NIOSH

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Evaluation of a Single-Reading Heat Stress Instrument

EVALUATION OF A SINGLE-READING HEAT STRESS INSTRUMENT

Robert T. Hughes

U. S. DEPARTMENT OF HEALTH, EDUCATION, AND WELFARE
Public Health Service
Center for Disease Control
National Institute for Occupational Safety and Health
Division of Laboratories and Criteria Development
Cincinnati, Ohio 45202

March 1975

HEW Publication No.(NIOSH) 75–148

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ABSTRACT

A heat stress instrument which integrates radiation, wet bulb and dry bulb temperature, air velocity and a simulation of metabolic rate into a single output reading was evaluated. Preliminary evaluation indicated the instrument not suitable for field heat stress measurements.

Test results are discussed and recommendations regarding heat stress instrumentation development are provided.

EVALUATION OF A SINGLE READING HEAT STRESS INSTRUMENT

INTRODUCTION

A single reading heat stress instrument was proposed by Professor Hatch of the Department of Labor (DOL) Heat Stress Advisory Committee as an alternative to the Wet Bulb Globe Temperature (WBGT) as the standard heat stress measurement. As proposed, the instrument would integrate the environmental parameters of radiation, wet bulb and dry bulb temperatures, and air velocity with a simulation of work rate into a single output reading. This reading would be utilized to determine if a worker could work at a given rate in a given thermal environment.

Man is in equilibrium with the thermal environment when his metabolic heat production (M) is just balanced by the summation of his heat gain or loss due to radiation (R), convection (C) and evaporation of sweat (E).[1] The interface at which this heat exchange takes place is mans skin, which is maintained at approximately 95°F at equilibrium conditions. The proposed instrument would be a simulation of man. It would be cylindrical in shape with an internal heat source representing metabolic heat or work rate. The cylinder wall would represent mans skin. A cloth covering would provide a mechanism for wetting the skin surface (sweat rate). Operation would consist of setting the internal heat source equal to a predetermined "work-rate" and allowing the instrument to come to thermal equilibrium with the environment. An equilibrium instrument skin temperature below 95°F would show that the heat transfer rate to the environment was sufficient to remove the metabolic heat and any heat load from the environment thereby indicating that the selected work rate was permissible. Conversely, a skin temperature above 95°F would show a heat transfer rate less than the metabolic and environmental heat load indicating a potential heat stress problem. An alternate method of operation would be to determine the maximum allowable continuous work rate by determining the energy input required to maintain a constant 95°F instrument skin temperature. The instrument would also have to be of sufficiently small size to be of practical use, i.e., portable, low power requirements, and not interfere with the work task.

It was realized that two major problems existed in developing a workable instrument configuration. The first and most difficult problem would be the inconsistency of heat transfer coefficients. As stated above, man achieves thermal balance when the metabolic heat (M) is balanced by the summation of the heat gain or loss at his skin surface due to radiation (R), evaporation of sweat (E) and convection (C). R and C may be either positive (heat flow to man) or negative (heat flow from man). E is always from man. For an instrument to accurately simulate mans thermal response, the ratio of the radiation heat transfer coefficient and the convective heat transfer coefficient for the instrument must be equal to the same ratio for man. This will not occur when trying to simulate man with a considerably smaller diameter instrument. The radiation coefficient is only a function of the bodies view of the radiation source and as such, it is relatively unchanged for bodies of similar shape. The convective coefficient, however, is an inverse function of the bodies diameter.[2] The required small instrument diameter would thereby result in an increased convective coefficient of approximately two times that of man, while the radiation coefficient would be essentially the same. The only practical nonelectronic method to "effect" a reduction in the convective coefficient would be to reduce the air velocity across the instrument, thereby lowering the convective transfer. This would require some sort of shielding or screening. However, shielding would be self defeating in that it would also act as a radiation shield and lower the radiant heat transfer.

The second problem would be the uniform application of water (representing sweat) to the surface of the instrument. The generally accepted maximum average sweat rate for man is 1 liter per hour (this figure will vary, but is considered satisfactory for evaluation of heat stress over an 8-hour period). In a very humid environment, the sweat rate may exceed the evaporative capacity of the atmosphere with the result of excess sweat dripping off the skin. In a dry environment, the evaporative capacity of the environment may exceed the sweat rate with the result that the entire amount of sweat secreted will be evaporated. A completely saturated instrument surface would satisfactorily represent the very humid condition. However, to represent the dry condition (all conditions where the environmental evaporative capacity is sufficient to evaporate the entire amount of sweat) the water rate to the instrument surface would have to be in proportion to its surface area so as to represent 1 liter per hour. For a 0.5 ft^2 area this translates to 25 cc per hour; less than .5 cubic centimeters per minute. Such a flow rate distributed over the instrument surface would be very difficult to achieve.

A test instrument three inches in diameter by six inches long with hemispherical ends was constructed as shown in Figure I. A two-phase test was planned. The first phase would evaluate the instruments heat transfer characteristics. Experimentally determined coefficients could be compared to analytical determined coefficients, thus providing a check on instrument operation, test set up and test procedures. The first phase would also serve as a feasibility study. The second phase, if the instrument was determined to be feasible, would evaluate the instrument over the range of environmental conditions encountered in industry. Evaluation of the test instrument showed the experimental heat transfer coefficients for the radiation, evaporative, and convective modes of heat transfer to be in considerable disagreement with the values calculated from standard heat transfer relationships (Appendix A). The tests were performed both with the instrument surface dry and with it saturated with water (representing a very humid environment). Subsequent tests to manually provide water at the rate equivalent to mans one liter per hour proved unsuccessful.

For the tests performed the instrument did not respond to the environment in a predictable manner and the predicted problems of applying the results to mans physiological response were indeed present. Further testing was discontinued at this juncture. Reasons for termination were two fold:

1. The first and most compelling reason was the experimental reconfirmation of the difference in the heat tranfer coefficient ratio and the difficulties in water distribution to the instrument surface as were discussed earlier.

2. The second reason was that during the test period the DOL Heat Stress Advisory Committee adopted the Wet Bulb Globe Temperature as the Heat Stress Index, thereby negating the need for an alternate measurement method.

DISCUSSION OF RESULTS

The experimental k_c of .04364 does not compare favorably with the predicted .078094. A possible reason is that the predicted k_c is predicted on a cylinder whose walls are at a uniform temperature over the entire surface and are exposed directly to the airstream. The test instrument wall was separated from the airstream by a cotton cover which would have lowered the energy input required to maintain the wall at a given temperature. This would tend to result in a lower k_c. The instrument construction resulted in an internal convective air flow which caused an uneven temperature distribution within the instrument. The result was a considerably higher temperature in the upper portion of the instrument. It is most likely that the thermocouple placement did not obtain a true overall average instrument skin temperature.

The evaporative coefficient k_e for a man is 3.6 times the convective coefficient.[1,2] While lower than would be predicted, the experimental k_e of .19876 was felt to be in reasonable agreement with 3.6 times the experimental k_c of .04364. The reasons for the low k_c discussed previously would apply to the low k_e.

The radiation coefficient for the instrument would not be expected to differ significantly from that for man as it is basically a function of the objects "view" of the radiation source. The experimental k_r of 1.2662 agreed very well with the unclothed k_r of 1.25 for man.

The convective and radiation heat transfer coefficients of a workable instrument must have the same ratio as those of man, i.e., (k_r/k_c) for the instrument must have equal (k_r/k_c) for man. For man $(k_r/k_c) = 23.0$. The ratio for the instrument using experimental coefficients was 28.09. This high level was due to the low experimental k_c (.04364). The predicted ratio based on man being equivalent to a 15-inch cylinder[2] would be (k_r/k_c) man $\div (d_{man}/d_{instrument})^{.4}$ or $(1.25/.054) \div (15/3)^{.4} = 12.15$. The diameter relationship for convective heat transfer is a well established and documented principle. As such, the differences between the analytically and experimentally determined coefficients strongly suggest that the experimental values are in error.

It is also concluded that modification to the instrument and test procedures could result in obtaining satisfactory agreement between experimental and predicted values. However, it is not felt that it would be possible to achieve a practical method for maintaining the instruments k_r/k_c value equal to that of man. This can readily be seen in equation 9 where k_c is a function of $V^{.618}/D^{.382}$. As shown above, reducing D would increase k_c while k_r remained essentially unchanged. The only practical solution to maintain k_c instrument equal to k_c man for the reduced diameter instrument would be to reduce the effective velocity (V) across the instrument by shielding to achieve a pseudo reduction in the instrument k_c. However, a shield would also reduce the radiation to the instrument with the effective result of a reduction in k_r.

No effort was made to establish a method for distributing water to the instruments surface. The generally accepted maximum average sweat rate for man over an 8-hour period is approximately one liter per hour.[1] Assuming a 20 ft^2 surface area for man and a required instrument sweat rate proportional to area, the .589 ft^2 test instrument would require a flow rate of less than .5 cubic centimeters per minute. This small amount of water spread over nearly .6 ft^2 would be an extremely difficult task. It is not readily apparent that a practical water distribution system which would simulate mans sweat rate system could be developed for use in the industrial environment.

CONCLUSIONS AND RECOMMENDATIONS

It was concluded that the originally perceived problems attendent to a simple single reading heat stress instrument were indeed true. It was also concluded that while extended effort might produce a laboratory instrument, an instrument of this nature that could be used in the industrial environment would not be either feasible or practical.

It is recommended that future efforts in the area of heat stress instrumentation for measurement of the industrial environment be directed toward development of personal monitoring devices (environmental parameters), WBGT measuring devices with improved air velocity sensitivity and improved methods for measurement of thermal radiation.

APPENDIX A

EXPERIMENTAL EVALUATION OF HEAT TRANSFER COEFFICIENTS

The heat transfer coefficients for the single reading heat stress instrument were determined experimentally using the equations which apply to man.

$$C = k_c A\, V^n\, (t_a - t_s)$$

$$R = k_r A \quad (t_w - t_s)$$

$$t_w = \left[(t_g + 460)^4 + .103 \times 10^9\, V^{.5}(t_g - t_a)\right]^{.25} - 460$$

$$E = k_e A\, V^n\, (P_s - P_a)$$

$$E = M + R + C \quad \text{(for man in thermal equilibrium)}$$

where

A	=	area of heat transfer surface	ft^2
C	=	convective heat transfer	$\frac{Btu}{hr}$
E	=	evaporative heat transfer	$\frac{Btu}{hr}$
k_c	=	coefficient of convective heat transfer	$\frac{Btu}{hr\ ft^2\ F(ft/m)^n}$
k_e	=	coefficient of evaporative heat transfer	$\frac{Btu}{hr\ ft^2\ (ft/m)^n}$
k_r	=	coefficient of radiation heat transfer	$\frac{Btu}{hr\ ft^2\ {}^oF}$

M	=	metabolic heat production	$\frac{Btu}{hr}$
n	=	dimensionless exponent	
P_a	=	vapor pressure of water in the air	mm Hg
P_s	=	vapor pressure of water at skin temperature	mm Hg
R	=	radiation heat transfer	$\frac{Btu}{hr}$
t_a	=	air temperature	^oF
t_g	=	black globe temperature	^oF
t_s	=	skin temperature or instrument surface temperature	^oF
t_w	=	mean radiant temperature	^oF
V	=	air velocity	$\frac{ft}{min}$

CONVECTION

To determine k_c and n test points were selected such that there would be no heat transfer by either evaporation or radiation. This was accomplished by maintaining the instrument skin dry ($P_s = P_a$ and $E = 0$) and by setting the instrument skin temperature equal to the mean radiant temperature of the test enclosure ($t_w = t_s$ and $R = 0$). With $E + R + 0$ the heat input to the instrument (M) can only be dissipated by convection (C). Several test points were run at varying air velocities, heat inputs and instrument skin temperatures. (Refer to Table 1.)

With $E = R = 0$ the heat balance equation reduces to:

$$M = -C = -k_c A\, V^n\, (t_a - t_s) \tag{1}$$

For a given shape k_c and n will essentially be the same for all conditions of temperature and velocity within the range of concern with heat stress.[2] Therefore:

$$k_c A\, V^n = -\frac{M}{t_a - t_s} = K \frac{\text{Btu}}{\text{hr } ^\circ\text{F}} \tag{2}$$

for test points 1, 2, 3, and 4

$$(k_c A\, V^n)_1 = K_1 = -\frac{14.077}{-19.5} = .7218 \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

$$(k_c A\, V^n)_2 = K_2 = -\frac{32.418}{-22} = 1.4735 \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

$$(k_c A\, V^n)_3 = K_3 = -\frac{14.62}{-17.9} = .816 \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

$$(k_c A\, V^n)_4 = K_4 = -\frac{40.168}{-25} = 1.60 \frac{\text{Btu}}{\text{hr } ^\circ\text{F}}$$

also $$\frac{(k_c A)_a \;\; V_a}{(k_c A)_b \;\; V_b} = \frac{K_a}{K_b}$$

$$(k_c A)_a = (k_c A)_b$$

$$\frac{V_a^{\,n}}{V_b^{\,n}} = \frac{K_a}{K_b}$$

$$\left(\frac{V_a}{V_b}\right)^n = \frac{K_a}{K_b}$$

and

$$\left(\frac{V_2}{V_1}\right)^n = \frac{K_2}{K_1}$$

$$\left(\frac{V_4}{V_3}\right)^n = \frac{K_4}{K_3}$$

For test points 1 and 2

$$\left(\frac{210}{85}\right)^n = \frac{1.4735}{.7218}$$

$$2.47^n = 2.04$$

$$n = .79$$

For test points 3 and 4

$$\left(\frac{210}{85}\right)^n = \frac{1.60}{.816}$$

$$2.47^n = 1.96$$

$$n = .745$$

$$n_{avg} = \frac{.79 + .745}{2} = .765$$

from equation 2

$$k_c = \frac{K}{A\ V^n} \qquad \frac{Btu}{hr\ ft^2\ F\ (ft/m)^n}$$

For test points 1, 2, 3, and 4

$$k_{c_1} = \frac{.7218}{(.589)(85)^{.765}} = .040958 \frac{Btu}{hr\ ft^2 F(ft/m)^{.765}}$$

$$k_{c_2} = \frac{1.4735}{(.589)(210)^{.765}} = .041854 \frac{Btu}{hr\ ft^2 F(ft/m)^{.765}}$$

$$k_{c_3} = \frac{.816}{(.589)(85)^{.765}} = .046303 \frac{Btu}{hr\ ft^2 F(ft/m)^{.765}}$$

$$k_{c_4} = \frac{1.60}{(.589)(210)^{.765}} = .046486 \frac{Btu}{hr\ ft^2 F(ft/m)^{.765}}$$

$$k_{c_{avg}} = .04364 \frac{Btu}{hr\ ft^2 F(ft/m)^{.765}}$$

RADIATION

The radiation coefficient (k_r) was determined by maintaining the skin dry ($P_s = P_a$ and $E = 0$) as above, but setting a difference in t_w and t_s to obtain a radiant heat flow. With only $E = 0$ the heat balance equation reduces to:

$$M = -C - R = -k_c A\, V^n\, (t_a - t_s) - k_r\, A\, (t_w - t_s) \qquad (3)$$

and

$$k_r = -\frac{M + k_c A\, V^n\, (t_a - t_s)}{A\, (t_w - t_s)} \qquad \frac{Btu}{hr\ ft^2\ {}^oF} \qquad (4)$$

for test points 5, 6, and 7

$$k_{r5} = -\frac{73.107 + \left[(.04364)(.589)(210)^{.765}\ (87 - 126.7)\right]}{(.587)\ (111.7 - 126.7)} = -\ 1.3712\ \frac{Btu}{hr\ ft^2\ {}^oF}$$

note: negative k_{r5} indicates radiant heat flow from the instrument to the test enclosure.

$$k_{r6} = -\frac{0 + \left[(.04364)\ (.589)(90)^{.765}\ (87 - 95.4)\right]}{(.589)(104.5 - 95.4)} = 1.2115\ \frac{Btu}{hr\ ft^2\ {}^oF}$$

$$k_{r7} = -\frac{1.6 + \left[(.04364)\ (.589)(90)^{.765}\ (87 - 96.3)\right]}{(.589)(104.5 - 96.3)} = 1.2159\ \frac{Btu}{hr\ ft^2\ {}^oF}$$

$$k_{r_{average}} = 1.2662 \qquad \frac{Btu}{hr\ ft^2\ {}^oF}$$

EVAPORATION

The evaporative coefficient k_e was determined by maintaining a difference in t_w and t_s, as in the determination of k_r, and saturating the instrument cover with water to obtain evaporative cooling. The heat balance equation for this situation was:

$$E = M + R + C = M + k_r A\,(t_w - t_s) + k_c A V^{.765}\,(t_a - t_s)\ \frac{Btu}{hr} \qquad (5)$$

for test points 8, 9, and 10

$$\begin{aligned} E_8 = 20.06 \quad & + \left[(1.2662)\ (.589)\ (104.5 - 68.7)\right] \\ & + \left[(.04364)\ (.587)\ (70)^{.765}\ (88 - 68.7)\right] \\ & = \quad 62.259\ \frac{Btu}{hr} \end{aligned}$$

$$\begin{aligned} E_9 = 38.486 \quad & + \left[(1.2662)\ (104.5 - 72.7)\right] \\ & + \left[(.04364)\ (.589)\ (70)^{.765}\ (88 - 72.7)\right] \\ & = \quad 74.492\ \frac{Btu}{hr} \end{aligned}$$

$$\begin{aligned} E_{10} = \quad 0 \quad & + \left[(1.2662)\ (.589)\ (104.5 - 58.1)\right] \\ & + \left[(.04364)\ (.589)\ (70)^{.765}\ (88 - 58.1)\right] \\ & = \quad 58.62\ \frac{Btu}{hr} \end{aligned}$$

However:

$$E = k_e A\ V^{.765}\ (P_s - P_a)\ \frac{Btu}{hr} \qquad (6)$$

and

$$k_e = \frac{E}{A\ V^{.765}\ (P_s - P_a)} \qquad \frac{Btu}{hr\ ft^2\ mm\ hg\ (ft/m)^{.785}} \tag{7}$$

for test points 8, 9, and 10

$$k_{e8} = \frac{62.259}{(.589)(90)^{.765}\ (18 - 3)} = .206026 \ \frac{Btu}{hr\ ft^2\ mm\ hg\ (ft/m)^{.785}}$$

$$k_{e9} = \frac{74.492}{(.589)(90)^{.765}(20 - 3)} = .217506 \ \frac{Btu}{hr\ ft^2\ mm\ hg\ (ft/m)^{.785}}$$

$$k_{e10} = \frac{58.62}{(.589)(215)^{.765}\ (12.5 - 4)} = .172843 \ \frac{Btu}{hr\ ft^2\ mm\ hg\ (ft/m)^{.785}}$$

$$k = .19876 \ \frac{Btu}{hr\ ft^2\ mm\ hg\ (ft/m)^{.765}}$$

PREDICTED CONVECTIVE COEFFICIENT

The predicted value for the coefficient of convection was derived from the Nusselt-Reynolds number correlation for airflow normal to a cylinder.[3]

$$\frac{hD}{k_f} = B\left(\frac{D\ U_\infty}{\gamma\ f}\right)^n \tag{8}$$

where

h = convective coefficient $\dfrac{Btu}{hr\ ft^2\ {}^\circ F\ (ft/hr)^n}$

D = diameter ft

k_f = thermal conductivity of air $\frac{Btu}{ft\ hr\ ^oF}$

B = dimensionless correlation coefficient

U_∞ = air velocity $\frac{ft}{hr}$

ν_f = kinematic viscosity $\frac{ft^2}{hr}$

n = dimensionless exponent

For an air temperature of 100°F and a velocity range of 80 to 250 $\frac{ft}{m}$:

$$B = .174$$

3

$$n = .618$$

and for D = .25 ft

$$h = \frac{B\ D^{.618}\ U_\infty^{.618}\ k_f}{D\ \nu^{.618}} = \frac{B\ U_\infty^{.618}\ k}{D^{.382}\ \nu^{.618}} \qquad (9)$$

$$h = \frac{(.174)\ (U)^{.618}\ (.016)}{(.25)^{.382}\ (.648)^{.618}}$$

$$h = .006219\ U^{.618} \qquad \frac{Btu}{hr\ ft^2\ ^oF\ (ft/hr)^{.618}}$$

$$h = .006219\ (60\ U)^{.618} \qquad \frac{Btu}{hr\ ft^2\ o^oF\ (ft/m)^{.618}}$$

finally:

$$h_{predicted} = .078094 \qquad \frac{Btu}{hr\ ft^2\ ^oF\ (ft/m)^{.618}}$$

TABLE 1

TEST DATA

	T_g Globe	T_w* Mrt	T_a Dry Bulb	T_{wb} Wet Bulb	V_a Air Velocity	P_a Vapor Press	T_s Inst Skin	E in Voltage	I Current	P_w Watts	M ** Btu
	°F	°F	°F	°F	ft/min	mm hg	°F	Volts DC	Amps	Watts	Btu
1	83.3	95.4	75.0	-	85	-	94.5	16.5	.250	4.125	14.077
2	80.9	96.5	74.0	-	210	-	96.5	23.0	.413	9.499	32.48
3	97.9	107.9	90.0	-	85	-	107.9	17.0	.252	4.284	14.62
4	95	110.0	86.0	-	210	-	110.0	26.75	.440	11.77	40.168
5	95.0	111.7	87.0	-	210	-	126.7	35.06	.611	21.421	73.107
6	94.3	104.5	87	-	90	-	95.4	0	0	0	0
7	94.3	104.5	88	-	90	-	96.3	5.056	.09	.455	1.6
8	94.3	104.5	88	57	90	3.0	68.7	18.093	.325	5.880	20.06
9	94.3	104.5	88	57	90	3.0	72.7	25.06	.450	11.217	38.486
10	81.0	81.0	81.0	55	215	4.0	58.1	0	0	0	0

* Calculated from t_w = $(t_g + 460)^4 + .103 \times 10^9 V^{.5} (t_g - t_a)^{.25} - 460$

** Calculated from $E_x I$ = P_w $P_w \times 3.41218 = M$

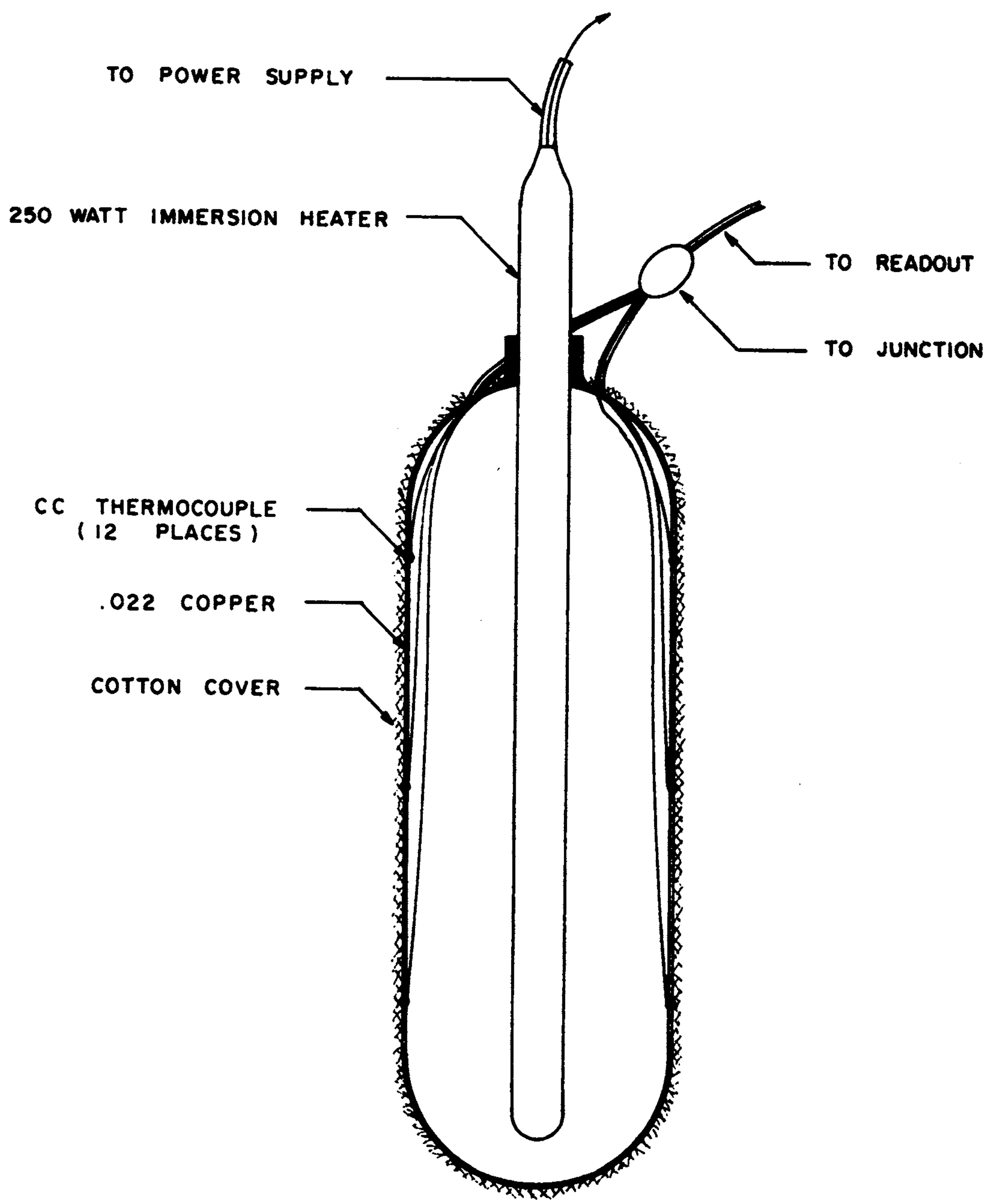

Figure 1. Single Reading Heat Stress Instrument.

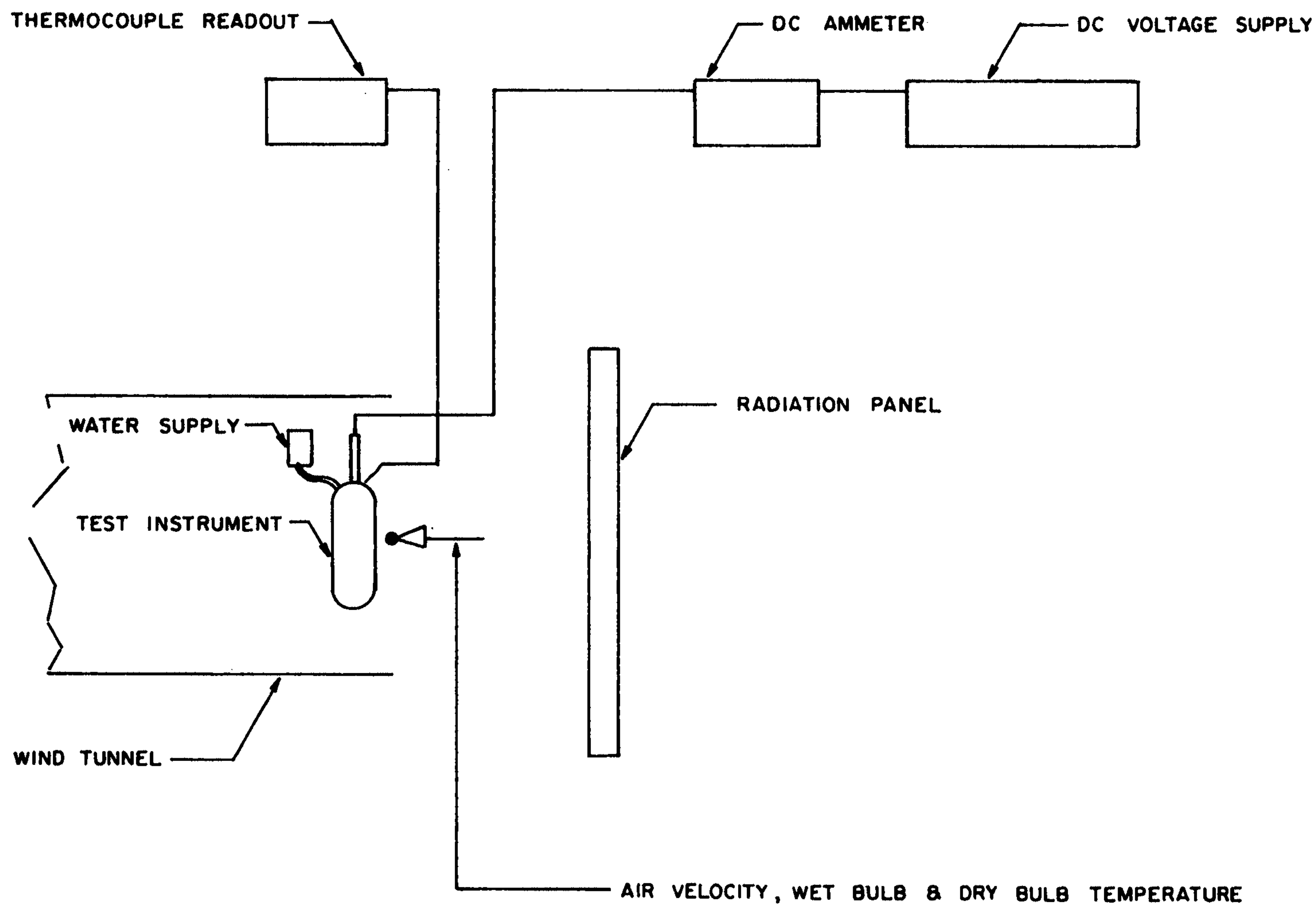

Figure 2. Test Setup.

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☆ U. S. GOVERNMENT PRINTING OFFICE: 1975-657-601/5538 Region No. 5-11

HEW Publication No.(NIOSH) 75–148

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