about an 800 BTU/hr metabolic rate). For continuous 8-hr work shifts, the recommended WBGT Index limit is 82 °F. To provide a factor of safety, use WBGT = 80°F as the design value. With light equipment, assume MRT = 100°F. Since workers are stationary, a fairly low velocity is desirable for comfort. Use 100 fpm as the design velocity.

Determination of Air Properties:

Substituting the design conditions into Eqn. (33), where WBGT and t_g are in °F, air velocity is in fpm, W_s and $W_{s,n}$ are in $lb_m H_2O/lb_m air$ and MRT is in °R:

$$W_a = W_{s,n} - 7.0055 \times 10^{-4} (80 - t_n)$$

$$+ 3.6354 \times 10^{-11}[(9.811 \times 10^{10}) - (266.67 - 2.333\ t_n + 460)^4]/184.88$$

$$- 3.7581 \times 10^{-12}[(9.811 \times 10^{10}) - (t_n + 460)^4]/57.63 \qquad \text{(F-1)}$$

Assumption for t_n:

$$t_n = 65\ °F$$

from the psychrometric chart at the saturation line:

$$W_{s,n} = 0.0133 \ lb_m H_2O/lb_m air$$

From equation (F-1),

$$W_a = -0.00086 \ lb_m H_2O/lb_m D.A.$$

Since W_a is negative, select a higher value of t_n.

$$\text{Let } t_n = 70 \ °F$$

from the psychrometric chart:

$$W_{s,n} = 0.0158 \ lb_m H_2O/lb_m D.A.$$

From Equation (F-1),

$$W_a = 0.0070 \ lb_m H_2O/lb_m D.A.$$

Substituting the design conditions into Equation (34):

$$t_{db} = t_g - 1.7298 \times 10^{-7}[(9.811 \times 10^{10}) - (t_g + 460)^4]/184.88 \quad \text{(F-2)}$$

For the assumed natural wet bulb temperature and the design WBGT, Globe temperature can be found. Using Equation (23):

$$t_g = 3.3333 \ WBGT - 2.3333 \ t_n \quad (23)$$

which yields

$$t_g = 103.33 \ °F$$

Thus, from (F-2)

$$t_{db} = 105.7 \ °F$$

This point may now be plotted on the psychrometric chart. This is done in Figure 10.

Since this point is located at the lower right-hand side of the chart (hot and dry), the next assumption of t_n should be chosen higher in order to obtain a point near the saturation line. Assume

$$t_n = 75\ °F$$

From the psychrometric chart

$$W_{s,n} = 0.0187\ \frac{lb_m\ H_2O}{lb_m\ air}$$

Solving Equations (F-1) and (F-2) yields:

$$W_a = 0.0152\ \frac{lb_m\ H_2O}{lb_m\ air}$$

$$t_{db} = 96.5\ °F$$

After locating this point on the psychrometric chart, a line passing through both points will yield all the possible dry bulb and wet bulb temperatures. This is done on Figure 10.

It is advisable to check the effects of deviation from the design conditions. To demonstrate these effects, 80 °F WBGT lines are plotted in Figure 11 for different values of velocity, and in Figure 12 for different values of MRT. It can be seen that a change in air velocity has little effect on the allowable dry bulb and wet bulb temperatures, differing by amounts that should still insure an allowable environment due to the factor of safety used.

From Figure 12, however, it can be seen that as MRT increases, the values dry bulb and wet bulb required to maintain the same level of heat stress must be decreased. If MRT is underestimated, the air conditioning equipment would not maintain the temperature level required to provide the desired WBGT Index. Conversely, if MRT is overestimated, the design would yield equipment which is larger than necessary.

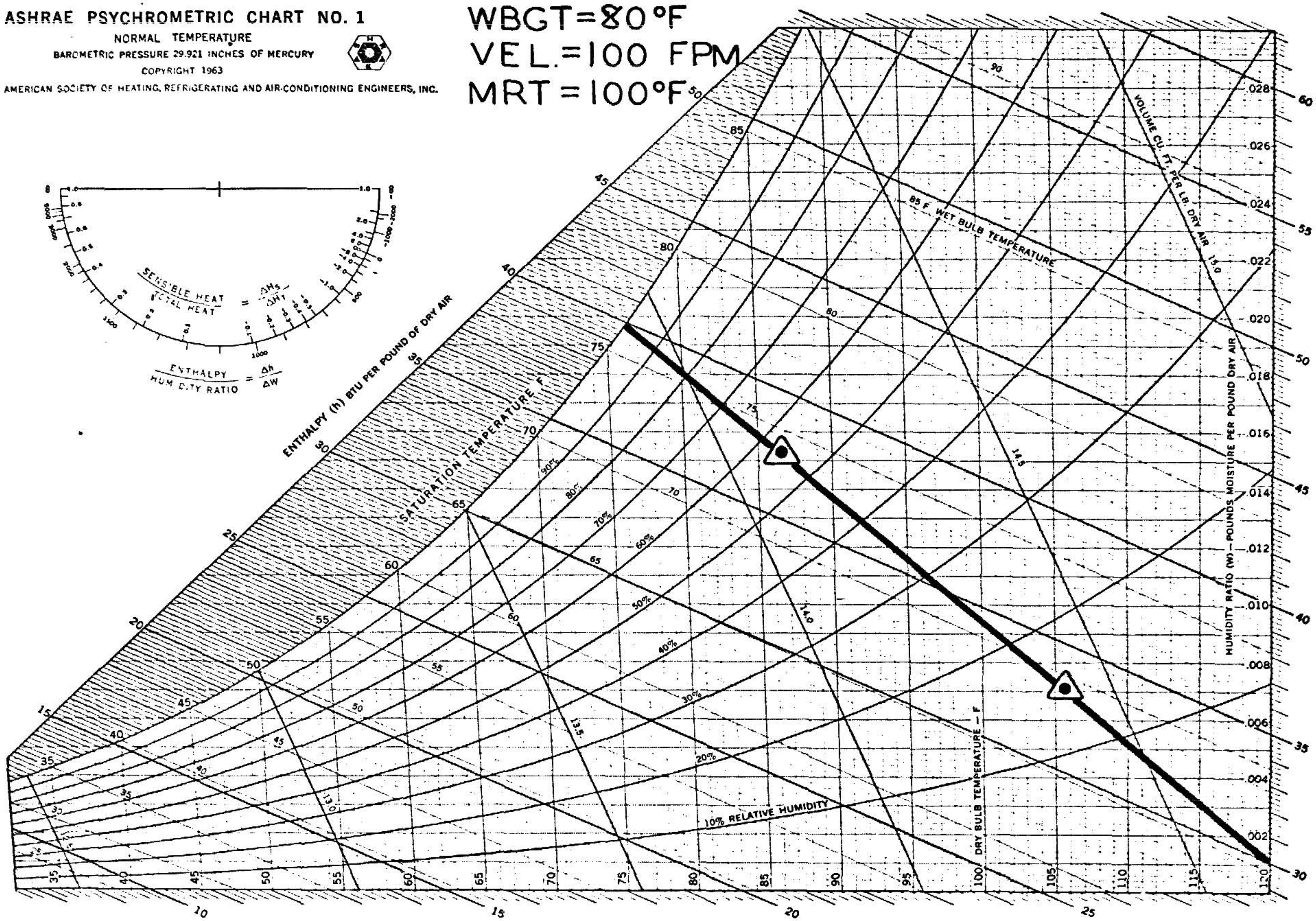

FIGURE 10

chart courtesy of ASHRAE

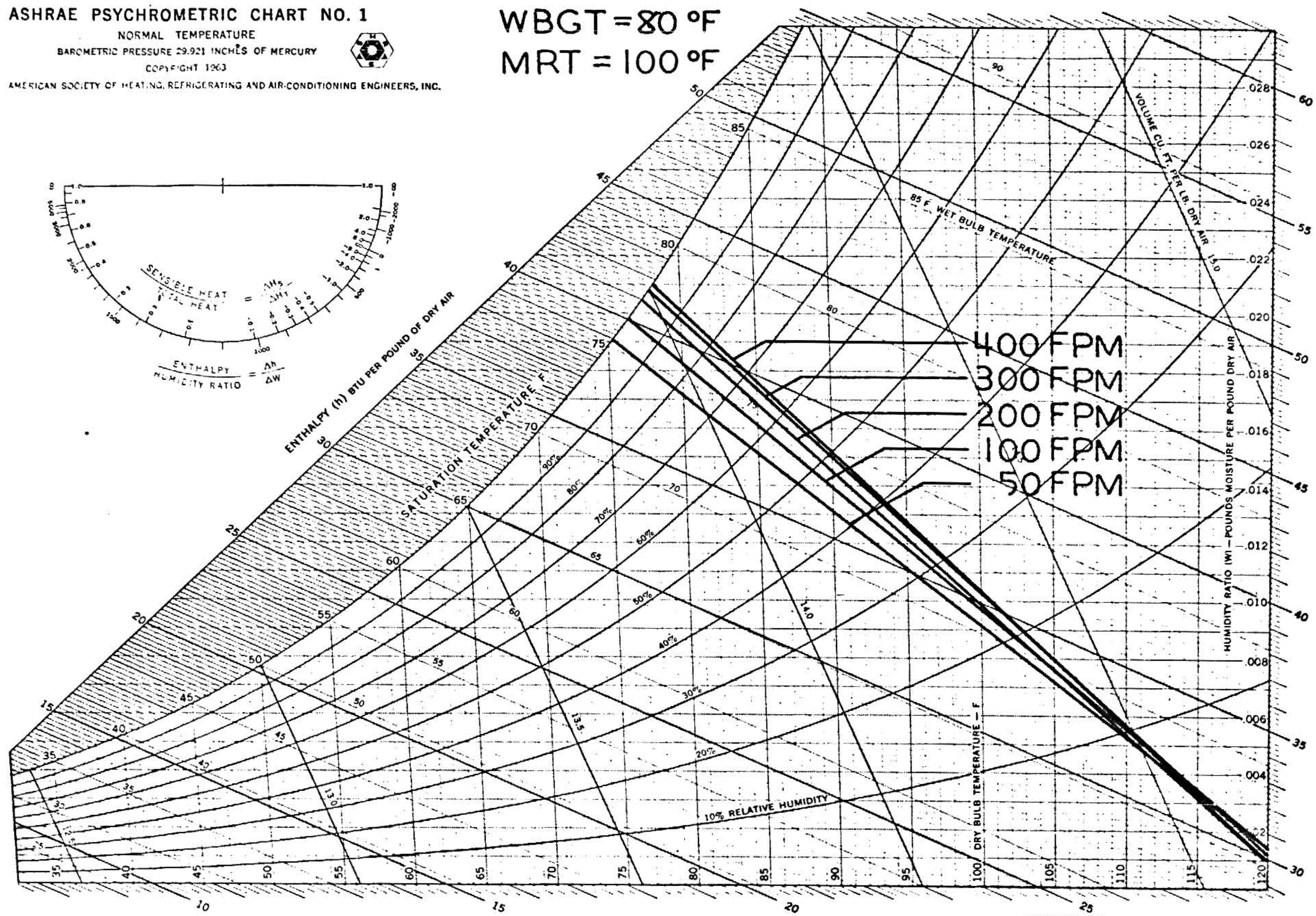

FIGURE 11

chart courtesy of ASHRAE

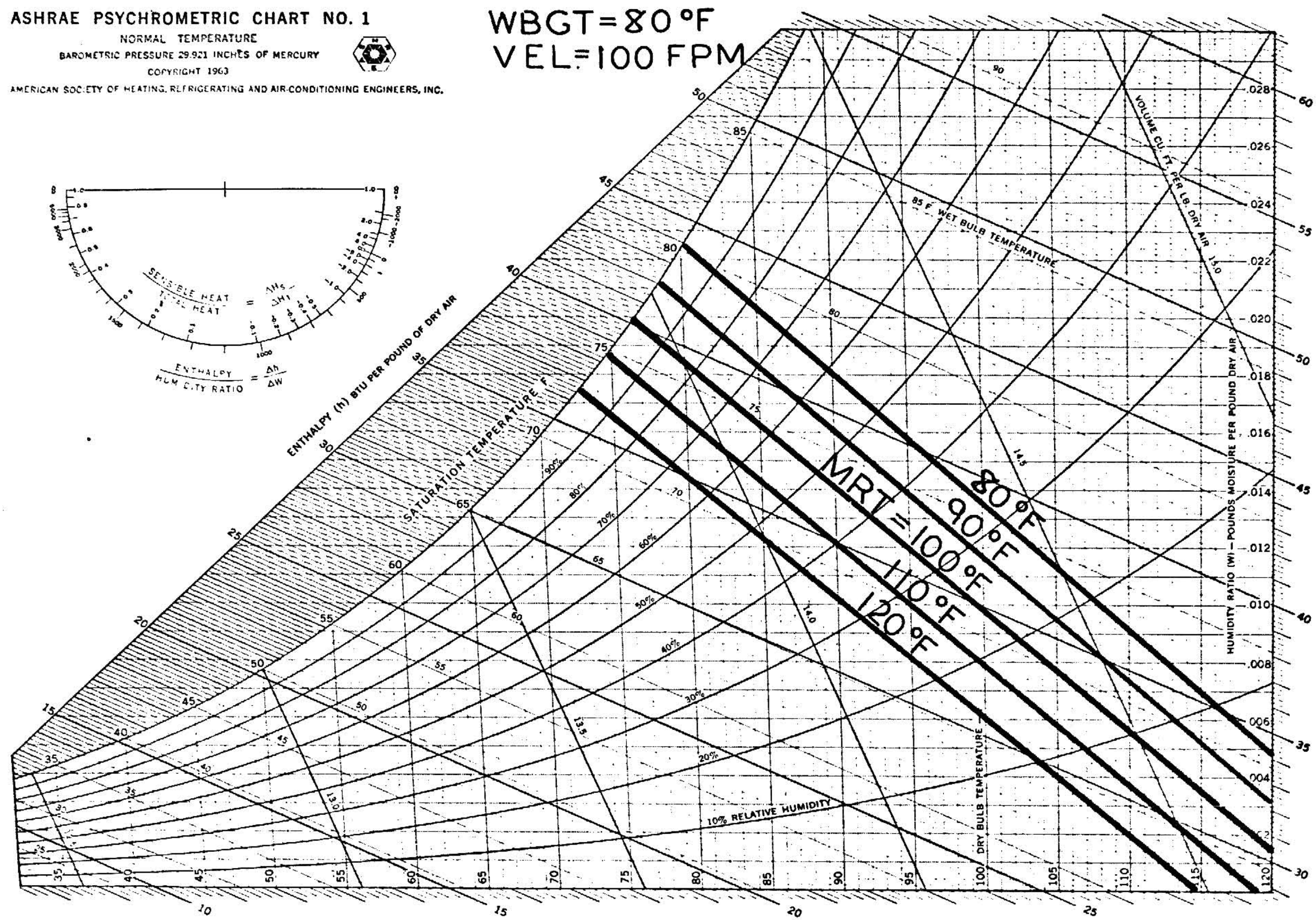

FIGURE 12

chart courtesy of ASHRAE

ACKNOWLEDGEMENTS

I wish to express my gratitude to the committee members, each of whom were influential in both my graduate and undergraduate studies; Dr. R. L. Gorton, who acted as my major professor, Dr. H. D. Ball, Prof. R. E. Crank, and Dr. E. C. Lindly.

I would also like to thank Mr. Thomas E. Shrimplin, Mr. Gerald E. Lockhart, and Mr. Charles E. Miller for their help in constructing the experimental apparatus.

Finally, I would like to thank my wife, Mary, for her unfailing support and encouragement throughout my entire study.

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A DIRECT DETERMINATION OF THE WET BULB GLOBE TEMPERATURE IN TERMS OF ENVIRONMENTAL PROPERTIES

by

CHARLES DANIEL SULLIVAN

B.S., Kansas State University, 1974

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the

requirements of the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

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1975

ABSTRACT

The purpose of this study was to relate the Wet Bulb Globe Temperature Index to the four basic climatic factors; dry bulb temperature, wet bulb temperature, air velocity, and mean radiant temperature. The ranges of variables considered were:

WBGT Index Value	70 °F to 90 °F
Dry Bulb Temperature	70 °F to 110 °F
Wet Bulb Temperature	45 °F to 85 °F
Mean Radiant Temperature	80 °F to 150 °F
Air Velocity	50 fpm to 200 fpm

The equations derived were based on empirical relations for heat and mass transfer. By comparison with experimentally obtained values, these equations were shown to properly predict the climatic factors which yield a particular WBGT Index.

A technique is presented which allows for conversion of the Wet Bulb Globe Temperature Index into the basic climatic factors. The WBGT Index can then be used as a basis for design and control of air conditioning systems.