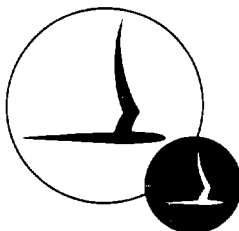


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**CORNELL AERONAUTICAL LABORATORY, INC.
BUFFALO, NEW YORK 14221**

DEVELOPMENT OF THE GLOBE THERMOMETER

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FOREWORD

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I. INTRODUCTION AND SUMMARY

The globe thermometer, as originally conceived, was intended to provide a temperature measurement that could be related to human comfort in an environment of convective air cooling with sources of radiative heating. It was claimed by the originator (Vernon, Reference 1) that the temperature within a liquid-filled blackened globe (thin walled) by itself was a measure of bodily comfort for a person exposed to both air currents and radiative energy. However, this application of the globe thermometer is not without its limitations, as pointed out by Bedford and Warner, Reference 2. According to Vernon's own tests, subjects that were comfortable at 62°F globe thermometer readings indoors were not comfortable outdoors where wind velocity exceeded 93 ft/min unless the globe thermometer readings exceeded 69.5°F. Vernon tried to show that the difference between air temperature and globe thermometer temperature was significant as an indicator of the "effectual radiation" imposed upon the human body. This is not true because the globe thermometer reading will not yield radiative heat intensity without benefit of the simultaneous determination of air temperature and velocity. This can be demonstrated theoretically as done by Bedford and Warner, Reference 2.

If both the air temperature and velocity are measured simultaneously with globe temperature, the globe thermometer is useful for evaluating an environment for human comfort, as demonstrated by Bedford and Warner. They documented evidence in Reference 2 that these measurements could be used to predict consistently, the equivalent temperatures that would be indicated by an eupatheoscope.* Equivalent temperature of an environment is the temperature of a uniform enclosure containing still air, within which the eupatheoscope would lose heat at the same rate as in the environment.

* The eupatheoscope is a blackened hollow copper cylinder that is heated electrically to maintain it at 75°F, and the heat loss from it is equal to the power input. The heat loss per unit area from the eupatheoscope is approximately that of a comfortable person in an environment where sweat cooling is not important.

Recently, it has been shown that the wet-bulb globe temperature index is one of the most effective heat stress indices yet devised, Reference 3. Furthermore, this index, known as WBGT, has been adopted by the American Conference of Governmental Industrial Hygienists as a standard and it has been published in a "Notice of Intent" for 1971 that all conditions of heat stress to which working men are exposed shall be monitored for WBGT so that permissible durations of exposure and intervening rest periods out of the job environment can be assigned each job. Of course, the globe thermometer reading is necessary to the computation of WBGT.

Because of this new prominence of the globe thermometer and because recent experiences with its use have been in more severe environments than it was ever tested for, the National Institute for Occupational Safety and Health of the Department of Health, Education and Welfare, sponsored a program at Cornell Aeronautical Laboratory for evaluating the globe thermometer in a variety of forms, by exposure to extreme environments of simultaneous cooling and heating. It was required that research be conducted to develop information on the optimum construction and use of the globe thermometer, and on the calculation of radiant heat exchange from globe temperatures.

Specific aims of this research were to include:

- (a) the determination of the effects of shell diameter and thickness, and sensor type and emissivities upon time lag.
- (b) the investigation of means for reducing time lag.
- (c) determination of the convection coefficients and their stability at air velocities between 15 and 1000 feet per minute.
- (d) the development of equations for calculating the mean radiant temperature with an error of not more than 3

percent for spaces having average surface temperatures from 30°F below to 275°F above air temperature and having air velocities from 15 to 1000 feet per minute. Air temperatures were to vary between 70°F and 140°F.

In order to conduct the required experiments for this research a heated-wall wind tunnel was assembled from sheet metal using an existing 32-inch axial-flow fan. A test section having a square cross-section, 3 feet by 3 feet long, was fabricated from 1/4 inch thick aluminum plate. The wind tunnel could be assembled in two configurations, using the same fan and test section, which was equipped to support globe thermometers. One configuration was a single-pass tunnel with the fan at the exit end, drawing air through the tunnel after it entered a bell-mouth inlet. This insured uniform flow for measuring heat transfer coefficients on spheres. The other configuration was a closed circuit tunnel from which some air could be dumped, and cool air inducted to control air temperature. This was used to expose globe thermometers to various conditions of tunnel wall temperature, air temperature and air velocity, for their evaluation.

The heat transfer coefficients that were measured on 4 and 6 inch diameter spheres agreed well with results of other researchers so that an equation could be used to describe them adequately.

Response of a variety of globe thermometers was measured, using both tunnels. It was found that when either a miniature bare wire thermocouple or a tiny commercial thermistor was used as a sensor, time lag was completely determined by the shell, whereas a glass thermometer caused increased time lag with most shells. Globe thermometers exhibited less time lag with thinner shells, and an aluminum shell permitted faster response than a copper shell of the same thickness. The fastest response was measured for a 4-inch diameter aluminum shell 0.015 inch thick using a miniature thermistor (0.15 inch max-dimension). This configuration reached equilibrium temperature in 8.5 minutes in air at 720 fpm velocity. By contrast, the same shell with a mercury-in-glass thermometer required 21 minutes to reach equilibrium.

The evaluation of globe thermometers revealed that globe thermometer temperatures can be measured accurately with any combination of relatively thin, 4 or 6 inch diameter, metallic globe, and good quality sensor (accurate to 1% of full scale). This was found to be the case for any of the specified conditions of air temperature, radiating source temperature, air velocity, and globe geometry. For example, a globe thermometer fabricated from a copper toilet float was as good as one using a 4-inch diameter spherical shell. In addition, badly dented shells (dents 1/2 inch deep in a 4-inch diameter shell) had little or no effect upon equilibrium temperature.

The evaluation included the computation of mean radiant temperature for each test (the temperature required of a complete, uniformly radiating enclosure to produce the same effect upon the globe). These were compared to the brightness^{*} temperatures of the heated walls of the tunnel, which acted as a 98.0% complete enclosure, and agreement was generally good. It was concluded that the formula derived for mean radiant temperature was a satisfactory one. It differs from the traditional one only because the updated correlation for Nusselt number was used.

All aims of this research have been satisfied, and considerable new data have been generated to aid in the future design and use of the globe thermometer over a broad range of temperature and wind speed conditions.

Finally, it should be emphasized that only reasonable care was taken in fabricating globe thermometers for test. Yet, the temperatures indicated by them were highly reproducible in terms of either re-exposing them or reorienting them at the same location and test conditions.

* Brightness temperature is the temperature required of a source to radiate with the same intensity if its emissivity was unity, i. e., if it radiated with 100% effectiveness.

II. THEORY OF THE GLOBE THERMOMETER

Consider a globe thermometer containing a fluid, which is placed within an enclosure of n surfaces, some of which may be hotter than the enclosed air. The air moves through the enclosure at uniform, constant temperature and velocity. A heat balance may be written for an element of the globe shell of thickness, δ . Neglecting heat conduction in any direction parallel to the globe surface, the expression following is obtained:

$$(q_r - q_c)_{\text{outer}} - (q_r - q_c)_{\text{inner}} = -\rho c \delta \frac{dT}{dt} \quad (1)$$

assuming constant convective and radiative heat transfer rates, q_c and q_r , respectively, that the inner and outer surface areas of the shell are equal, and that there are no conduction losses to any supports or sensor leads. The neglect of heat conduction along the surface direction is only justified for metallic shells that are thin relative to their overall dimension, so that radial temperature gradient is insignificant. That this is a realistic approach was demonstrated during measurements of heat transfer coefficients for this project. At a radiative enclosure temperature of 250°F when exposed to the highest wind velocity of 1000 fpm, (which produces a large variation of convective heat transfer rate along the surface), the temperature gradient along the surface was about one degree Fahrenheit per inch. Even if the radiative source had been at 420°F (the limiting condition of interest) and localized to reinforce the effect upon shell temperature caused by variation of convective heating around it, the temperature gradient would have been less than 3 degrees per inch. In the air filled globe there are neither heat sinks nor sources within the shell so that the radial temperature gradient must be less than this. Consequently, differences between internal and external shell temperatures would be difficult to detect.

For the usual applications the steady heat balance is the one of interest, i. e., the globe thermometer is used by waiting for its temperature sensor to indicate a steady value. For this case the right hand side of Equation 1 becomes zero.

When the integral of the expression thereby obtained is taken over the whole globe the fact can be used that the integrated heat loss from the inner surface is zero (elements of the inner surface radiate to each other and internal fluid currents redistribute heat from one point to another). An integral heat balance for the outer surface is left:

$$\int q_c dA = \int q_r dA$$

or

$$\int h(\theta) [T_s(\theta) - T_a] dA = \sum_{i=1}^n R_i \bar{F}_{si} A_i - \epsilon_i \sigma [T_s(\theta)]^4 dA \quad (2)$$

The first term on the right side of the expression represents the thermal radiation incident upon the globe. The quantity R_i is the radiative flux from any surface, i , of the enclosure. In the case of diffuse surfaces it is called the radiosity, which is equivalent to

$$R_i = \epsilon_i \sigma T_i^4 + \rho_i \sum_{j=1}^n R_j \bar{F}_{ij}$$

for gray surfaces. In this definition the globe surface is necessarily included as one of the surfaces of the enclosure. However, if the globe is small compared to the other surfaces or if it is distant from an intense concentrated source its influence upon the radiation from the source is negligible. Consequently, it is rational to consider an equivalent enclosure that has a uniform temperature, which is capable of duplicating the combined effect of the surfaces of the enclosure. Therefore,

$$\sum_{i=1}^n R_i \bar{F}_{si} A_i = \alpha_w \sigma T_w^4 A_s$$

where T_w is the temperature of the fictitious equivalent enclosure. A brightness temperature, T'_w , can also be postulated, which is more consistent with the original concept of mean radiant temperature:

$$\sum_{i=1}^n R_i \bar{F}_{si} A_i = \sigma (T'_w)^4 A_s$$

The second term on the right side of expression (2) may not be significant to the source radiation, but it is an important factor in the heat balance of the globe thermometer. Its integration requires knowledge of the distribution of T_s with θ . However, an important function of the globe is to act as an integrator, which it accomplishes by virtue of the internal radiation that transfers heat among elements of its inside surface as well as to the sensor within it. As a result, the second term becomes simply $\sigma \alpha_s T_s^4$ where T_s is an average shell temperature indicated by a sensor centrally located within the shell.

The term on the left side of the equation is the only one remaining that involves a function of surface distribution. However, a major part of the work of this program was devoted to the measurement of $h(\theta)$; the distribution of the heat transfer coefficient of convection. Furthermore, it was found that, over the range of conditions of interest to the application of the globe thermometer, the integrated value of $h(\theta)$, $\bar{h}$, could be correlated in a conventional manner with Reynolds number, through the medium of the Nusselt number, $Nu = \bar{h}d/k$. In other words, a simple relation was found that satisfactorily represented $\bar{h}$, which was computed from the experimental data for $h(\theta)$ as $\bar{h} = \int h(\theta) dA$.

The net effect of the foregoing considerations is that the heat balance equation is simplified to the following:

$$\bar{h}(T_s - T_a) = \alpha_w \sigma T_w^4 - \alpha_s \sigma T_s^4 \quad (3)$$

III. EXPERIMENTAL APPARATUS

Test Apparatus

Two low-speed wind tunnels were built using a common test section. One was designed to generate a uniform and symmetrical flow in the test section. The other generated an air current of known velocity and temperature, to which globes were exposed within a radiating enclosure, where almost all surfaces viewed by the globe were at the same temperature. Overall views of the two tunnels are shown by the photographs of Figure 1 and 2.

The uniform flow or single-pass tunnel, was one in which flow was induced in a straight, open-ended duct by a fan at the exhaust end (Figure 1). The square cross-section was 35-3/4 in. on each side. A bell-mouth was fitted at the inlet end for smooth entry of air. Its opening was 5 ft. x 5 ft., having a curve of one-foot radius. Tunnel walls were 20 ga. galvanized steel except for the test section, which was made from 1/4 in. aluminum alloy plate to insure rigidity and flatness of the walls for parallel airflow and to provide moderate thermal capacity for assistance in maintaining uniform wall temperature distribution for those surfaces nearest the globe model. Other pertinent dimensions and details are given by Figure 3.

The test section was wrapped with 220-volt "Chromalox" blanket heaters. These were woven from fiberglass yarn and resistance wires insulated by Fiberglas to permit operation at temperatures up to 800°F. Wire spacing was very close, i.e., 1/8 in. to produce uniform heat distribution. They were designed to operate at an energy density of 2.4 watts per square inch at a temperature of 400°F. Each heater was 18 x 36 in. A total of 24 heaters were required to cover the test section when placed edge to edge, with the long dimension just sufficient to reach across the tunnel wall. In addition to the test section, a three-foot length of the tunnel duct directly downstream from the test section was covered with eight heaters in the same manner.

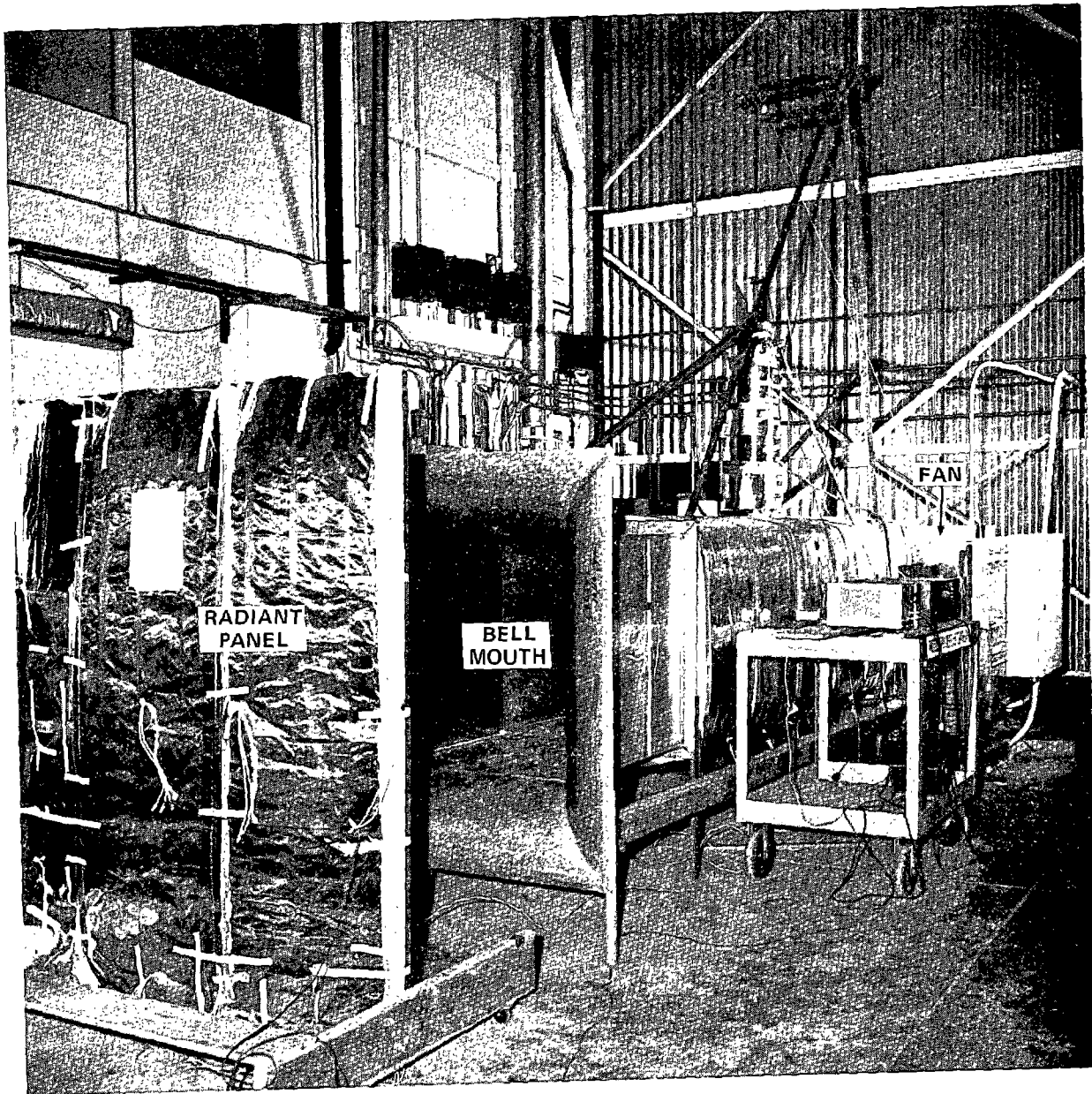


Figure 1

PHOTOGRAPH OF SINGLE-PASS WIND TUNNEL

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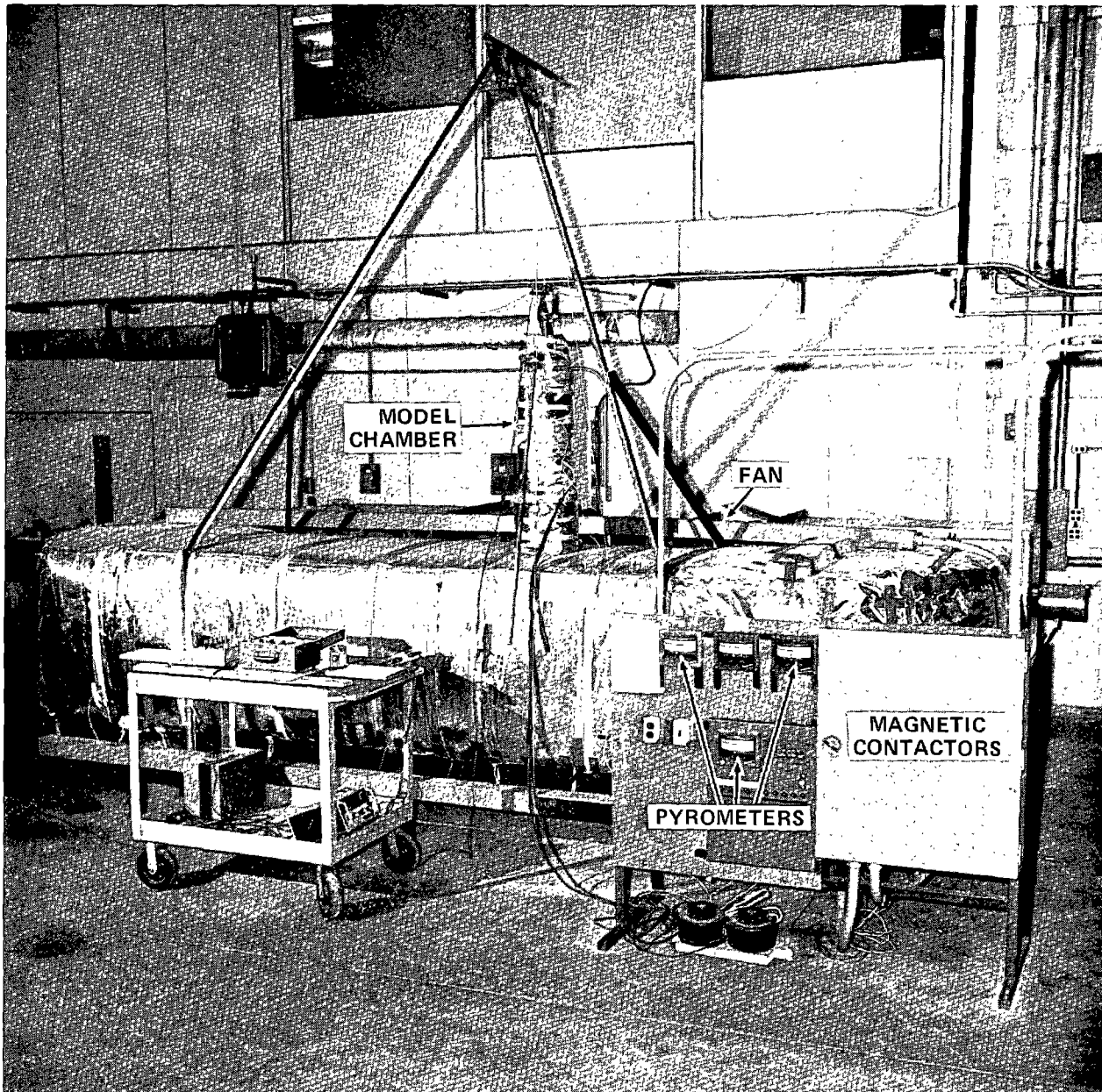


Figure 2

PHOTOGRAPH OF CLOSED CIRCUIT WIND TUNNEL

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Two electrical circuits were used for the single-pass tunnel with all the top heaters on one circuit and the rest on the other. This permitted separate temperature control of the tunnel roof, which required less heat input to maintain a given wall temperature than did either the side or floor heaters. West Instrument Company controlling pyrometers were used to obtain automatic control of wall temperatures which were sensed by iron-constantan thermocouples welded to the sheet metal. The pyrometers could be preset to any temperature above room temperature, up to 850°F. They can be seen in Figure 2 on the control panel, where three are mounted in line, near the top on the left side. Power to each circuit was brought in through a magnetic contactor, which was controlled by one of the West pyrometers (for on-off control). A fourth West pyrometer, mounted below the others, was used to indicate temperatures from thermocouples at several other points in the tunnel wall through a monitor switching device, which is also on the panel. The thermocouple locations in the tunnel walls are identified by Figure 4, which is a view of the "unfolded" tunnel walls.

The closed-circuit tunnel used the same test chamber with corner ducting at each end and connecting ductwork to the driving fan, which was located in the opposite leg. The plan is given by Figure 5, which gives details and dimensions. The upstream corner was equipped with a double set of turning vanes with five in each set, and each turned the flow by 45°. A single set of three 90° turning vanes was provided inside the next corner upstream, i. e., the one directly downstream from the fan.

Each corner was partially covered with heaters with two on the roof, two under the floor, and several on the side (outside) of the corner. The corner upstream from the test section had four side heaters and the downstream one had five side heaters. The radiative enclosure formed by the heated walls thereby was almost complete in the downstream direction but was reduced somewhat in the upstream direction because of the presence of the unheated turning vanes, which obstructed the view. The location of the test station, which was much nearer the downstream corner, helped to minimize this obstruction by the turning vanes. The net proportion of heated surface

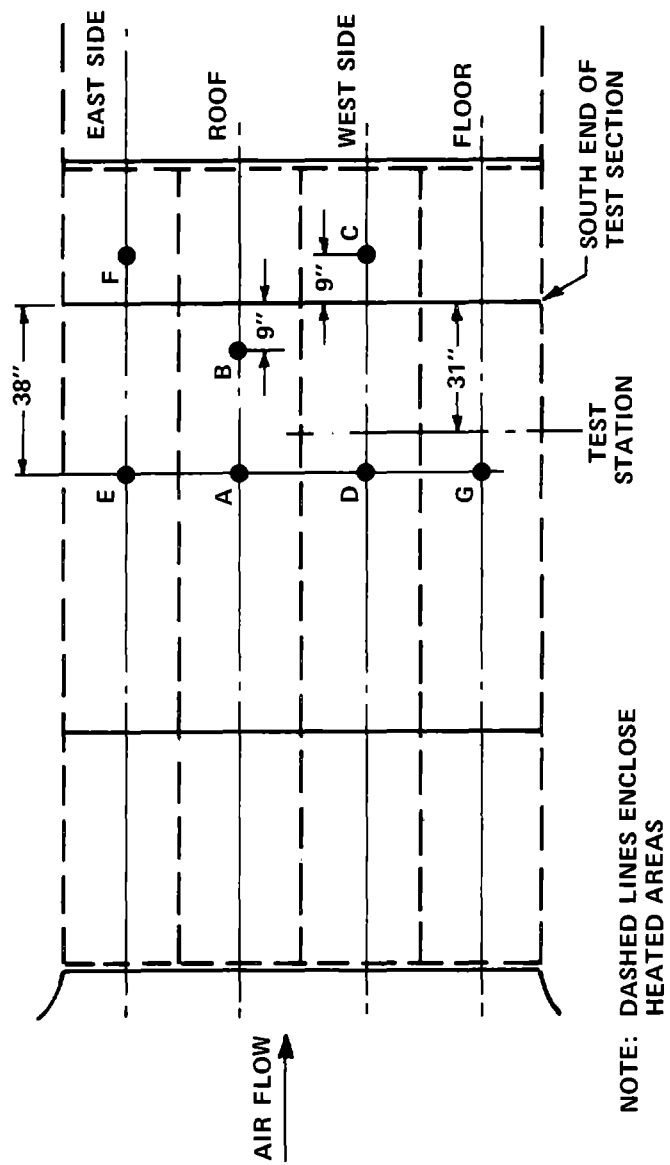


Figure 4

THERMOCOUPLE LOCATIONS IN THE WALLS OF THE SINGLE-PASS TUNNEL

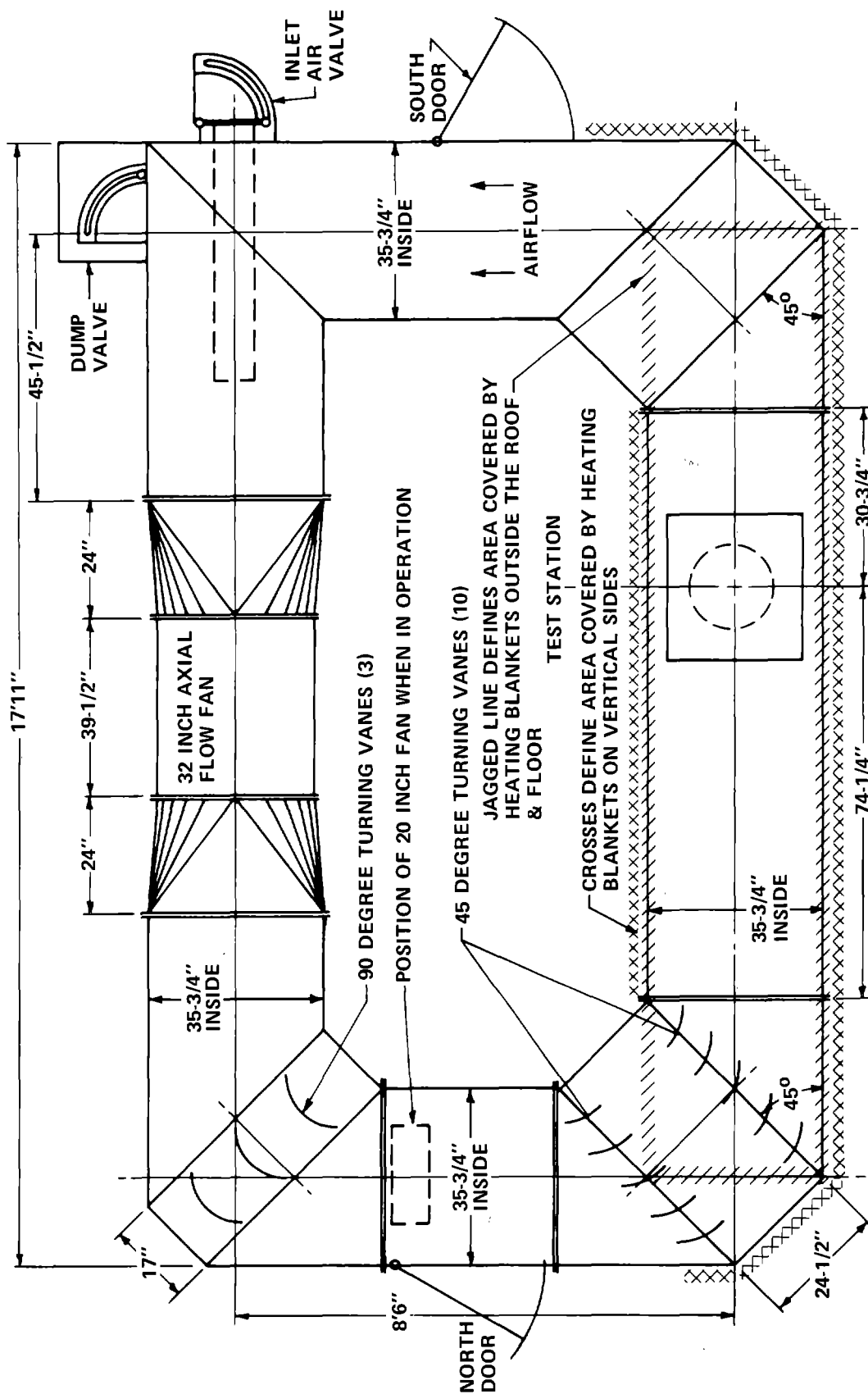


Figure 5
PLAN OF CLOSED-CIRCUIT WIND TUNNEL

effective as a radiative enclosure of a globe thermometer in the test location was computed to be 0.98. This value was used later as the angle factor $F_{S, HW}$ for this tunnel. All heated walls were covered outside by 4 inches of Fiberglas wool insulation. The thermocouples for measuring tunnel wall temperatures were located as shown by Figure 6.

Two fans were used separately for driving the tunnels. One was a Buffalo Forge 32 in. diameter axial flow fan (four blade, single stage) which was driven by either a 5 HP, 1725 rpm motor at about 1500 rpm or a 1HP, 1100 rpm motor which ran the fan at about 500 fpm. The other fan was a 20-in. diameter, three-speed fan which was placed downstream from the large fan when in operation. Its purpose was to generate tunnel air speeds below 250 fpm. Very low tunnel air speeds were generated in the single-pass tunnel by placing the smaller fan at a distance from the tunnel exit. In the closed circuit tunnel the fan outlet was filtered to produce lower wind speeds.

At tunnel wind velocities below 70 fpm, the secondary air currents induced by the heated walls (natural convection) were of sufficient magnitude to disturb the flow in the core of the stream of this horizontal tunnel. Therefore, a convergent nozzle, constructed from plywood, was used to direct the stream past the test globe at the lower speeds. It was of square cross-section three feet on each side at the upstream end, one foot on each side at the downstream end, and four feet long. The angle of its convergence and its location in the test section were such that it could not shade the globe from any radiation from the heated tunnel walls.

A model storage chamber was provided above the test section directly over the test station so that the test globe could be withdrawn from the stream and preheated to an elevated temperature (see Figure 3). The test chamber was fabricated from 7-in. I. D. pipe 42 in. long with a hinged break 4 in. above the lower end so that it could be tipped down to a horizontal position for making model changes. It was heated by wrapping it with circular band heaters 2 in. wide placed at 3-in. intervals along the whole length. The heaters were wired

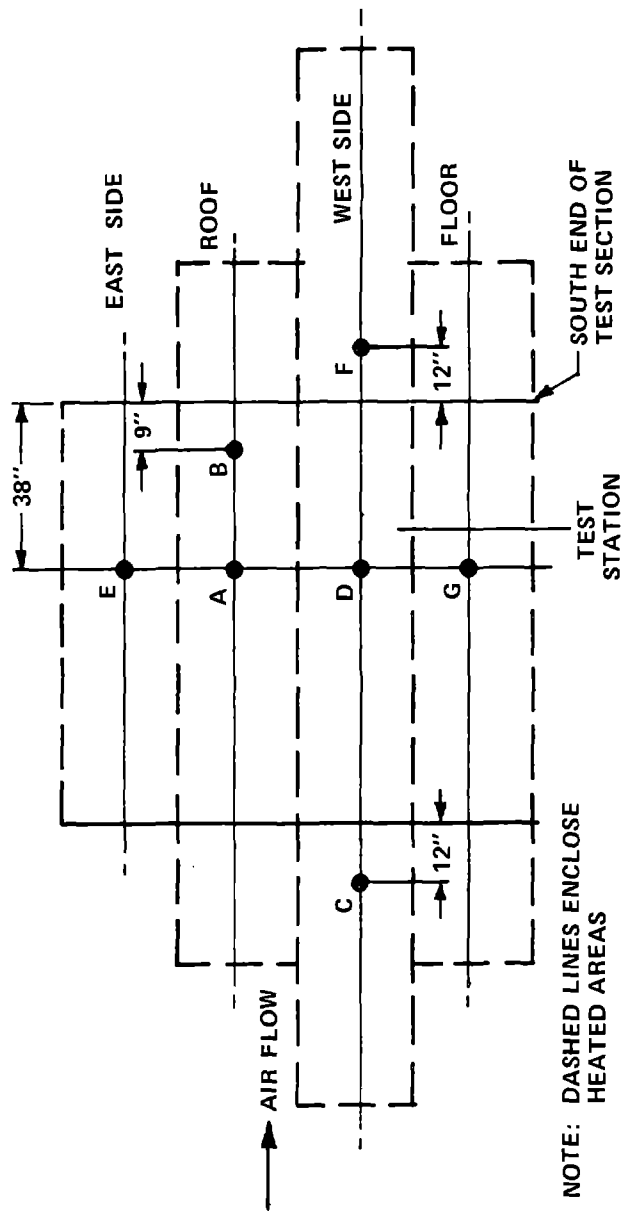


Figure 6
THERMOCOUPLE LOCATIONS IN THE WALLS OF THE CLOSED CIRCUIT TUNNEL

in parallel in two 15-ampere, 110-volt circuits. Power to each was manually controlled through a Variac transformer. The model chamber was provided a single iron-constantan thermocouple located at mid-height.

A light derrick frame was constructed over the test section for manually raising and lowering models between the tunnel and the storage chamber by means of a cord attached to the upper end of the tubular model support. This frame can be seen in the photographs of Figures 1 and 2. A latch was also provided so that the model could be held in the raised position. A view inside the tunnel is given by Figure 7.

A radiant panel was used with the straight flow tunnel to complete the radiative enclosure at the upstream end. It consisted of a 16-gage steel sheet 5 ft. wide by 6 ft. high mounted vertically on a wooden base with casters. One surface of the sheet was covered by a drapery of the blanket heaters, which were laced together. With a double layer of heaters covering the lower half of the panel and with all heaters plugged into a 110-volt line, a fairly uniform wall temperature of 250°F could be maintained at a room temperature of 60°F. The drapery of heaters was backed up by 4 in. of Fiberglas insulation. This panel was designed for effective use at 42 in. distance from the tunnel inlet, and it is depicted in place in Figure 3, and in the photograph of Figure 2.

Instrumentation

The instrumentation used during globe thermometer experiments included anemometers for measuring mean wind velocity, liquid-in-glass thermometers for determining stream air temperatures as well as globe temperature, thermistors, and thermocouples for use as sensors in the globe thermometers.

Two anemometers were used to insure availability of one at all times. One, a Flowtronic Model 55, was manufactured by Flow Corporation of Cambridge, Massachusetts (now C. G. S. Scientific Corporation, Datametrics Division, 127 Coolidge Hill Road, Watertown, Massachusetts). It uses a probe with a heated wire filament and has a transistor circuit to maintain a

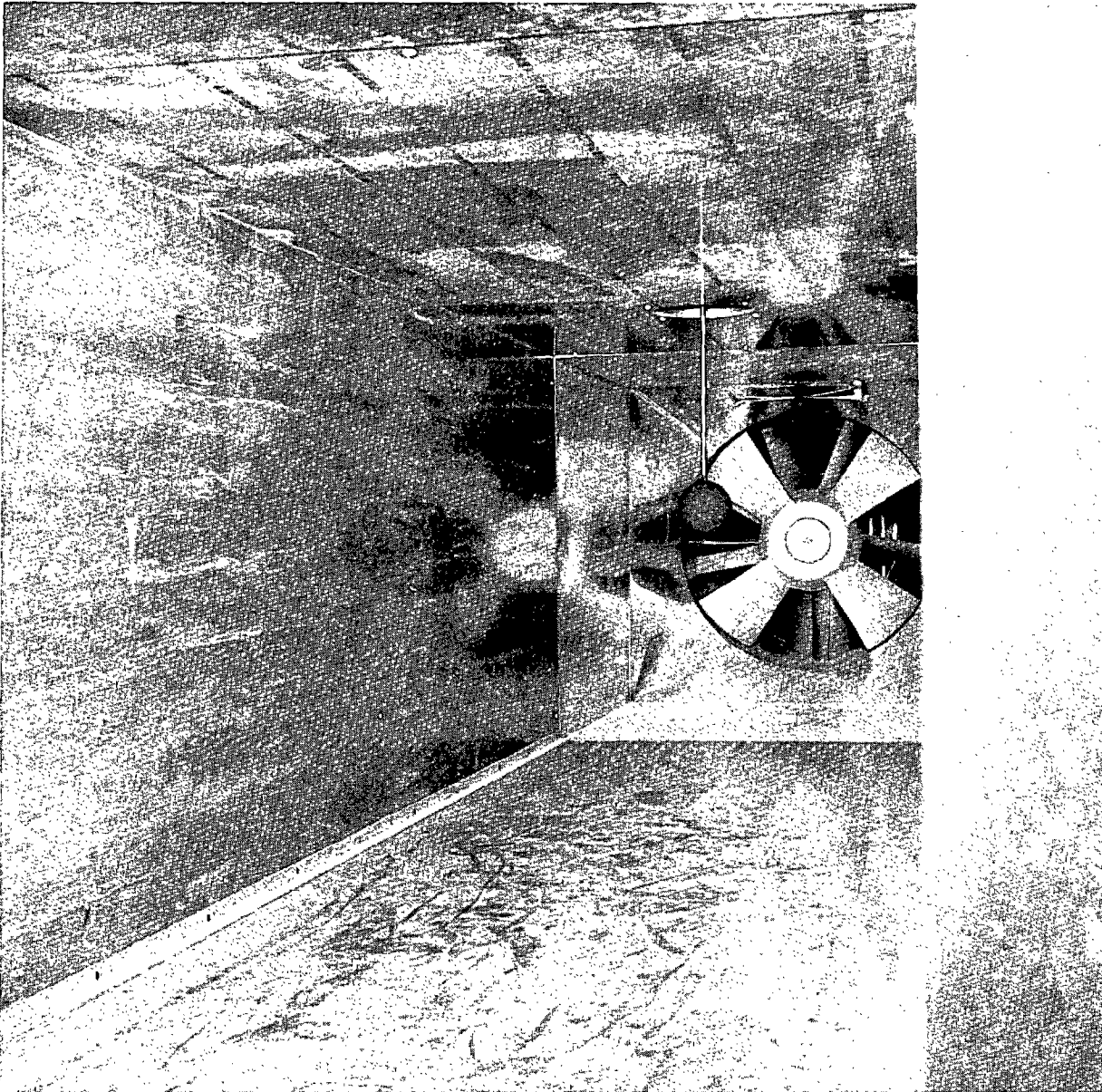


Figure 7

INSTALLATION OF GLOBE THERMOMETER IN TUNNEL

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constant resistance ratio of the hot and cold filaments. Its signal output is nonlinear with velocity. Consequently, it was calibrated before use in this program. This was done in an 8-in. duct at velocities up to 120 fpm by measuring the flow with a sharp-edged orifice plate using an inclined manometer. At velocities above 120 fpm, it was calibrated by comparison with the other anemometer at different velocities in the wind tunnel.

The other anemometer was manufactured by Thermo-Systems, Inc. of Saint Paul, Minnesota (Model 1053B). Its probe (Model 1330-60-18) used a heated platinum film on a quartz cylinder of a diameter equal to 0.006 in. Its operation depends upon the relationship between the cooling effect on the sensor and the mass flow rate, just as does the Flowtronic anemometer. It also consists of a power supply for heating the sensor, which is in a 0.1% bridge circuit. Its output signal is amplified. This anemometer was delivered to CAL early during this test program and had been calibrated over its full range by the manufacturer. Overall accuracy was about 2%.

During operation of either anemometer, its output jack was connected to a Consolidated Electrodynamics Corporation amplifier for impedance matching to a galvanometer to permit recording by oscillograph.

Heat Transfer Measurement on Spheres

Three spherical copper shells were instrumented with thermocouples for measurement of convective heat transfer coefficients on spheres. Three different shell diameters were used, 2, 4, and 6 in. Shell thickness of the 2-in. diameter sphere was 0.013 in. and the others were 0.016 in. Each thermocouple junction was created by drilling a 0.005 in. hole through the shell, inserting a constantan wire through it, and soldering it in place. When the wire was cut off at the outside and the shell surface polished, the presence of neither the hole nor the wire could be detected by naked eye. The thermocouple circuits were then completed by the use of a common copper lead wire soldered to the copper shell. Eleven thermocouples were formed in this manner and distributed around the equator of both the 4- and 6-in. diameter

spheres as shown by Figure 8. Eight thermocouples were used for the 2-in. diameter sphere and were placed in the same positions as for the other spheres except at $\theta = 30^\circ$, 70° , and 120° (see Figure 8). In all cases, thermocouple wire size was 3 mils to minimize errors caused by heat conduction along the wires.

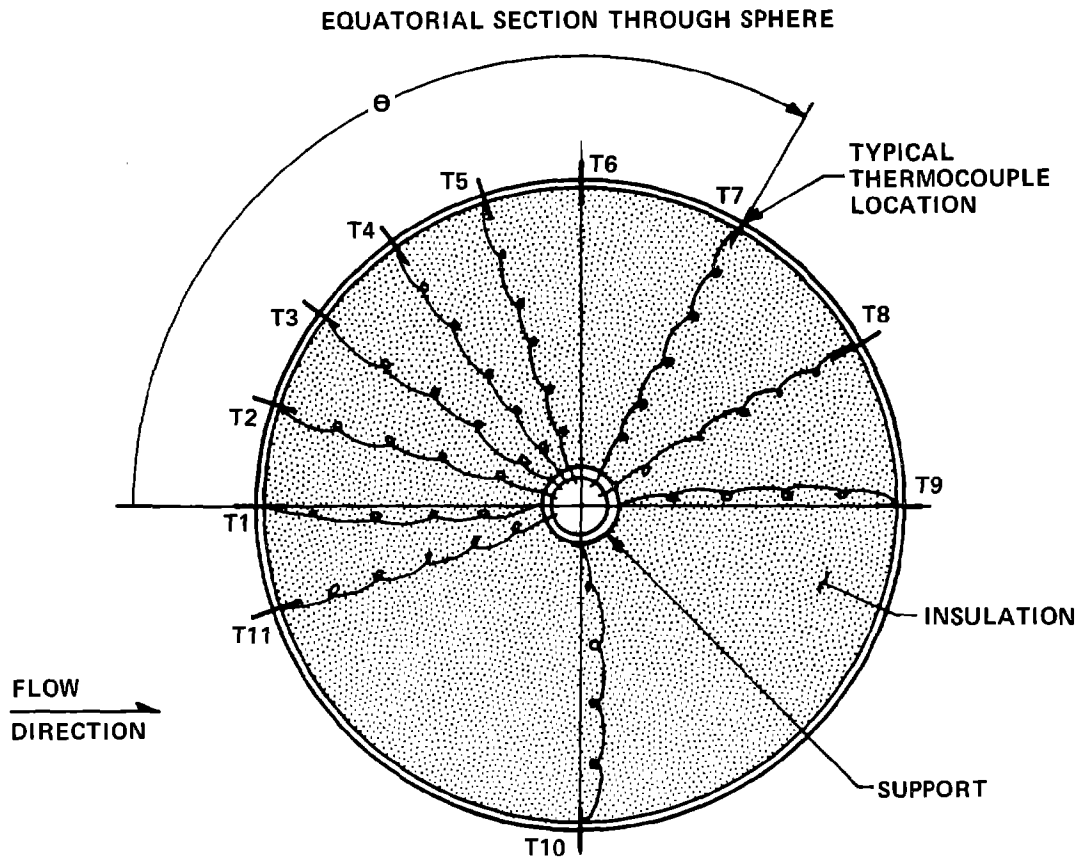
Each sphere was supported by a straight length of mechanical tubing which was 1/2-in. diameter (outside) for the 4- and 6-in. diameter spheres and 3/8 in. for the 2-in. sphere. The tubing extended through the sphere along a diameter that was oriented vertically when in use so that the circle of thermocouples was in the horizontal plane. The lower end of the tubing extended 18 in. below the center of the sphere so that it could make contact with the tunnel floor and thereby place the center of the sphere on the tunnel center-line when the sphere was lowered from a storage location above the tunnel.

All three instrumented spheres were stuffed loosely with glass wool insulation to prohibit radiative energy exchange among thermocouples, which would impair their use as heat meters.

The Globe Thermometers

A variety of globe thermometers were fabricated to provide for variations in globe material, thickness and diameter, as well as sensor type. The sensor was fixed so that it was centered within the sphere in all cases, as shown in Figure 9. Details are given in this figure for a removable thermocouple or thermistor sensor assembly that is held in the center location by a set screw that protruded into a groove. Each globe was identified by an upper case letter, and the key to this system is provided by Table I.

All globe thermometers were supported by a hollow, vertical tube 3/8 in. in outside diameter, which had a threaded joint about 6 in. from the globe to facilitate changing globes between tests. In all cases but one, this tube was soldered to the globe, including an aluminum globe. The exception was a 4-in. diameter copper globe which had a thermocouple attached to its



THERMOCOUPLE DESIGNATION	θ , ANGLE FROM FORWARD STAGNATION POINT - DEGREES
T1	0
T2	15
T3	30
T4	50
T5	70
T6	90
T7	120
T8	150
T9	180
T10	270
T11	345

Figure 8

INSTRUMENTATION LAYOUT FOR SPHERE MODEL

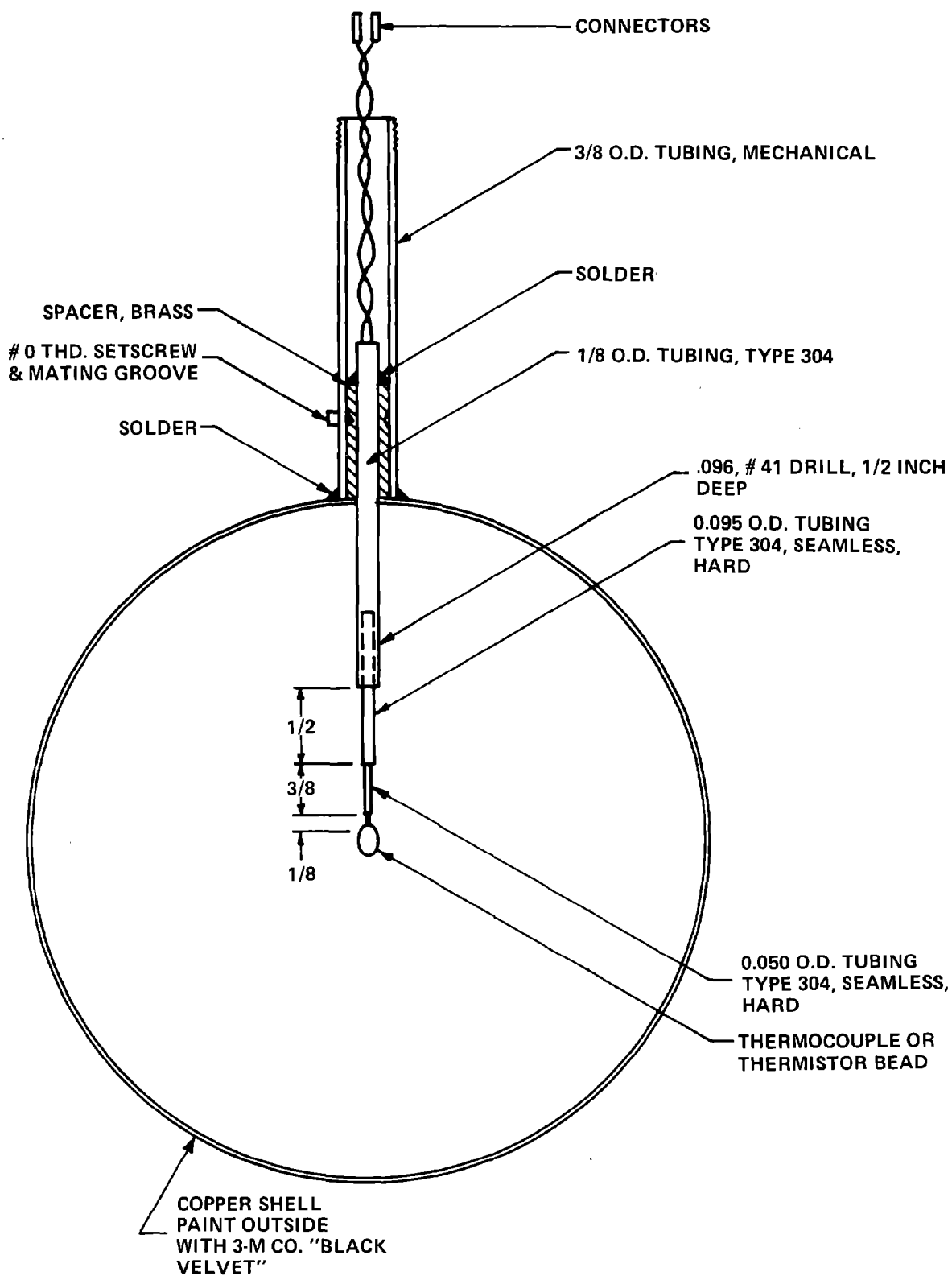


Figure 9
GLOBE THERMOMETER DETAILS

TABLE I
RESPONSE TEST CONFIGURATIONS

Designation	Shell				Sensor		Remarks
	Diameter in.	Thickness* in.	Material	Mode	Type	Mode	
A	4	0.022	Cu	BO	TS	Pl	Cl
B	4	0.016↙	Cu	BO	TS	Pl	
C	4	0.016	Cu	BOI↙	TS	3/4GB↙	
D	4	0.016	Cu	BO	TS	B↙	
E	6↙	0.016	Cu	BO	TS	Pl	
F	4	0.016	Cu	BOI	TS	Pl	Cl, TCS90↙
G	4	0.016	Cu	BOI	TS	B↙	Cl, TCS90
H	4	0.016	Al↙	BOI	TS	B	Cl, TCS90
I	4	0.015	Al	BO↙	TS	B	
J	4	0.016	Cu	BO	TC↙	Pl	
K	4	0.016	Cu	BOD↙	TS	B	
L	4	0.015	Al	BO	TM↙	Pl	
M	6	0.016	Cu	BO	TS	B↙	
N	Oblong↙	0.020	Cu	BO	TS	B	Toilet tank float

NOTES:

BO - black outside, B - black, BOI - black outside and inside, BOD - black and dented, Al - aluminum, Cu - copper, Cl - clamped to sting, Pl - plain, TC - thermocouple, TM - thermometer, TS - thermistor, 3/4GB - 3/4 in. dia. graphite ball, TCS90 - thermocouple on shell at 90° from stagnation point.

* Maximum variation is 0.002 in.

↙ - Indicates the characteristic that was different than previously.

inner surface at the equator (globe F in Table I), and this required that it be clamped to the support.

The thermocouples used as globe thermometer sensors during the program were made from 0.003-in. diameter copper and constantan wire welded together for a bare junction.

The thermistors used as globe thermometer sensors were purchased from Allied Radio Corporation. They were manufactured by Yellow Springs Instrument Company of Yellow Springs, Ohio. Each thermistor device consisted of a network of two thermistors (in a single bead) and two precision resistors for generating a linear signal between 0 and 100°C. The circuit used was one recommended by Yellow Springs on their instruction sheet. The bead was oblong in shape, 0.150 in. across its greatest length and 0.110 in. was its maximum dimension in the other direction. Its surface was smooth and shiny. Linearity deviation of the thermistor network is specified to be within $\pm 0.22^\circ\text{C}$. Absolute accuracy and interchangeability are within $\pm 0.15^\circ\text{C}$. Its sensitivity is $.005348 \text{ E}/^\circ\text{C}$ where E is the emf of the power supply.

A globe thermometer was fabricated from a toilet tank float by soldering a piece of 3/8 in. O.D. mechanical tubing to it at one end so that the tubing was aligned with its long axis. There was a flange at the midsection perpendicular to the long axis, and the float was oriented in the wind tunnel so that the plane of the flange was parallel to the flow.

The 4-in. diameter, 0.016 in. thick globe was dented to a prescribed depth on opposite sides to evaluate the performance of a severely damaged globe. Details of the dents are shown in Figure 10.

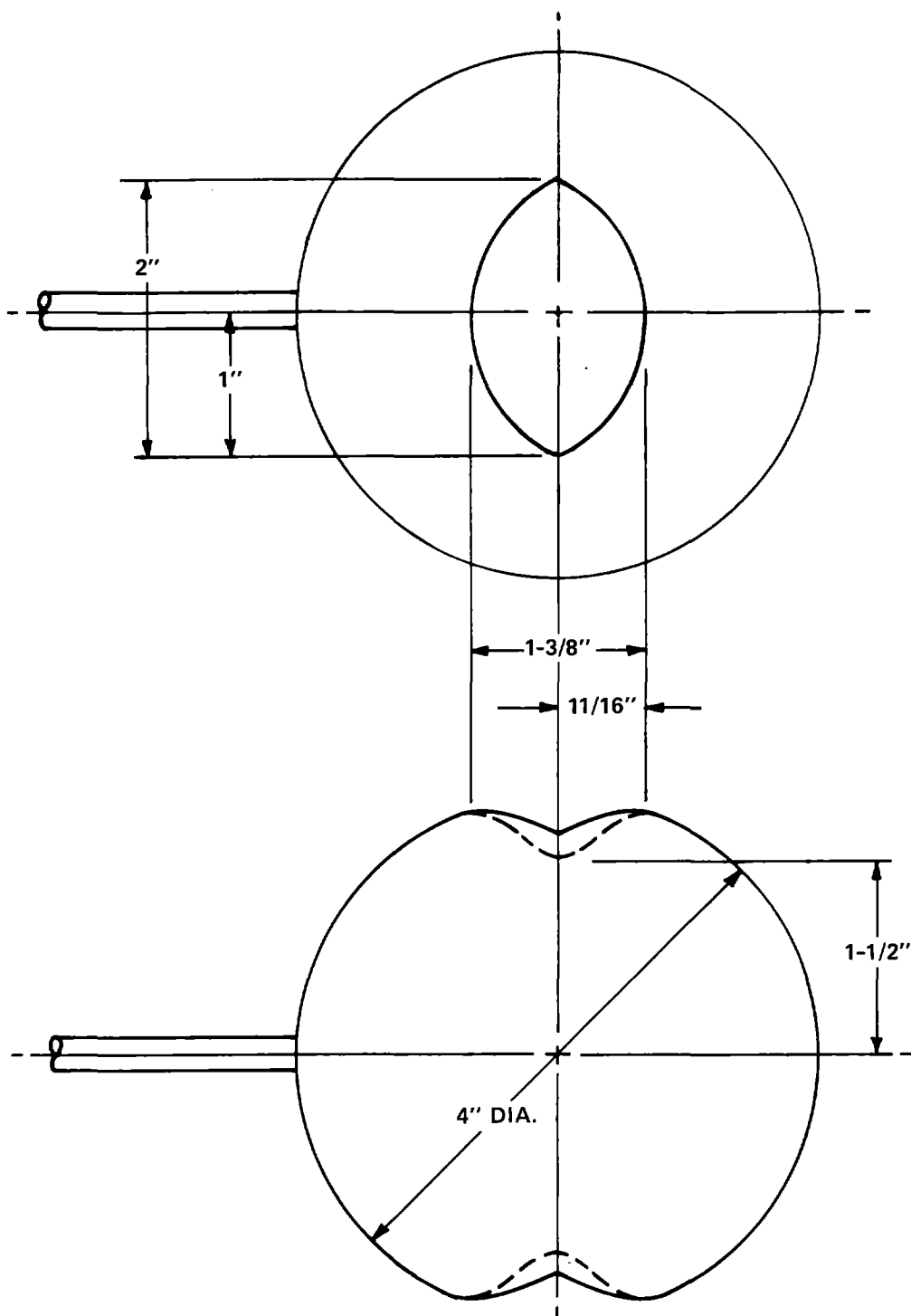


Figure 10
DENTED GLOBE THERMOMETER DETAILS

IV. EXPERIMENTAL PROCEDURES

The procedures of the experimental effort were specific to the test conditions in most cases and in at least one case, i. e., the measurement of convective heat transfer coefficients, the methods used were completely different from traditional methods. Therefore, some explanations are in order.

Heat Transfer Measurements

First, all measurements of convective heat transfer coefficient were done in the single-pass tunnel of Figure 1. To start a run, power to all heaters was turned on. The temperatures of the tunnel wall and radiant panel were preset to a level sufficient to produce a useful thermocouple signal, which was usually about 180°F. The model chamber was permitted to heat to a higher temperature. The sphere was raised into the model chamber and kept there until its temperature was slightly in excess of tunnel wall temperatures. When all temperatures were steady, wind speed was measured. The sphere was lowered rapidly into the air stream immediately afterward. One minute of temperature recording was sufficient to gather optimum data.

The temperature histories obtained for each thermocouple location were reduced to heat transfer rate by the use of the following relation:

$$q = \rho c \delta \frac{\Delta T}{\Delta t}$$

where $\Delta T / \Delta t$ is the rate of temperature change of the thermocouple. Use of this relation is justified because the shell is very thin and the gradient of heating rate along the surface is moderate so that heat conduction is essentially one dimensional, i. e., normal to the surface. Another consideration is that the Fourier modulus, $\frac{\alpha t}{\delta^2}$, becomes sufficiently large in a fraction of a second after exposure to the stream, to insure that the rate of temperature change is the same at any point along a radius within

the shell material, which is the necessary and sufficient condition for validity of this relation in one dimensional heat conduction (Reference 4).

After computing the heat transfer rate it was reduced to heat transfer coefficient by the definitive relation: $h = q/(T_s - T_{air})$. Both specific heat and h were evaluated at the average temperature during the time interval when rate of temperature change was measured. Each run was conducted so that this average temperature was the same or nearly the same as tunnel wall temperature. All surfaces were at the same temperature immediately after insertion of the sphere into the stream. There was no radiation exchange involved and no significant conduction losses through the sphere support. It is these conditions that are necessary to the validity of this technique for measuring convective heat transfer coefficient.

Response Tests

A variety of globe thermometers were subjected to known conditions in the wind tunnels for the purpose of measuring response characteristics. Each run was conducted by waiting for equilibrium temperatures to prevail for a given condition of wind velocity. Then the globe, which was preheated to 250°F, was lowered rapidly into the air stream and a temperature history of its sensor was recorded until it reached equilibrium. This was assured by continuing to record for several minutes after the temperature stopped changing. Of course, a continuous oscillograph record was obtained for globe thermometers having either thermistor or thermocouple sensors, but the liquid-in-glass thermometer had to be read at short intervals through a port in the wind tunnel wall.

Both wind tunnels were used for response tests but most of the runs were conducted in the single-pass tunnel. In fact, response of all except configurations K and N was measured in the single-pass tunnel.

Globe Thermometer Evaluation Tests

Tests designed to evaluate globe thermometers in a radiation environment were relatively simple. The desired measurements were radiative source temperature (tunnel wall temperature), stream air temperature, globe sensor temperature, and stream air velocity. All values were to be obtained when the temperatures were at equilibrium and the wind velocity was constant. Consequently, the conditions were preset, the globe thermometer was inserted into the stream, and temperatures were observed until they became steady.

Data Reduction for Evaluation Tests

The measured tunnel wall temperatures for each run were converted to a tunnel wall brightness temperature so that this could be compared with the mean radiant brightness temperature. First, the quantity $B = \bar{h}(T_s - T_a) + \sigma T_s^4$ was computed. Then, the average of all seven wall temperatures, T_{HW} , was obtained. Next, the radiative interchange factor was obtained from several runs for the same tunnel wall temperature by the following relation:

$$\bar{F}_{S, HW} = \frac{B}{\sigma} \left/ \left[T_{HW}^4 + \left(\frac{\epsilon_{CW}}{\epsilon_{HW}} \right) \frac{(1 - F_{S, HW})}{F_{S, HW}} T_{CW}^4 \right] \right.$$

where $F_{S, HW}$ is the angle or view factor for radiative interchange. T_{CW} is the temperature of the unheated openings viewed by the globe, which was assumed to be equal to the air temperature. The ratio $\epsilon_{CW}/\epsilon_{HW}$ was assumed to be 2.5. When a value of $\bar{F}_{S, HW}$ was obtained for several runs that was practically identical, it was used for obtaining the mean tunnel wall brightness temperature for all runs in the following expression:

$$(\bar{T}_{HW})^4 = \bar{F}_{S, HW} (T_{HW})^4 + \left(\frac{\epsilon_{CW}}{\epsilon_{HW}} \right) \frac{(1 - F_{S, HW})}{F_{S, HW}} (T_{CW})^4$$

This procedure may be described briefly as one that provides first, a solution for the radiative interchange characteristics of the surfaces in terms of measured quantities. These characteristics are dependent upon surface geometry and optical properties of the surfaces, which were not measured. Then the radiative interchange characteristic or factor was used to compute the tunnel wall brightness temperature that would be required to produce the same radiative heat flux as did the actual walls at temperature T_{HW} .

V. DISCUSSION OF EXPERIMENTAL RESULTS

Air Flow Quality

One of the anemometers (Flowtronic) used in the wind tunnels was capable of measuring velocity fluctuations up to 1000 cps. It showed that over most of the velocity range the amplitude of fluctuations was 12 percent of the average velocity.

Flow distribution across the test section was almost perfectly uniform in the single-pass tunnel, as would be expected. Traverses were made at several air velocities to demonstrate this. Flow distribution in the closed circuit tunnel was much better than expected; in fact, it was practically as good as in the single-pass tunnel at air velocities above 400 fpm. The velocity varied less than 2 percent in the region between any points 2 inches or more from the walls.

Flow distribution at lower air velocities showed a slight but constant slope as the probe was moved towards the inner wall of the tunnel loop, i. e., velocity decreased with distance. The slope measured about -0.8 percent velocity change per inch of traverse from centerline.

Distribution of Convective Heat Transfer Coefficients

The distribution of convective heat transfer coefficient was obtained for a large variety of conditions as well as for the three sphere diameters (2, 4 and 6 inch). These were especially desired for the higher velocities because these distributions indicate boundary layer transition to turbulence if it exists.

No evidence of transition was detected on any of the spheres. Typically, the heat transfer coefficient decreases with distance from the forward stagnation point and reaches a minimum at 90 degrees or beyond, depending upon velocity. The minimum value moves aft as velocity decreases. If transition had occurred, the distribution of heat transfer coefficient would show a rapid rise to values higher than those near the forward stagnation point. Actually, beyond the point where the minimum occurs the

rise is gradual with distance, and the heat transfer coefficient does not reach the same level at the aft stagnation point as at the forward one. Some of the distributions are shown in Figure 11a for the six-inch diameter sphere. Distributions over the 4 inch sphere are shown in Figure 11b.

It is interesting to compare these distributions with those obtained by other workers using different methods. For example, some very recent work was presented in Reference 5 for distributions of heat transfer coefficient on 2 inch spheres at Reynolds numbers within the range of interest here although stream velocities were higher. The minimum velocity for which data was presented was 6 meters/sec (about 1200 fpm). It is presented as the parameter $\frac{Nu}{\sqrt{Re}} / \left(\frac{Nu}{\sqrt{Re}} \right)_o$ but the Reynolds number here is a free stream value based upon diameter so that this parameter can be reduced to the ratio h/h_o , where the subscript o indicates the upstream stagnation-point condition.

Upon comparison a few differences stand out. First, the ratio of h/h_o is a minimum at $\theta = 90^\circ$ for the reference data at 6 m/sec whereas most of the data of this program reached a minimum at about 120° from the forward stagnation point. However, this is consistent with the trend of our data to show a minimum that is farther aft for lower velocity. Another significant difference is the level that h/h_o descends to at the minimum point. The smallest value obtained for h/h_o from the data of this program is 0.59 at 1030 fpm for the 6-inch diameter sphere, whereas the referenced work shows a value of about 0.17.

The copper shells used in this program probably could not support the steep lateral temperature gradients that are exhibited by such a distribution. Although this suggests that our data is in error, it affects only a small part of the distributions, and has little effect upon the subsequent integration of the distribution (which was the ultimate object).

Finally, the overall shape of the curve of Reference 5 is different, especially at the stagnation point where it is cusp-like. Heat transfer distributions obtained by other researchers, Reference 6 and 7, are shown in Reference 5 that are shaped like the plots of this report. In addition,

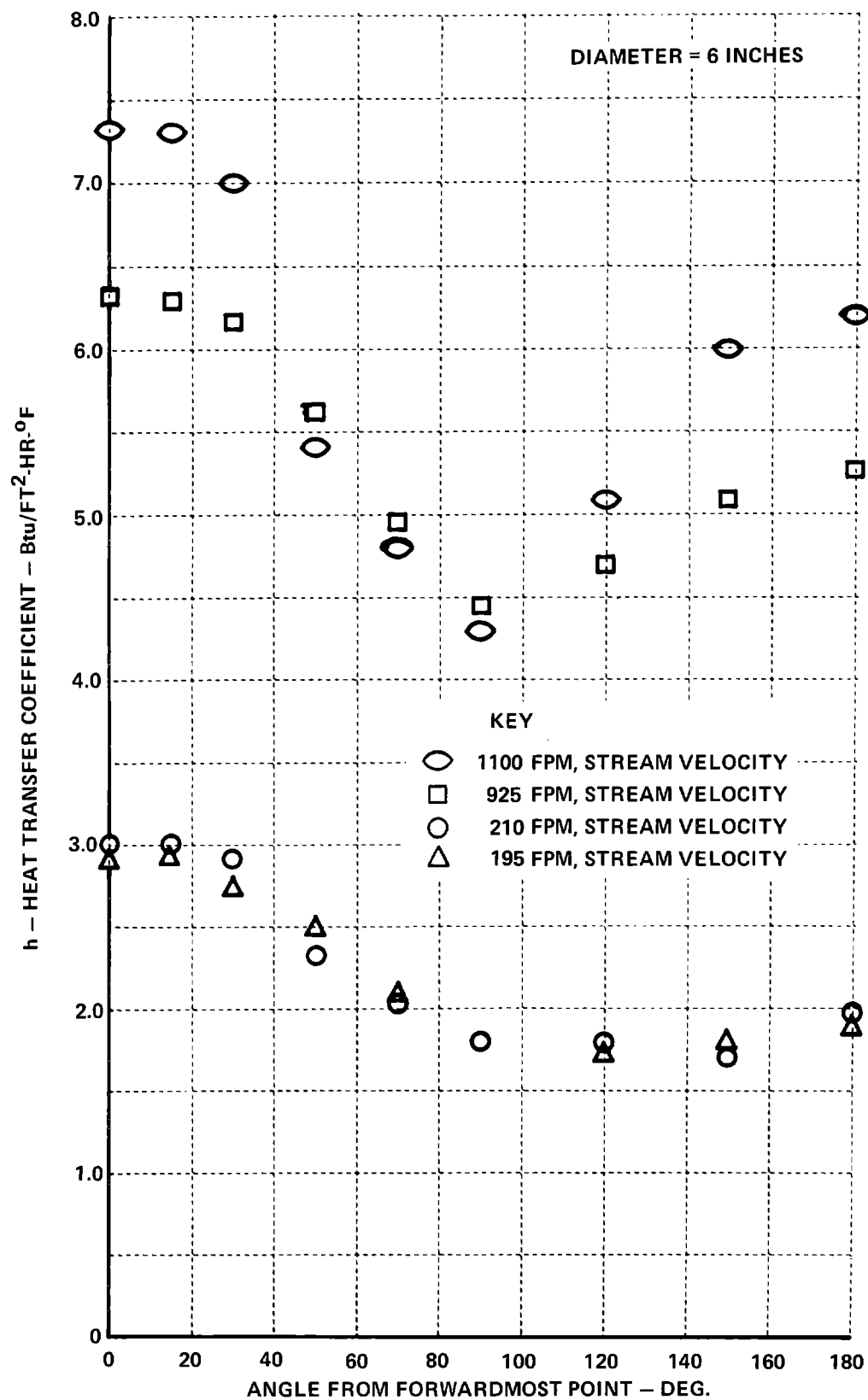


Figure 11a
LOCAL AERODYNAMIC HEAT TRANSFER COEFFICIENT ON
SPHERES IN AIR AT ATMOSPHERIC PRESSURE

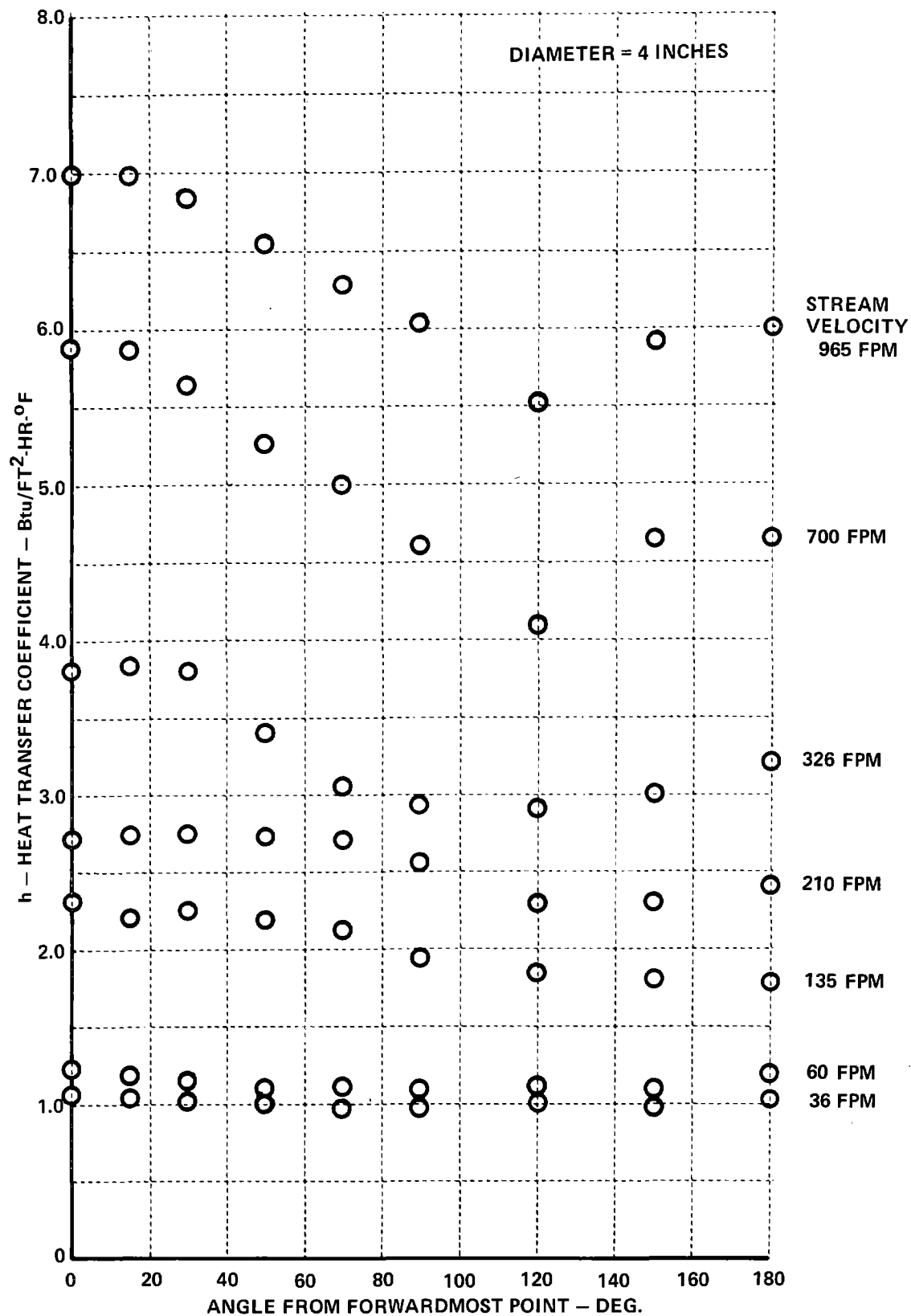


Figure 11b
LOCAL AERODYNAMIC HEAT TRANSFER COEFFICIENT ON
SPHERES IN AIR AT ATMOSPHERIC PRESSURE

some calculations were made based upon theory that yielded results exactly like those of References 6 and 7, and consequently, ours. It can be concluded, therefore, that our data is reasonably well-behaved.

Integrated Heat Transfer Coefficients

In order to obtain average heat transfer coefficients for spheres, as were needed for subsequent development of globe thermometer formulae, the values for $h(\theta)$ were integrated over the semicircle. The integral relation for average heat transfer coefficient is

$$\bar{h} = \frac{1}{2} \int_0^{\pi} h(\theta) \sin \theta \, d\theta$$

where θ is the angle formed between radii from the local point and the forward stagnation point. Integration was carried out numerically. Average Nusselt number (Nu) was then obtained, and the corresponding Reynolds number (Re) for each run was computed. Results are plotted in Figure 12 and are summarized in Table II. Also shown are test results obtained by Lavender and Pei, Reference 8, who made a careful study of heat transfer from spheres that were submerged in air exhibiting various degrees of free turbulence. Only their low turbulence data are shown in Figure 12 (turbulence intensity $< 3\%$ @ $Re > 10,000$). In their paper, comparisons were made with data of other workers, and their conclusions were that the correlation between Nu and Re should be

$$Nu = 2 + 0.629 (Re)^{0.5} (Re_T)^{0.35} \text{ for } Re_T < 1000$$

and

$$Nu = 2 + 0.145 (Re)^{0.5} (Re_T)^{0.25} \text{ for } Re_T > 1000$$

where Re_T is the turbulent Reynolds parameter $= \frac{d^2 \sqrt{u^2}}{\nu}$, d is diameter, u is the fluctuating velocity component, and ν is kinematic viscosity.

Data obtained by Raithby and Eckert, Reference 9, for spheres supported by a tube oriented in crossflow with respect to the stream are

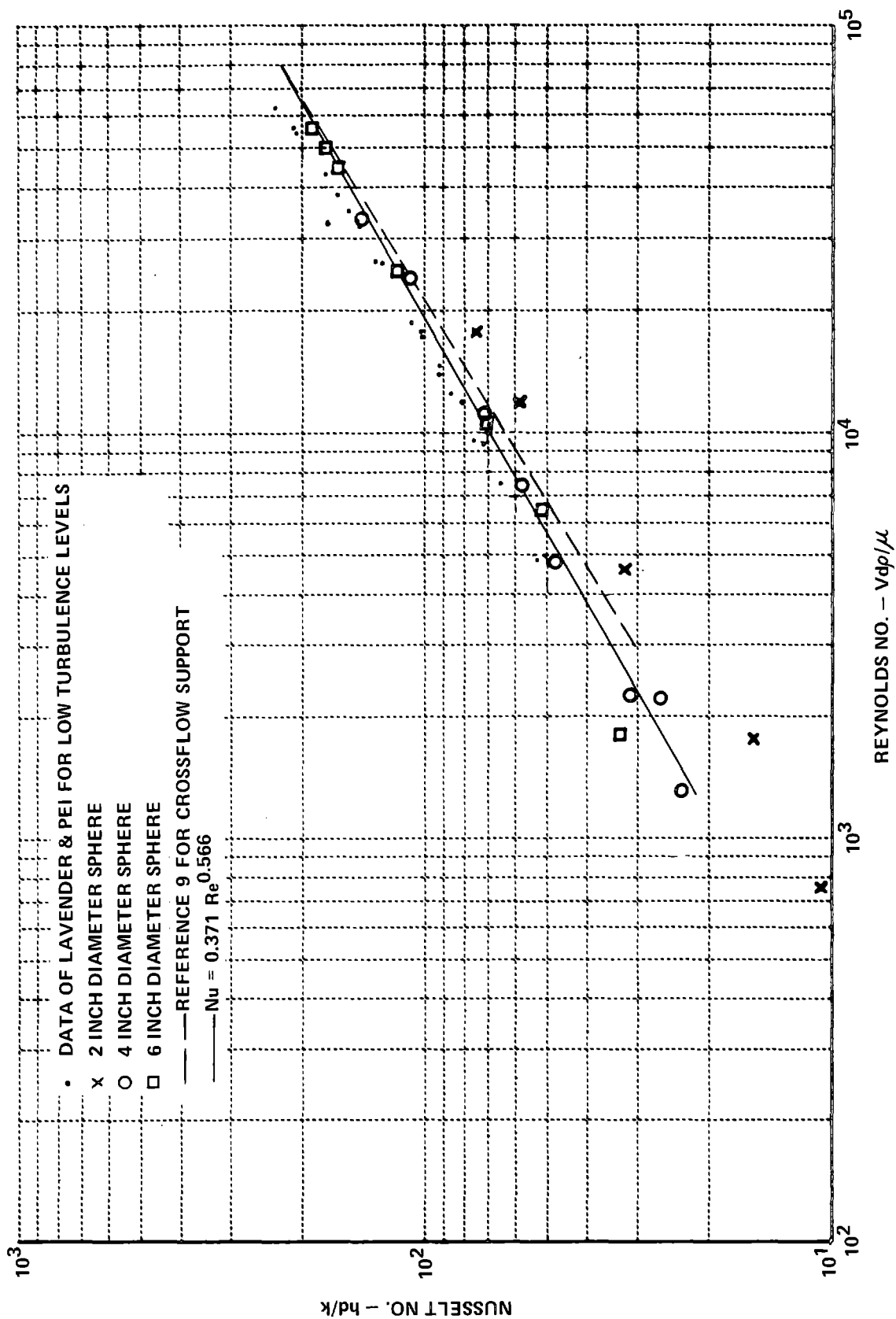


Figure 12
INTEGRATED HEAT TRANSFER RATE ON SPHERES

Table II
HEAT TRANSFER CORRELATION DATA

RUN NO.	SPHERE DIAMETER inches	STREAM AIR			$Nu = \frac{hd}{k}$	$Re = \frac{Vd\rho}{\mu}$
		VELOCITY fpm	TEMPERATURE °F	PRESSURE inches mercury		
11	6 ↓	925	58.0	29.73	174.5	49,200
12		210	55.0	29.73	71.0	10,700
13		93	53.0	29.73	—	5,000
14		130	56.0	29.21	52.1	6,400
18		34	49.0	29.21	33.0	1,800
19		1050	55.0	29.10	164.0	54,600
20		1100	53.0	29.10	194.5	56,000
21		470	50.0	29.10	117.5	25,050
23	4 ↓	965	54	29.14	141.3	33,700
24		210	52	29.14	58.1	7,410
25		135	52	29.14	48.0	4,770
26		326	60	29.39	71.5	11,300
32		36	48	29.38	23.3	1,300
34		61	46	29.38	26.3	2,200
37		1260	59	28.91	—	—
38		700	58	28.91	109.0	24,200
43	2 ↓	250	45	29.66	36.7	4,620
44		94	45	29.66	15.3	1,730
45		700	54	29.24	59.5	12,300
46		700	52	29.24	58.9	12,300
48		1060	54	29.22	80.3	18,200
50		44	55	29.24	10.70	760

also shown on Figure 12. It can be seen that our data for 4 and 6 inch diameter spheres agrees best with these results. This statement is still valid if comparison is made with results of other workers, of which several are represented in comparisons made in Reference 9. At low Reynolds numbers results are somewhat scattered, probably because of flow disturbances arising from natural convection currents within the stream. These were moderated considerably by the use of the convergent nozzle with its 12 x 12 inch exit located 10 inches upstream from the sphere. Even so, smooth uniform flow could not be obtained between 30 and 70 fpm. All data for the 2-in. diameter sphere depart considerably from the other data in spite of the fact that tests were made in smooth uniform flow.

In searching for an explanation for the anomalous behavior of the data from the 2-inch diameter spheres (Figure 12) other references were studied. Some interesting and significant facts were revealed but none are totally satisfying. Apparently, the best explanation of this behavior is that the shell thickness of 0.013 inch was excessive for the small diameter so that lateral heat conduction effects were not insignificant, as would be required for good heat transfer measurements.

In order to use the heat transfer data in a formula for mean radiant temperature it must first be fitted with a mathematical relation. The minimum Reynolds number of interest is 410 (15 fpm, 140°F air, 4 inch diameter). However, no dependable data are shown on Figure 12 for Reynolds number less than 1300. An expression of the type $Nu = 2 + A Re^r$ would be fitted only if it was required to accommodate very low Reynolds numbers. The relation $Nu \propto Re^r$ is more convenient for further development of formulae. Consequently, the relation $Nu = 0.371 Re^{0.566}$ is proposed. It fits the data well in the medium range of Reynolds, and favors the low Reynolds number data, while agreeing well with the Reference 9 relation at high Reynolds number.

It is desirable to convert this relation to an expression for heat transfer coefficient in terms of sphere diameter, temperature, and velocity. This can be done by making curve fits for viscosity and thermal conductivity of air in

the temperature range from 70°F to 140°F. The relations $\mu = 3.978 \times 10^{-4} T^{0.75}$ lb/hr-ft and $k = 0.626 \times 10^{-4} (T)^{0.872}$ Btu/hr-ft-°F satisfy the requirements (T is absolute temperature, °R). Using the equation of state, the following relation for heat transfer coefficient can be derived for sea level pressure:

$$\bar{h} = \frac{0.1365 V^{0.566}}{d^{0.444} T^{0.101}}$$

where V is in fpm and d is in feet. Obviously, temperature has a weak effect. If the average temperature of 105°F is used to evaluate $\bar{h}$, the error incurred for either extreme temperature is only 1/2 percent. In this case, the relation for a 6-in. diameter sphere is:

$$\bar{h} = 0.098 V^{0.566} \quad 70^\circ\text{F} \leq T \leq 140^\circ\text{F}$$

This is quite different than the Bedford and Warner relation, Reference 2, which is $\bar{h} = 0.169 \sqrt{V}$. The value of $\bar{h}$ computed from this new relation is lower than that obtained from the Bedford and Warner formula by an almost constant value of 0.45 over most of the velocity range. The new value is only 10% lower at 1000 fpm but it is 27% lower at 100 fpm.

Two subjects that were studied during the literature search dealt with the effects of supports and free air turbulence upon convective heat transfer from spheres.

Reference 9 shows results for heat transfer from spheres with crossflow supports that were 10 percent higher than heat transfer from spheres with rear supports. The parameter, support diameter divided by sphere diameter, caused no appreciable change over the range $0.11 \leq d/D \leq 0.17$. In Reference 5, a flow visualization test picture for a sphere with $d/D = 0.048$ shows that on the surface where $\theta^* = 75^\circ$, separation occurs and the flow is disturbed by the presence of a crossflow support, even though the support is relatively small. No evidence has been found in the literature of any effect upon heat transfer at $d/D > 0.17$, although this possibility cannot be dismissed. However, the d/D ratio of our 2-in. sphere was only slightly larger, being equal to 0.19.

Of potential significance, the Lavender and Pei data, used in the figure for comparison, were criticized in Reference 9 because, in obtaining it by the method of measuring the power to maintain the temperature of a heated sphere exposed to the airstream, a guard heater was not used for their supports, causing another increase of 10 percent. This criticism cannot be easily leveled at the apparatus used in this program because both the sting and sphere were initially at the same temperature prior to test.

The discussion of supports is relevant to globe thermometers, inasmuch as the globe must have at least one support so that it is not in contact with other surfaces and to isolate it from any auxiliary apparatus that might disturb the prevailing air currents of the environment being studied. The direction of these currents relative to the globe thermometer will, in practice, be different for every placement, if the support is fixed and if no effort is made to orient it relative to the direction of the flow. Therefore, a 10 percent uncertainty in the heat transfer effect is possible, if the results of Reference 9 are valid.

The effect of free-air turbulence upon the heat transfer rate to spheres appears to be a subject of continuing study. Reference 9 is a recent report on this subject that includes a review of previous work. It was concluded that increased turbulence intensity causes increased overall Nusselt number and the increase is more pronounced at the higher Reynolds numbers. In addition, an effect of resonance between free-stream eddies and vortex shedding has been revealed. The pertinent parameter for studying this effect is the ratio of the sphere diameter to the scale of turbulence, D/L_f . Resonance was estimated to occur at $D/L_f = 8.4$, and its effect was to increase Nusselt number. At low turbulence intensity (2 percent), the maximum increase is only 1 percent, while at a turbulence intensity of 7 percent it can be estimated to be 10 percent for spheres with rear supports at $Re = 27,000$. At lower Reynolds numbers the effect was definitely reduced; for example at $Re = 14,000$ and turbulence intensity = 7 percent, the increase in Nu was 4 percent.

A large initial jump in Nusselt number was noted when turbulence was introduced to the stream using rear supports, whereas this jump did not occur for crossflow supports. Apparently, the effect was masked by

the disturbances caused by the supports. This would be an argument in favor of the use of crossflow for the globe thermometer, since the uncertainty caused by neglecting turbulence would be reduced.

Concerning the 2-in. sphere data, little can be said about the contribution of turbulence to the departure of the data because turbulence scale has not been measured, except to speculate that the 6- and 4-in. spheres might have been near resonance, causing Nusselt numbers to be higher than for the 2-in. sphere. The fact that the bulk of our data ranks among those results of other workers showing the highest Nusselt numbers lends credence to this statement. Another consideration is that many of the test spheres used by other researchers were 2 in. in diameter or smaller, and References 6 and 8 are typical in this respect.

In summary, the greatest error that would be incurred by the use of a globe thermometer with a crossflow support would be 10% in a highly turbulent, rapidly flowing stream. It would be unusual to find such conditions during the use of a globe thermometer. An example might be found in the discharge of a large fan near its outlet.

Response of Globe Thermometers

The globe thermometers that were tested for response, i. e., time lag, include all configurations of Table I except K, N and M, which differed from the other 6-in. diameter configuration only because its thermistor was black. Response was obtained at three or four air speeds, and the results are given by Table III.

Several significant observations can be made. First, the thinner globe has faster response--at 200 fpm air speed the time to equilibrium was 12.5 minutes for the 0.016-in. thick copper globe compared to 18 minutes for the one with 0.022-in. wall. Second, slightly shorter response time (1/2 minute faster) was recorded for the 6-in. globe as compared to the 4-in. one. Third, no perceptible change could be detected as a result of painting the thermistor with "Velvet Black." Fourth, coating the internal surface of the shell with "Velvet Black" had no effect upon response. Fifth, the thermocouple, when

TABLE III
RESPONSE TEST RESULTS

Run No.	Config.	Description	Air Vel., fpm	Equil. Temp., °F	Final Air Temp., °F	Time Lag min.
59	C	Lg. TS, 0.016" Cu shell - 4" d.	210	92	53	33
60	C	" " "	156	97	55	34
61	A	Pl. TS 0.022" Cu shell - 4" d.	156	96.5	56.5	18
62	A	" " "	192	91.0	56	18
63	A	" " "	990	74	56	14
64	A	" " "	90	118.5	55	20
65	B	Pl. TS, 0.016" Cu shell - 4" d.	200	94	53	12.5
66	B	" " "	285	89.5	52	13
67	B	" " "	1280	71.5	53	9
68	D	Bl. TS, 0.016" Cu shell - 4" d.	210	113	74	12
69	D	" " "	290	102.5	77.5	13
70	E	Pl. TS, 0.016 Cu shell, -6" d.	210	118.5	74	11.5
71	E	" " "	1270	91	70	8.5
72	E	" " "	750	101	71	9.5
73	D	Bl. TS, 0.016" Cu shell - 4" d.	750	98.5	70	9.5
74	F	Pl. TS, 0.016" Cu shell-4"d., TC	210	?	67	?
75	F	" " "	720	TC 95 TS 94	67	10.5
76	F	" " "	1260	TC 90 TS 90	66	9
77	F	" " "	200	TC 104 TS 100	58	13
78	J	Pl. TC, 0.016" Cu shell - 4" d.	200	100	58	13
79	J	" " "	760	85	59	9.5
80	J	" " "	1280	99	76	9.0
81	J	Pl. TC, 0.016" Cu shell - 4" d.	720	98.5	72	10
82	L	TM, 0.015" Al shell - 4" d.	720	101.5	77	21
83	L	" " "	210	116.7	75.5	21.5
84	L	" " "	1260	93	72.3	20
85	I	Bl. TS, 0.015" Al shell - 4" d.	1260	87.5	72.5	7.5
86	I	" " "	720	95.5	72	8.5
87	I	" " "	210	112.0	71.5	9.5

used as the sensor, responded at a rate similar to the thermistor. Sixth, the time to equilibrium is increased as air speed is decreased. Seventh, the shell reaches equilibrium a minute or two before the central sensor; at least this was indicated to be the case for the point of thermocouple attachment. Finally, response is better for the aluminum shell than for the copper shell of practically the same thickness by at least 1 minute at any speed. In fact, the aluminum globe thermometer was the fastest one tested.

The globe using a liquid-in-glass thermometer exhibited a very long response time in spite of its aluminum globe. For example, the time to equilibrium was 21 minutes at 700 fpm air velocity.

The thermometer used in globe L was subjected to a response test by first storing it in a freezer at 0°F for an hour. It was removed and fixed horizontally in a room (where air was at 80°F) so that the bulb was suspended in midair, free of contact with any material. Its return to room temperature was timed and was found to require 15-1/2 minutes. Obviously, the glass thermometer is the limiting factor that determines response characteristic of a globe thermometer that uses it. Response might be improved by stirring the air inside the globe, but such a provision would compromise the greatest advantage of this type of globe thermometer, namely its simplicity.

Reduced time to reach equilibrium (absolute response time) is a very worthwhile goal. Inasmuch as it has been Federally mandated that the globe thermometer be used to monitor heat stress in industry, it is anticipated that intensive use will result and repetitive readings will be involved, which would otherwise be time-consuming. The results of this research indicate that the shell is the component that is critical to any effort for reducing absolute response time to less than 12 minutes. The properties of the shell that are significant here are its density, specific heat, and thickness. Metals exhibit a limited range of the product: density, ρ , times specific heat, c . For copper, $\rho c \simeq 55 \text{ Btu/ft}^3 \cdot ^\circ\text{F}$, which is near the upper end of the range, whereas aluminum has a ρc product of 35, representative of the low extreme. Obviously, the use of aluminum has improved response, but by a limited amount; and

little improvement over aluminum is in prospect by substituting other materials. Consequently, the only means remaining is to use thinner walls.

Thin-walled globe thermometers will be highly susceptible to damage from handling. The rigidity of a spherical shell increases with decreasing radius for a given material, although at the present time, the 4-in. diameter seems a reasonable compromise. Tempering this damage susceptibility is our demonstration that a shell can sustain severe denting and still perform satisfactorily. This is described in the next section.

VI. DEVELOPMENT OF EQUATION FOR MEAN RADIANT TEMPERATURE

Derivation

Equation 3, the simple heat balance equation, involves the mean radiant temperature, T_w , for which an explicit solution may be made. This was done during the early development of the globe thermometer, Reference 1, using the same heat balance equation, which had been obtained directly by more or less intuitive means. Furthermore, an expression for the average heat transfer coefficient was available at the time, which was $\bar{h} = 0.169 \sqrt{V}$. Using $\alpha = 0.95$ for both the sphere and the radiative source, the traditional formula for the globe thermometer was produced as:

$$T_w^4 = T_s^4 + 0.1028 \times 10^9 \sqrt{V} (T_s - T_a)$$

for a six-inch diameter globe.

The results of the experiments of this program revealed that $\bar{h} = 0.0717 V^{0.566} / d^{0.444}$. Substituting this relation in the simple heat balance yields the following formula for mean radiant temperature:

$$T_w^4 = \frac{1}{\alpha_w} \left[\alpha_s T_s^4 + 0.415 \times 10^8 \frac{V^{0.566}}{d^{0.444}} (T_s - T_a) \right]$$

At this point it is pertinent to question the definition of mean radiant temperature, as it concerns the concept of a "black body" source of radiation. It can be seen from the above relation that an absorptivity of the source, α_w , is involved. This seems pointless because it probably is safe to predict that no user of the globe thermometer will ever make any effort to measure it, and the failure of the user to consider its measurement probably will be unrelated to the fact that it is a difficult measurement to make even when it is possible. Furthermore, it is obvious that there is no relation between the emission characteristics of the source and those of the globe thermometer as implied in the derivation for the traditional formula. The logical conclusion of this reasoning is that it makes more sense to discuss a "brightness"

temperature of the equivalent enclosure, i. e., the temperature directly corresponding to the radiative energy, without considering its source characteristics. Then the mean radiant temperature is

$$(T_w)^4 = \alpha_s T_s^4 + 0.415 \times 10^8 \frac{V}{d} \frac{0.566}{0.444} (T_s - T_a) \quad (4)$$

Comparison Between Theory and Experiment

The basic intent of the experiments to evaluate globe thermometers at a variety of conditions was to expose them to a known environment of convective cooling and radiative heating. The convective cooling environment became a known condition as a result of the correlation of data obtained from the measurement of heat transfer coefficients on spheres. However, the radiative environment was not completely known because the emissive characteristics had not been determined and this limited somewhat the scope of the evaluation.

When a small object is located within a large enclosure the emissivity of the surfaces of the enclosure is effectively unity because all reflected energy is returned to its source for re-emission. However, a small opening in the enclosure upsets this situation by permitting energy loss, and this may be aggravated in the case of specular reflections. Because the wind tunnels used in these experiments were not complete enclosures, the emissive power of their heated surfaces was substantially reduced. Consequently, their wall temperature did not constitute a mean radiant temperature that could be measured.

One method that was used to interpret the data was an attempt to simulate a comparison between the mean radiant temperature that is computed from the formula derived above and the measured tunnel wall temperature. This was done by computing the radiative interchange factors required to force a balance between heat radiated to the globe thermometer from the walls and the combination of heat lost by the globe due to its radiation and to convective cooling of the globe. This is the basis for the procedure presented in Section V. Of course, the effect is to achieve a comparison between a corrected value for

the measured wall temperature and the computed mean radiant temperature through the use of the radiative interchange factors. However, it should be stated that the best this method can achieve is to illustrate the consistency of the results and that it is not the only method by which this may be done.

The effectiveness of the comparison depends upon the discovery that the total cooling effect of the globe, i. e., the sum of $\alpha_s \sigma T_s^4$ and $\bar{h}(T_s - T_a)$, exhibited very little deviation from a mean value at a given tunnel wall temperature. This happened regardless of the other conditions, although if tunnel wall temperature was changed, this mean value also changed. Consequently, for each condition of tunnel wall temperature, a run was selected that exhibited a mean value for total cooling, and its data was used to compute the radiative interchange factor for the experimental arrangement at that tunnel wall temperature. The results were as follows: for $T_w = 185^\circ\text{F}$, $\bar{F} = 0.61$; for $T_w = 260^\circ\text{F}$, $\bar{F} = 0.55$; and $T_w = 360^\circ\text{F}$, $\bar{F} = 0.50$.

The only assumption that was made in the data reduction procedure was that emissivity of the unheated radiating areas was 2-1/2 times that of the heated walls. This was based mostly upon the fact that much of the unheated area was distant from the globe being tested and was very much like a cavity, which of course would exhibit an emissivity close to unity. Furthermore, a precise value for this was not needed because the incident radiation from the unheated areas was less than 2% of the total except at the lowest levels of tunnel wall temperature.

The previously mentioned consistency of the total cooling effect is manifested by generally good agreement between mean radiant temperature and the mean tunnel wall temperature. This is understandable when it is considered that mean radiant temperature is a result of the expression for a heat balance. The quality of the results can be appreciated by referring to Table IV where computed mean radiant temperature is presented beside the mean tunnel wall brightness temperature, i. e., the measured value corrected to include the radiant interchange factor and an emissivity of unity. The two results are nearly the same for all runs with the notable exception of runs 72, 94, 100, 107, and 109.

Table IV
EVALUATION OF GLOBE THERMOMETERS

RUN NO.	GLOBE DIAMETER inches	GLOBE TYPE	STREAM AIR VELOCITY fpm	T _a STREAM AIR TEMPERATURE °F	T _w TUNNEL WALL TEMPERATURE °F	(T _w - T _a) °F	GLOBE THERMOMETER TEMPERATURE °F	MEAN TUNNEL WALL BRIGHTNESS TEMPERATURE °F	COMPUTED MEAN RADIANT TEMPERATURE °F
100	4	I	40	164.0	360	196.0	234.0	235.0	253.0
64		A	90	55.0	260	205.0	118.5	167.0	165.0
65		B	200	53.0	260	207.0	94.0	167.0	162.5
68		D	210	74.0	260	191.0	113.0	167.0	170.0
88		I	220	71.5	260	188.5	112.0	167.0	170.0
69		D	290	77.5	260	183.5	102.5	167.0	165.5
107		I	295	114.0	350	246.0	153.0	225.0	210.0
109		L	295	114.5	350	245.5	155.0	225.0	211.0
106		I	630	126.0	260	134.0	137.5	166.0	162.5
110		L	600	140.0	360	220.0	173.0	234.0	234.0
79		J	760	59.0	260	201	85.0	167.0	171.5
86		I	770	72.0	260	188	95.5	167.0	171.5
97		I	850	150.0	360	210	177.7	235.0	243.0
96		I	920	127.0	260	133	137.0	158.0	165.0
94		I	1000	122.0	185	63	124.5	115.5	126.0
67		B	1280	53.0	260	207	71.5	167.0	160.0
98		I	1160	121.0	360	239	151.0	234.0	231.0
103	6	M	128	127.0	260	133	151.0	165.0	166.5
102		M	150	102.5	185	82.5	111.0	115.0	115.0
70		E	200	74.0	260	186	118.5	167.0	170.0
89		M	220	71.5	260	188.5	114.5	167.0	168.0
108		M	295	114.0	350	236	162.5	225.0	223.0
104		M	580	115.0	260	145	130.5	165.5	167.5
111		M	600	145.0	360	215	183.0	235.0	240.0
113		M	575	143.5	360	216.5	182.5	235.0	243.0
105		M	640	124.0	260	136	138.5	166.0	166.5
101		M	790	93.5	185	91.5	100.5	115.0	117.5
72		E	750	71.0	260	189.0	101.0	167.0	179.0
92		M	760	71.5	260	188.5	99.0	167.0	175.0
112		M	840	110.0	365	255.0	154.5	241.0	249.0
71		E	1270	70.0	260	190.0	91.0	167.0	168.0
93		M	1290	72.0	260	188.0	93.0	167.0	171.0

NOTE: RUNS 94 AND HIGHER WERE CONDUCTED IN THE CLOSED CIRCUIT TUNNEL.

The property that is peculiar about Run 94 is the tunnel wall brightness temperature, which is considerably lower than the globe temperature. This indicates that the tunnel walls may have some nonuniformity in radiative output. However, there may also be a contribution by accumulated errors, for example, the difference, $(T_s - T_a)$ is very small, only 2.5 degrees, so that errors in temperature measurement can have a significant effect in this case.

The results of Run 100 are relatively good when the flow quality and precision of velocity measurement for such a low speed are considered. Again, it may be pointed out that secondary currents caused by natural convection inside the tunnel caused some flow disturbance at low speeds.

No explanation has been obtained for the behavior of the results of Runs 107 and 109. Run 108 was conducted on the same day using the same instruments and it is in much better agreement so the fault is not likely to be in instrument malfunction. However, the results are well within the desired 3% on an absolute temperature basis.

It was interesting that although a great variety of globe thermometers was fabricated, they all performed similarly regarding the equilibrium temperature exhibited. This is a conclusion that may be obtained from the general agreement among the results of Table IV because several different globe thermometers are represented by those results. For example, Runs 88 and 89 were run under exactly the same conditions with a 4 inch and 6 inch diameter globe respectively, using the same thermistor in close sequence. Another example is Runs 107 and 109, where two different sensors were used. The globe for Run 107 was a 4 inch diameter aluminum shell using a thermistor. The globe for 109 was a 4 inch aluminum shell with a liquid-in-glass thermometer.

In addition, the comparisons made during Runs 115 to 120 (results are presented in Table V), showed that the toilet float globe indicates the same equilibrium temperature within 1°F as a 4-inch diameter copper globe under the same conditions, and finally, that severe dents in a 4-inch diameter globe had no perceptible effect.

Finally, it should be pointed out that it is the globe thermometer temperature for the 6-inch diameter globe that has been treated in most of the literature and has been accepted by the agencies of the U. S. Government to be used in compliance with heat-stress regulations. A globe of diameter other than 6 inches will indicate a different temperature for a given set of conditions. However, it is possible to correct for the effect of diameter through the use of the formula for mean radiant temperature and an estimated velocity. The procedure is an iterative one because the formula cannot be solved explicitly for T_s , globe temperature. It is described as follows.

First, solve for mean radiant temperature, T_w , using Equation 4, pg. 45, by substituting the estimated velocity and the value for $(T_s - T_a)$ that was measured by using the off-size globe thermometer. Then solve for T_s from the following formula, using the estimated velocity and measured $(T_s - T_a)$:

$$T_s^4 = \frac{T_w^4}{\alpha_s} - 0.437 \times 10^8 \frac{v^{.566}}{d^{.444}} (T_s - T_a)$$

This will yield a value for T_s that is slightly high for undersize globes. Consequently, at least one more solution must be made, using a value for T_s halfway between the previously computed value and the measured one. A third computation may be made if the difference between the computed values is felt to be excessive, but it will be found that the iteration converges rapidly. Of course, conveniences such as a nomogram would speed up this procedure rapidly.

It is obvious that in the usual case where it is the 6-inch globe reading that is desired, and if many readings are involved, the use of an off-size globe thermometer introduces unwarranted additional work.

Table V
EVALUATION OF UNORTHODOX GLOBE THERMOMETERS

Run No.	Globe Diameter inches	Globe Type	Stream Air Velocity fpm	Stream Air Temperature °F	Tunnel Wall Temperature °F	Globe Thermometer Temperature °F
115	4	D	775	118.5	260	137.0
116	—	N	775	118.0	↓	136.0
117	—	N	65	141.0		173.5
118	4	D	65	139.5		172.5
119	4	K*	65	140.0		171.5
120	4	K**	775	119.0		137.0

* Globe oriented with dents at stagnation regions

** Globe oriented with dents 90° from stagnation regions

VII CONCLUSIONS

The scope of the program for the development of globe thermometry was a comprehensive one and required investigation of most of the technical aspects of globe thermometers. Specific objectives have been fulfilled without departing from the original program plan. It was demonstrated that design of globe thermometers is not very critical to their performance by fabricating a wide variety of globe thermometers, all of which provided similar temperature readings for a given set of conditions. Furthermore, the globe thermometer yields a predictable temperature reading over a wide range of known conditions of air temperature, air velocity, and radiative intensity. This, in fact, was a significant result of a major effort of this program, which was to evaluate the globe thermometer at specified conditions. Other conclusions are detailed below in connection with specific objectives.

First, the effects upon time lag of several parameters were measured and it was determined that:

1. Time lag was reduced by decreasing globe thickness.
2. Time lag was maintained at a minimum by the use of either a miniature thermocouple or a tiny thermistor as a sensor. It was large when a liquid-in-glass thermometer was used.
3. Time lag was little affected by either sensor emissivity or globe diameter.

Various other means of reducing time lag in globe thermometers were evaluated with the following conclusions:

1. Time lag was reduced by the use of an aluminum shell. It is not likely that any other shell

material of the same thickness could reduce time lag further.

2. Time lag was not affected by emissivity of the inside surface of the shell.

It was established that a good design for a globe thermometer of minimum time lag would be one with a 0.015-inch thick aluminum globe, 4 inches in diameter. It used a thermistor 0.15 inch long, located in the center of the globe. This globe thermometer required 10.5 minutes to reach equilibrium in an air stream flowing at 720 fpm after preheating it to 260°F. However, its usage would require correction to the 6-inch globe reading by means of the formula for mean radiant temperature.

The 260°F starting condition is an extreme one that was employed to exaggerate differences in response caused by changes in globe thermometer design. It would be highly unusual to start from this condition in any conceivable application of the globe thermometer. Consequently, the time lag of globe thermometers during use would always be less than the values reported herein.

Second, the heat transfer coefficients that were measured on 4 and 6 inch diameter spheres correlated well by a straight line on a log-log plot. It was found that recent results of similar work agreed well with the correlation, which fell about 10% below their upper bound and placed favorably among them. The data correlated well over the whole velocity range, demonstrating the overall stability of convective coefficients. A particular phenomenon of significance to stability is boundary-layer transition to turbulence, but no evidence of this was found in the distributions of convective coefficients at air velocities below 1100 fpm on the largest sphere tested.

Third, as a consequence of the straight line correlation for convective heat transfer coefficients, it was possible to obtain a single relation for mean radiant temperature, similar to the traditional one, although it

was derived from more fundamental concepts of physics. It was also demonstrated experimentally that the derived relation for mean radiant temperature could be used to compute the brightness temperature of a radiating enclosure from temperatures of a globe thermometer located within it. In addition, this served to show that consistent results could be obtained from any of a variety of globe thermometers, including one made from a toilet tank float. Consequently, ease of manufacture can be a chief consideration in the choice of an optimum globe thermometer for a specified usage.

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NOMENCLATURE

		<u>Typical Units</u>
A_i	Area of i^{th} element of an enclosure	ft^2
A_s	Area of sphere surface	ft^2
c	Specific heat	$\text{Btu/lb} \cdot ^\circ\text{F}$
d	Support diameter	
D, d	Sphere diameter	ft
F_{ij}	Shape factor for radiation exchange between elements i and j	
$\bar{F}_{ij}$	Radiation interchange factor for element i with j	
$h =$	$q_c / (T_s - T_a)$, heat transfer coefficient	$\text{Btu/ft}^2 \cdot \text{hr} \cdot ^\circ\text{F}$
i, j	Indices to identify elements of an enclosure	
k	Thermal conductivity	$\text{Btu/ft} \cdot \text{hr} \cdot ^\circ\text{F}$
q_c	Convective heat transfer rate	$\text{Btu/ft}^2 \cdot \text{hr}$
q_r	Radiative heat transfer rate	$\text{Btu/ft}^2 \cdot \text{hr}$
s	Distance from the forward stagnation point	
T	Temperature	$^\circ\text{R}$
T_w	Mean radiant temperature of an enclosure	$^\circ\text{R}$
T_{wi}	Temperature of i^{th} element of an enclosure	$^\circ\text{R}$
T_s	Temperature of globe shell	$^\circ\text{R}$
t	Time	min.
V, v	Velocity	ft/min
x	Distance in x direction	
y	Distance in y direction	
α	Radiative absorptivity	

NOMENCLATURE (Cont.)

		<u>Typical Units</u>
ϵ	Radiative emissivity	
ρ	Density	lb/ft^3
μ	Absolute viscosity	lb/sec-ft
σ	Stefan-Boltzmann constant	
θ	Angle formed between radii from the local point of interest on the surface and from the forward stagnation point.	degree
δ	Thickness	

APPENDIX I

ERROR ANALYSIS FOR EXPERIMENTAL VARIABLES

An error analysis was made for the globe thermometer to determine the experiment measurement precision required by the 3 percent accuracy specification of the contract. The analysis was based upon the traditional formula for mean radiant temperature. It was started by use of the accepted criterion for evaluating system accuracy, i. e., the standard deviation*, S, which is computed from the formula:

$$S_v = \sqrt{\left(\frac{\partial V}{\partial x}\right)^2 (s_x)^2 + \left(\frac{\partial V}{\partial y}\right)^2 (s_y)^2}$$

where $V = f(x, y)$ is any variable. In this case, T_w is of interest. The total derivative for mean radiant temperature, $T_w = T_w(T_s, v, T_a)$ is

$$dT_w = \left(\frac{4T_s^3 + k\sqrt{v}}{4T_w^3}\right) dT_s + \left(\frac{kT_s - kT_a}{8T_w^3\sqrt{v}}\right) dv - \left(\frac{k\sqrt{v}}{4T_w^3}\right) dT_a$$

where the coefficients are partial derivatives, which are used to evaluate standard deviation. This was done for the extreme values of the test conditions and it was found that the combination $T_a = 140^\circ\text{F}$, $T_w = 70^\circ\text{F}$, $v = 1000$ fpm, was a critical one. For this case, $T_s = 587.3^\circ\text{R}$ and the accuracy requirement of 3 percent on T_w is 15.9°F . To meet this requirement, the deviations must be less than 0.3 percent for T_s , 0.33 percent for T_a and 2 percent for v . In other words, this establishes the precision for measurement of these quantities during test at the given conditions. At other conditions, the 3 percent requirement is not so exacting.

The heat transfer results reported herein showed that velocity is involved in the above relations by the 0.566 power instead of 1/2, used above. The use of the new relation for T_w was checked and it was found that the deviations are practically the same as given above. Consequently, it may be stated that the experimental measurements of this program were within the precision established by the analysis, above.

*See Beers, Y., Theory of Error, Addison Wesley Press, 1956.

Nomenclature

k	constant of proportionality
S	standard deviation
T_a	ambient air temperature
T_s	temperature of globe shell
T_w	mean radiant temperature
v	air velocity
V	dependent variable
x	independent variable
y	independent variable