



Thermistor anemometry including effects of ambient pressure and humidity change
by Billy Ray Cousineau

A thesis submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of
MASTEE OF SCIENCE in Mechanical Engineering
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Abstract:

The thermistor and its use in anemometry is explained. Selection and use of an electrical circuit for a thermistor anemometer is covered in detail.

A thermistor anemometer was constructed and calibrated. A closed system was constructed so that the parameters of interest for this research could be isolated and controlled.

The effects of ambient temperature, humidity and pressure must be known before the thermistor anemometer can be used with any degree of confidence.

Temperature was held constant in the closed system since the electrical resistance of the thermistor is a function of temperature.

Pressure was made to vary from 0 to 1 psig and the effect on the accuracy of the velocity determination is noted. The magnitude of this effect is shown by both experimental and analytical results.

Humidity was varied from 0 to 100% relative humidity and the effect on the accuracy of the velocity determination is shown to be negligible. This is verified by both experimental and analytical results.

THERMISTOR ANEMOMETRY INCLUDING EFFECTS
OF AMBIENT PRESSURE AND HUMIDITY CHANGE

by

BILLY R. COUSINEAU

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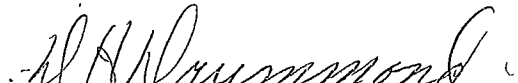
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Abstract

The thermistor and its use in anemometry is explained. Selection and use of an electrical circuit for a thermistor anemometer is covered in detail.

A thermistor anemometer was constructed and calibrated. A closed system was constructed so that the parameters of interest for this research could be isolated and controlled.

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INTRODUCTION

Anemometers are instruments used to determine velocity, generally used for low-velocity air streams. Hereinafter, the anemometer will be referred to with its most predominant use understood; measuring the velocity of air at or near atmospheric pressure.

Those in use at present consist of three basic types; mechanical, electro-mechanical, and solid state. The first two devices are characterized by moving parts and have mechanical and electrical outputs respectively.

An example of a mechanical device is the familiar vane type anemometer. This device consists essentially of a rotating element whose speed of rotation varies with the velocity of flow. The relation of speed of rotation to air velocity is found by calibration. This type of device extracts mechanical work from the air stream.

Solid state anemometers have no moving parts and usually operate on the principle that the cooling rate of a heated object is a function of the velocity of the air moving past the object.

An example of a solid state anemometer is the hot-wire anemometer. A hot-wire anemometer consists essentially of a heated wire sensing element and associated circuitry to measure changes in the electrical resistance of that element.

The sensing element consists basically of a thin (0.001 to 0.005 inch diameter) electrically heated wire stretched between the ends of two prongs. When this wire is exposed to an air stream, it loses heat by convection. The wire changes resistance upon cooling and the resistance change is thus a function of the air flow.

The circuitry used to measure the resistance change of the sensing element depends on the type of operation selected for the sensing element. In one method of operation, the wire can be maintained at a constant temperature by adjusting the current through the wire to offset the changes in the rate of convective cooling. The measured value of current through the wire may be used to determine the air flow. The wire is used as one leg of a Wheatstone Bridge. To maintain the wire at constant temperature and therefore constant resistance requires the use of an amplified alternating-current Wheatstone Bridge because of the small resistance change in the wire. Another method of operating the hot-wire anemometer sensing element is to maintain a constant heating current in the wire and allow the resistance to change. This causes a voltage change across the wire which must be amplified and accurately determined. The voltage change is a function of the air flow.

Either type of operation of the hot-wire anemometer requires expensive electronic equipment.

Thermistor anemometers are also solid state. In principle, they are identical to the hot-wire anemometer except that the resistance of the wire increases with temperature and the resistance of the thermistor decreases with increasing temperature. The thermistor performs the same function as the wire. The main advantage is that the resistance change of the thermistor per unit of temperature change is ten times greater than that for metals. See Chapter 1.1. This allows the use of simple electronic circuitry when compared to the hot-wire anemometer.

The effects of ambient conditions (i.e., pressure - temperature -

-XX-

humidity) must be known to accurately use the solid state anemometer. A closed system was constructed for this research so that these parameters could be isolated and investigated independent of each other.

Temperature change is discussed but is not a part of the experimental research. Research has been previously preformed in this area and time limits did not allow the extensive research which would have been required to reach a conclusion.

Pressure and humidity are experimentally investigated as no previous research is known to have been performed in this area. The results of this research show that pressure changes have a significant effect on the heat transferred from the thermistor but humidity changes cause only a negligible effect on the heat transferred.

CHAPTER 1

THERMISTOR ANEMOMETER THEORY

1.1 - THERMISTOR CHARACTERISTICS

Thermistors are semiconductors which exhibit an extremely large change in resistance with a small temperature change. They can be made with either a positive or negative temperature coefficient of resistance. In general most thermistors are now made with a negative temperature coefficient (i.e., an increase in temperature causes a decrease in resistance). This is opposite to the effect of temperature change on the electrical resistance of metals.

Thermistors are made from specific mixtures of very pure metallic oxides such as iron, cobalt, copper, nickel, manganese, magnesium, titanium, and uranium. A mixture of these oxides in the desired shape is sintered under carefully controlled conditions of temperature and atmosphere. The sintered piece has leads applied by a method such as silver soldering or ultrasonic welding.

Standard thermistor forms now available are:

Beads. Beads are small ellipsoids of thermistor material formed on two fine wires. These wires become electrical leads after the material is sintered. Some beads are coated with glass for protection. They can be sealed into the tips of glass rods to form probes. Resistance values of 300 ohms to over 100 megohm are produced in beads.

Disc. Discs are formed by pressing thermistor material in a round die to form a flat round configuration. The thermistor material is then sintered. Leads are attached to the two flat sides.

Washer. Washers are formed like discs except that a hole is formed in the center so that the washer may be placed over a bolt.

Rod. Rods are formed by extruding thermistor material through dies to make long cylindrical units. The thermistor material is then sintered.

There are three important characteristics, which follow, that thermistors exhibit that make them useful in electronic circuits.

Resistance - Temperature Characteristic:

The resistance of a thermistor is a function of its temperature.

The resistivity variation follows the empirical equation:

$$\rho_t = \rho_o e^{(\beta/t - \beta/t_o)}$$

where ρ_t = resistivity at temperature t ,

ρ_o = resistivity at temperature t_o ,

β = constant which varies for different thermistor materials

but is usually in the order of 4,000,

e = constant (2.718).

The temperature coefficient of resistance of a thermistor " α " is expressed in the following equation:

$$\alpha = \frac{1}{R_o} \frac{dR_o}{dT} \text{ ohm/ohm/}^\circ\text{C}$$

which is approximately equal to $-\frac{\beta}{T^2}$.

where T = absolute temperature,

R_o = resistance at a reference temperature.

"The value of ' α ' may also be expressed in %/ $^\circ\text{C}$ and in some cases is as high as -5.8% at room temperature as compared to

0.36% for platinum."¹ See Figure 1 on page 4.

Voltage - Current Characteristic:

When a current flows through a thermistor, the resistance obeys Ohm's Law until the power is sufficient to increase the temperature of the thermistor above that of its surroundings. When the current is increased beyond this point, the resistance of the thermistor is lowered due to its increase in temperature. See Figure 2 on page 5.

Current - Time Characteristic:

When a voltage is applied to a thermistor and fixed resistor in series, a current flows which is determined by the total resistance of the circuit. If the current is sufficient to cause self-heating in the thermistor, its resistance will be lowered which causes more current to flow in the circuit. After a period of time, a point will be reached where the power dissipated in the circuit balances the power applied and the current flow cannot increase. The time to reach this point is determined by the voltage applied, the resistance of the thermistor and its thermal inertia, and the value of the fixed resistor. See Figure 3 on page 6.

Thermistors are inherently quite stable and reliable devices when used properly. They are being used on various military and private missile and space projects where high reliability is essential. They are subject to two types of damage; thermal and physical shock. Suitable mounting and

1. "Thermistor Manual", Fenwal Electronics, Inc., EMC-5.

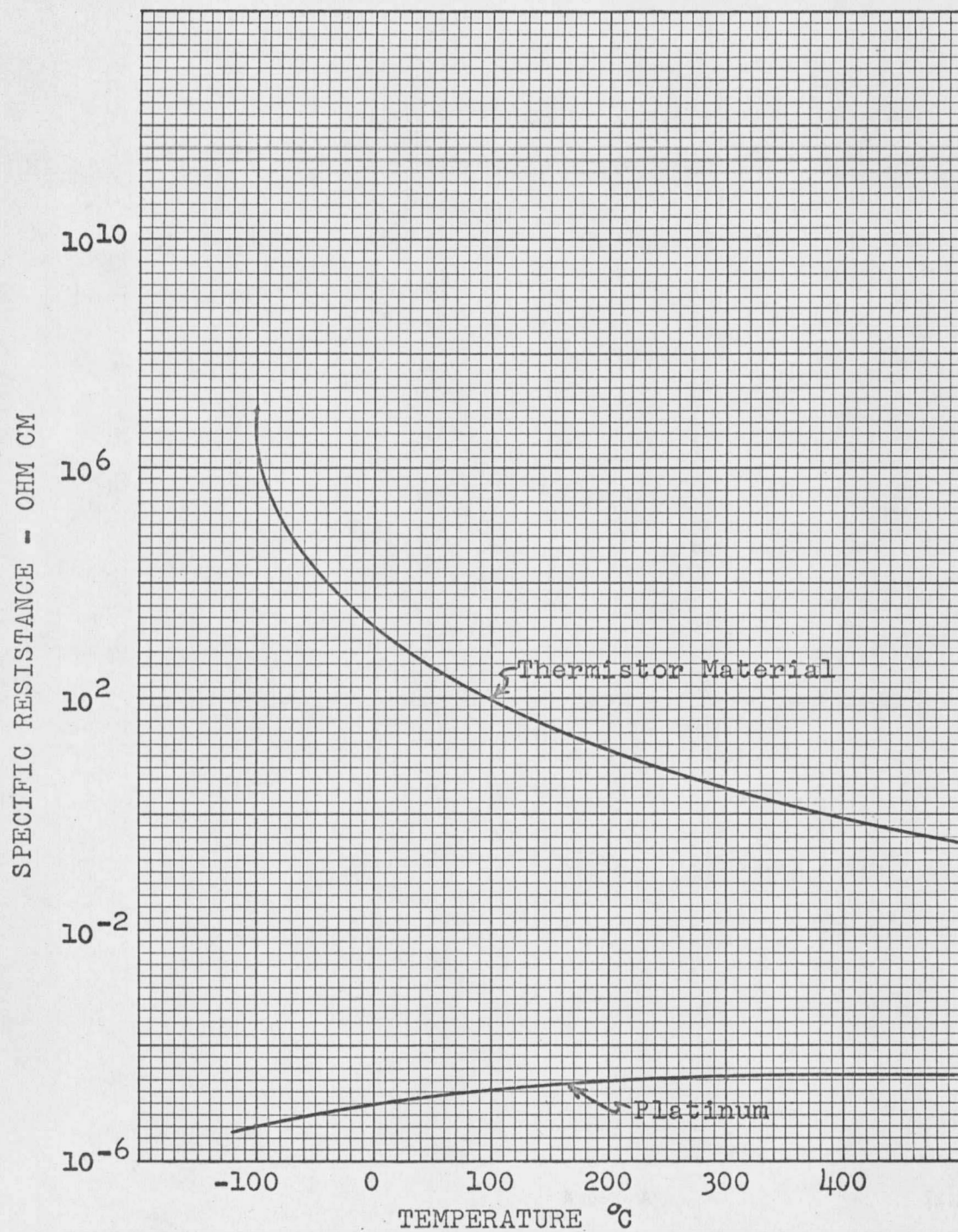


FIGURE 1. RESISTANCE - TEMPERATURE CHARACTERISTIC¹

1. "Thermistor Manual", Fenwal Electronics, Inc., EMC-5.

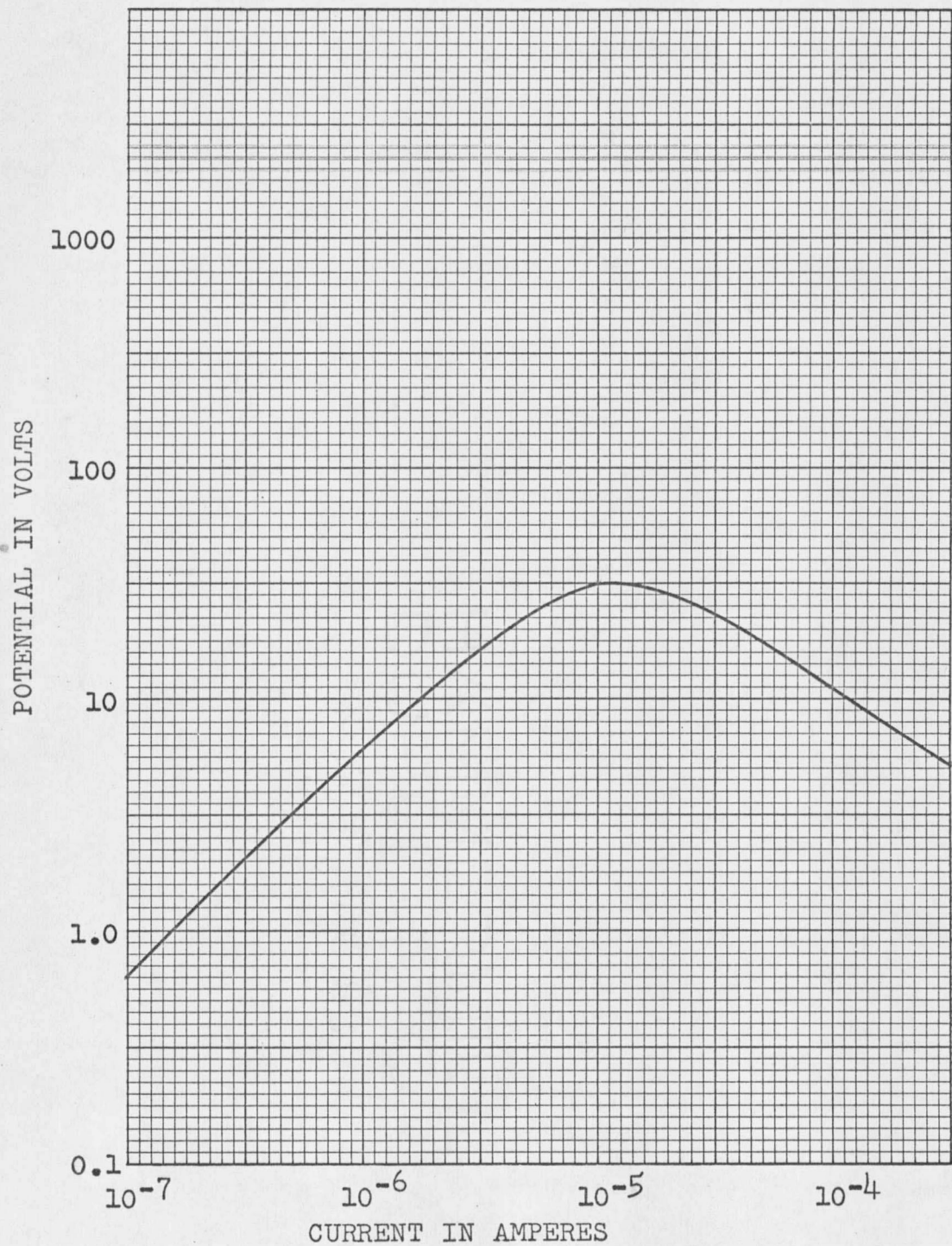


FIGURE 2. VOLTAGE - CURRENT CHARACTERISTIC¹

1. "Thermistor Manual", Fenwal Electronics, Inc., EMC-5.

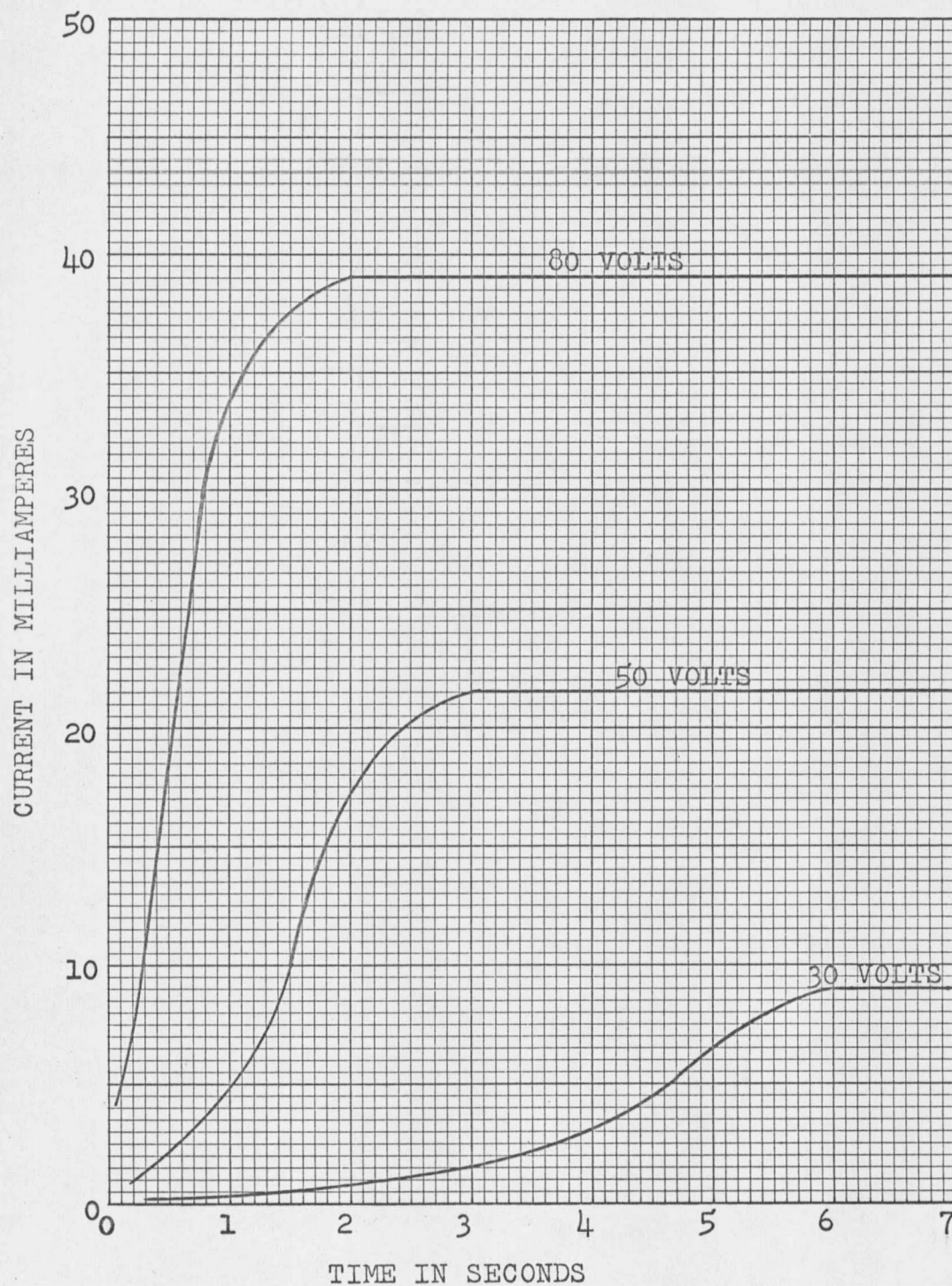


FIGURE 3. CURRENT - TIME CHARACTERISTIC¹

¹. "Thermistor Manual", Fenwal Electronics, Inc., EMC-5.

encapsulation of the thermistor will minimize the effects of physical shock. Thermal shock appears to be the major cause of failure.

There are two effects of thermal shock on a thermistor:

- 1.) Changes in the electrical and/or thermal contact between the thermistor material and the attached electrical leads.
- 2.) Changes in the physical impurity level and/or lattice structure of the thermistor material.

The change in electrical contact appears to be dominant. However, bead thermistors which have thermistor material and leads sealed in glass are much more stable in this respect than other types.

Most thermistors of the oxide type exhibit an aging effect. This is in the form of an increase in resistance with time. As much as 0.5 to 1.5% resistance change may occur during the first week of use. Changes of as much as 1% may still occur after several months. They may be pre-aged by exposing them for a period of time to a temperature slightly higher than that at which it is to be used. The length of time should be determined by the thermistor manufacturer.

Pre-aged thermistors may vary by as little as 0.2% per year. Bead thermistors enclosed in glass are less subject to aging effects than other types. Humidity, which may have an effect on aging of thermistors, is not significant to glass enclosed beads as the glass makes an effective moisture seal.

1.2 - SELECTION AND USE OF THERMISTORS

As discussed in Chapter 1.1, thermistors are available in a variety of sizes and types. In addition, custom-made thermistors can be obtained from several different manufacturers. There is no general rule covering what thermistor to use in a certain application and the designer must make this decision. Any thermistor manufacturer will help in selecting a thermistor for a given application.

Reference material on previous research on thermistor anemometers indicates that bead thermistors are generally used for this type of device. One reason for this preference is the ease of mounting a bead thermistor. They are small in diameter (0.01 inch for a glass enclosed bead) and present little restriction to the air flow. They can be quite easily suspended in the center of the air flow by mounting them with epoxy cement to a small plastic or metal probe. The probe is then positioned as desired in the air flow. See Figure 16 on page 50.

As discussed in Chapter 1.1, glass enclosed beads are more stable than bare beads which would be an advantage for anemometer work. In addition, the glass protects them from any reaction with the media to which they will be exposed.

There are many points to consider when using thermistors but the proper procedures for testing them cannot be overemphasized. One who is not experienced with thermistors would do well to read "Considerations in Testing Thermistors" by M. L. Miller, available from Victory Engineering Corporation, before attempting any work with them.

Temperature control is very important in obtaining good results with

thermistors. An error in temperature measurement of 1 °C can result in a fairly large error in the measured resistance. A Fenwal Electronics, Inc., GA51P68 thermistor as used in this research, has a temperature coefficient " α " of -4.6%/ °C at 25 °C. At an operating temperature of about 295 °C, the resistance is 120 ohms and the temperature coefficient " α " is about -1.5%/ °C. An error in temperature measurement of 1 °C will cause an indicated resistance error of 1.5%.

Thermistors can be destroyed during checking the voltage-current characteristic. When the voltage is sufficient to reach the peak on the curve (see Figure 2 on page 5), self-heating of the thermistor starts to lower the resistance which allows more current to flow and therefore more heating of the thermistor. The voltage cannot be manually reduced in time to prevent the thermistor from damage. If the thermistor is not completely destroyed, a change of resistance as described in Chapter 1.1 is probable. There are two basic rules which must be observed in this type of testing: 1.) Never place a thermistor directly across a voltage source without a limiting series resistor. 2.) Never exceed the maximum temperature even for a short period of time.

1.3 - ANEMOMETER CIRCUITS

The basic principle of the thermistor anemometer is that when a self-heated thermistor is exposed to a stream of moving air, a cooling action takes place which changes the resistance of the thermistor. The change in resistance is a function of the rate of flow.

The basic requirement then for an anemometer circuit is that it be able to accurately determine a resistance change in the sensing thermistor. This may be accomplished by the simple circuit of Figure 4a on page 11. By measuring the current (I) and the voltage across the thermistor (E_o), the resistance of the thermistor may be calculated and plotted against the known air velocity. This would be the calibration for the instrument for fixed air and thermistor properties. While not extremely accurate, this circuit would make a basic anemometer.

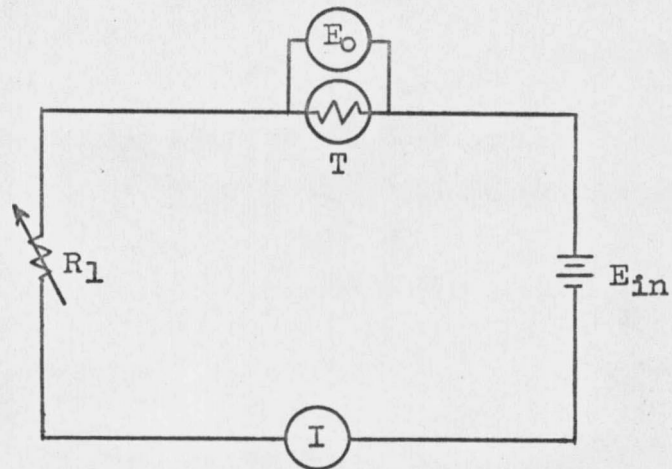
To improve accuracy, a Wheatstone Bridge is used for this research. A good Wheatstone Bridge is capable of determining resistance with an accuracy of 0.5%. See Figure 4b on page 11.

R_t is the resistance of the thermistor and R_4 is a variable resistor used to balance the bridge. The theory of a Wheatstone Bridge shows that:

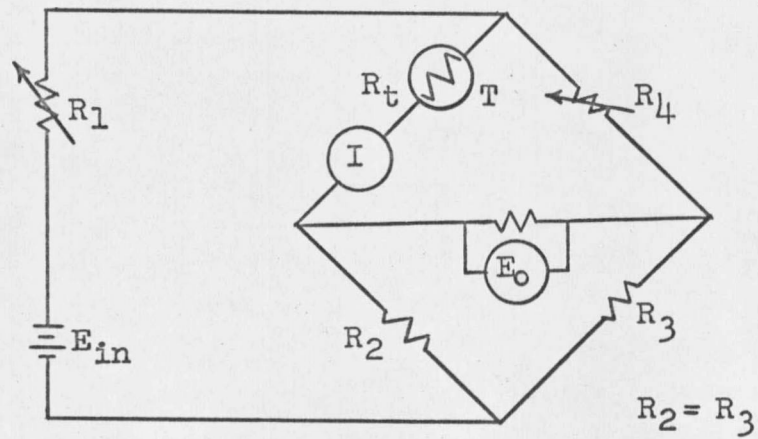
$$R_t = (R_2/R_3) \times R_4$$

when the bridge is balanced. If R_2 is made equal to R_3 , then R_t will equal R_4 when the bridge is at balance. The accuracy of the bridge is maximum when all four resistances are equal.

The signal from a thermistor is sufficient so that the bridge may be balanced without amplification. If amplification were required, the use of



(a)



(b)

FIGURE 4. RESISTANCE MEASUREMENT CIRCUITS

an alternating-current circuit would have an advantage since alternating current is easier to amplify. When the bridge is to be rebalanced for each observation, a d-c source is generally used because of the inherent simplicity of performance of d-c bridges.

There are two methods for operating the bridge for anemometer work. The bridge must first be balanced with the thermistor exposed to the desired air temperature and no air flow. When air is moved past the thermistor, convective cooling causes a resistance change and a consequent unbalance of the bridge which will be indicated by a voltage value, E_0 .

The first method of operating the bridge then is to let the bridge remain unbalanced and use the voltage E_0 as an indication of the air flow. The second method is to increase the voltage applied (E_{in}) until the bridge is balanced and used the current flow (I) as the air flow indication.

The first method has the disadvantage that sufficient air flow will cool the thermistor to the point where self-heating of the thermistor will not be sufficient to maintain operation on the right side of the voltage-current peak shown in Figure 2 on page 5. The thermistor is generally operated on the right hand side of the voltage peak because of the current required to heat the thermistor sufficiently to avoid the steepest portion of the resistance-temperature curve in Figure 1 on page 4. Maximum stability of the thermistor is obtained when it is operated in this manner.

The second method of operation requires the thermistor to be at a constant temperature for any air flow. By increasing the voltage across the bridge, current flow increases which heats the thermistor to offset the convective cooling. The bridge is balanced only by increasing this voltage.

If sufficient voltage is available, a wide range of air flow may be measured.

One caution which must be observed in the second method of operation is that the bridge voltage must be lowered before the air flow is reduced. If this is not done, the decrease in convective cooling will permit the thermistor to exceed its rated temperature with subsequent damage to the thermistor.

It would be desirable if the output of a thermistor anemometer were a linear function of the air velocity. However, this is not the case as may be seen in Figures 6, 7, and 8, Chapter 3.1. It is possible to construct an electronic circuit which would provide a linear output for flow velocity plotted against bridge output but it is both expensive and time consuming. This is again a decision for the designer of the device depending on the accuracy required in the range of air flows to be measured.

1.4 - TEMPERATURE EFFECTS

The effect of ambient temperature change on the calibration of the thermistor anemometer is considerably more complex than a pressure or humidity change. Ambient pressure and humidity changes are assumed to have no effect on the thermistor physical properties. This is not the case with a temperature change as the resistance of the thermistor is a function only of its temperature. See Chapter 1.1.

Since the velocity of the air flow is indicated as a bridge unbalance, any bridge unbalance due to an ambient temperature change would cause error in the calibration curve. Consider a thermistor anemometer in a system where the temperature may be regulated. The temperature of the system is changed but there is no air flow. The bridge will become unbalanced and indicate an air velocity when there is no velocity. Consider this as a static unbalance. Rebalance the bridge. Now let the temperature remain constant and let air flow which cools the heated thermistor. The bridge will become unbalanced and properly indicate an air flow. Consider this as a dynamic unbalance. The thermistor anemometer indicates air flow by dynamic unbalance and any static unbalance will cause error in the calibration.

Again, let there be no air flow. The static unbalance may be eliminated by placing an identical thermistor in the other leg of the bridge. The identical thermistor must be subjected to only static conditions. Assuming that this is possible, the bridge balance with no air flow would be independent of temperature. Now, let there be an air flow and consider only the thermistor exposed to that air flow. Different ambient temperatures will cause a different film temperature (see Chapter 2.3) and will cause

the thermistor operating resistance value and temperature to shift either left or right on its resistance-temperature characteristic curve (see Figure 1 on page 4). Both of these changes will affect the bridge output and therefore, the velocity determination. The magnitude of this effect would have to be determined by experimental research.

One definite method of contending with temperature change of the air would be to run a series of calibration curves over the range of temperature expected with about 2 °C increments of temperature change. The result would be a series of calibration curves for different air temperatures.

CHAPTER 2

SYSTEM DESIGN AND ANALYSIS

2.1 - TEST EQUIPMENT AND PURPOSE

The equipment used to construct and calibrate a thermistor anemometer was:

1. Air Blower
2. Reference Flow Meter
3. Thermistors
4. Wheatstone Bridge
5. Multimeters
6. Temperature Indicator
7. A Closed Air System

The air blower was used to provide air flow. The blower selected has a variable speed control so the rate of air flow could be regulated. See Figure 20 on page 52.

A Meriam Laminar Flow Meter was used for a reference flow meter. This unit is a complete primary flow measuring element calibrated within $\pm 0.5\%$ of the Meriam Instrument Company flow standards.

The thermistors selected for this research were Fenwal Electronics, Inc., glass coated beads. Three different thermistors were used with resistance values at 78 °F of 20,000 ohm $\pm 20\%$, 50,000 ohm $\pm 5\%$ and 100,000 ohm $\pm 5\%$. The part numbers for these thermistors are GB42P2, GA45P27, and GA51P68 respectively. These thermistors have temperature coefficients " α " of $-4.4\%/^{\circ}\text{C}$, $-4.6\%/^{\circ}\text{C}$ and $-4.6\%/^{\circ}\text{C}$ respectively at 78 °F.

Resistance values less than 20,000 ohm were not used because of the low resistance at the operating temperature. The 20,000 ohm thermistor has

a resistance of 38 ohms at an operating temperature of about 300 °C. If lower than 20,000 ohm thermistors had been used, the effect of lead resistance could have become significant at the operating temperature.

Higher resistance values were not used because of the power required to drive them over their voltage-current peak. See Figure 2 on page 5.

The thermistors were mounted in a copper tube with epoxy to form a probe as shown in Figure 16 on page 50. The probe assembly was placed in the test section so that the bead was at the center of the test section. The probe was held in place by a compression fitting which also formed a pressure seal. See Figure 21 on page 53.

The elements of the Wheatstone Bridge consisted of four decade resistors and direct-current power supplies. Three of the decade resistors were used as legs of the Wheatstone Bridge and one for voltage control across the bridge. The bridge was operated with the thermistor at a constant temperature and therefore constant resistance. The reasons for using the thermistor and bridge in this manner were discussed in Chapter 1.3.

The multimeters were used to measure voltage and current at the thermistor.

Temperature measurements were made with a thermocouple. The thermocouple probe was mounted immediately downstream from the thermistor probe as seen in Figure 21 on page 53. A type J iron-constantan thermocouple was used and the output was read on a Brown Potentiometer.

A closed system was constructed for this research by using a "U" shaped section of 4 inch diameter pipe and making pressure tight connections to the air blower. A portion of this pipe may be seen in Figure 20 on page 52.

The methods used to isolate the various parameters of a thermistor anemometer may be placed in the following categories:

1. Humidifying
2. Dehumidifying
3. Pressurizing
4. Temperature regulating

Humidifying the closed system was accomplished by introducing steam until the desired relative humidity was reached. The relative humidity was determined by a wet and dry bulb thermometer. The wet and dry bulb thermometer was mounted in the downstream portion of the test section. See Figure 21 on page 53.

Dehumidifying the closed system was accomplished with a mechanical dehydrating unit. The air output of this unit was connected to one side of the closed system and the air return was connected to the other. By circulating air through the drier in this manner, the closed system could be dried in 48 hours. See Figure 19 on page 51.

The dehydrating unit was also used to pressurize the closed system. When the return to the unit was opened to the atmosphere and the return from the closed system was blocked, a static pressure was built up in the system. The compressor in the dehydrating unit is capable of supplying 30 psig. The pressure in the closed system was limited to about 1 psig due to the type of construction of the heat exchanger and pipe. Pressure regulation was achieved by a column of water of sufficient length to sustain 1 psig. See Figure 19 on page 51. Pressure was read directly on a water manometer calibrated in psig.

Temperature regulation was achieved by a heat exchanger installed in the air stream. Cold water (45 °F) was circulated through the heat exchanger as required to maintain the air flow at the desired temperature. Temperature regulation by this method was slow and difficult since the water flow was controlled manually. However, with patience, good temperature control was possible. See Figure 21 on page 53.

2.2 - FLOW CHARACTERISTICS

The only flow characteristic of concern for this research is the flow regime (i.e., laminar, transition, or turbulent). The flow regime is determined from Reynolds number:

$$R_e = \frac{V D \rho}{\mu}$$

where V = mean velocity of the fluid in the pipe in feet per second,
 D = diameter of the pipe in feet,
 ρ = mass density of the fluid in the pipe in lb-sec²/ft⁴,
 μ = viscosity of the fluid in the pipe in lb-sec/ft².

For the conditions of this experiment:

$V = 4$ ft/sec (this is the lowest air flow reading taken for calibration),

$D = 0.324$ ft,

$\rho = 1.94 \times 10^{-3}$ lb-sec²/ft⁴ (at 25 inches Hg and 70 °F),

and $\mu = 3.8 \times 10^{-7}$ lb-sec/ft² (at 70 °F).

Then:

$$R_e = \frac{4 \times 0.324 \times 1.94 \times 10^{-3}}{3.8 \times 10^{-7}}$$

and $R_e = 6620$.

This is the lowest value used in the performance of this test.

Since the flow regimes are defined as laminar for $R_e < 2100$, transition for $2100 \leq R_e \leq 4000$ and turbulent for $R_e > 4000$, it is concluded that all the work done in this research is in the turbulent flow regime.

2.3 - HEAT TRANSFER CHARACTERISTICS

¹Consider the three principle modes of heat transfer; conduction, radiation, and convection. Conduction is a process by which heat flows from a region of higher temperature to a region of lower temperature within a medium or between different mediums in direct physical contact. Radiation is a process by which heat flows from a high temperature body to a body at a lower temperature when the bodies are separated in space, even when a vacuum exists between them. Convection is a process of energy transport by the combined action of heat conduction, energy storage, and mixing motion. Convection is most important as the mechanism of energy transfer between a solid surface and a gas.

A complete heat transfer analysis then will include all three modes of heat transfer. However, as stated above, convection includes the effects of conduction so only radiation and convection need be considered.

The unit surface conductance for radiant heat transfer " $\bar{h}_r$ " may be defined as:

$$\bar{h}_r = \frac{\sigma \epsilon (T_s^4 - T_\infty^4)}{(T_s - T_\infty)}$$

where σ = Stefan - Boltzman constant

$$(0.1714 \times 10^{-8} \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R}^4),$$

ϵ = emissivity (0.90 for glass),

T_s = 781 $^\circ\text{R}$ - Absolute temperature in $^\circ\text{R}$ of the surface from

which the radiation takes place. (Since the thermistor is

operated at 572 $^\circ\text{F}$ and the air temperature is 70 $^\circ\text{F}$, the sur-

1. "Principles of Heat Transfer", by Frank Kreith, 4th Ed. International Textbook Company, Scranton, Pennsylvania

face and film temperature are assumed to be 321°F.),

$T_{\infty} = 530^{\circ}\text{R}$ - absolute temperature of the pipe walls.

Then:

$$\bar{h}_r = 0.1714 \times 10^{-8} \times 0.90 \frac{(781^4 - 530^4)}{(781 - 530)}$$

$$\bar{h}_r = 1.795 \text{ Btu/hr ft}^2 \text{ }^{\circ}\text{R.}$$

The convection heat transfer in this case is actually forced convection over an exterior surface. While the heat transferred by radiation is a function only of temperature difference, the heat transfer by forced convection depends on the behavior of the fluid boundary layer around the object exposed to the air stream.

Consider a sphere exposed to an air stream. For an ideal fluid, particles striking the most forward point on the sphere are stopped and the pressure increases one velocity head. This point on the sphere is called the stagnation point, the flow must divide and flow around the sphere. The fluid must accelerate when it flows around the sphere and reaches a maximum velocity on both sides, then falls again to zero at the stagnation point in the rear. This is an idealized flow pattern and is not approached by a real fluid except for very low Reynolds number.

At other than very low Reynolds numbers, skin-friction drag causes an adverse pressure gradient with a resultant separation of the flow from the sphere. For flow of real fluid, the particles at rest at the forward stagnation point accelerate to maximum velocity on the sides of the sphere,

the particles separate from the surface of the sphere and a wake is formed at the downstream side. See Figure 5 on page 24.

For the real fluid, boundary layer growth will begin at the stagnation point and energy will be dissipated in overcoming resistance of the adverse pressure gradient. The angle " θ " as shown in Figure 5 is dependent on the Reynolds number, decreasing with increasing Reynolds number.

It is concluded that the heat transferred by forced convection is some function of the Reynolds number due to the change of surface of the sphere in direct physical contact with the moving fluid. In the Reynolds number range from about 25. to 100,000, the equation recommended by McAdams (ref. 4) for calculating the average unit surface conductance " $\bar{h}_c$ " for spheres heated or cooled by a gas is:

$$\frac{\bar{h}_c D_o}{K_f} = 0.37 \left(\frac{V \rho D_o}{\mu_f} \right)^{0.6}$$

where D_o = sphere diameter in feet,

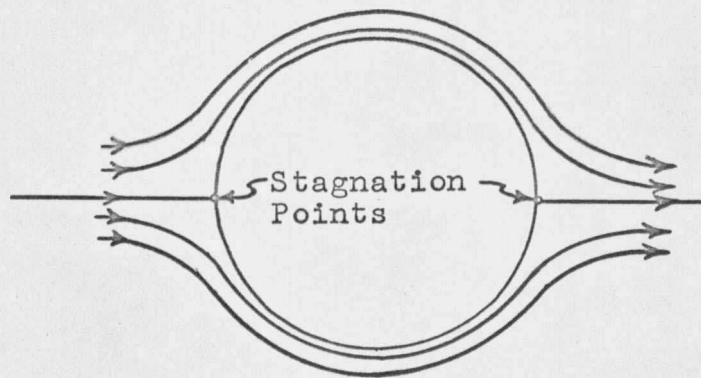
K_f = thermal conductivity of air at mean film temperature in
Btu/hr ft $^{\circ}$ R,

V = velocity of free stream in ft/sec,

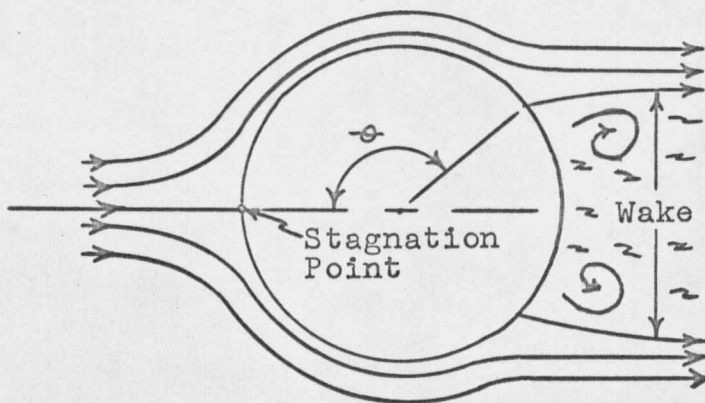
ρ = mass density of the free stream in lb - sec²/ft⁴,

μ_f = viscosity of air at mean film temperature in lb - sec/ft².

An example calculation of the average unit surface conductance " $\bar{h}_c$ " is of value for comparison of its magnitude to the magnitude of the unit surface conductance for radiant heat transfer " $\bar{h}_r$ ". Since $\bar{h}_c$ is a function of velocity and velocity is a variable in this research, the lowest velocity used for calibration (4.ft/sec) shall be used in this calculation. A mean



IDEAL FLUID



REAL FLUID

FIGURE 5. FLOW OF IDEAL AND REAL FLUID ABOUT A SPHERE

film temperature of 321 °F is assumed.

Then: $D_o = 0.00833$ ft,

$K_f = 0.0197$ Btu/hr ft °R (at mean film temperature of 321 °F),

$V = 4$ ft/sec,

$\rho = 1.94 \times 10^{-3}$ lb-sec²/ft⁴ (at 25 inches.Hg and 70 °F),

$\mu_f = 5.1 \times 10^{-7}$ lb-sec/ft² (at mean film temperature of 321 °F),

and

$$\bar{h}_c = \frac{0.0197}{0.00833} \times 0.37 \left(\frac{4 \times 1.94 \times 10^{-3} \times 0.00833}{5.1 \times 10^{-7}} \right)^{0.6}$$

$$\bar{h}_c = 15.95 \text{ Btu/hr ft}^2 \text{ °R.}$$

Since $\bar{h}_c$ is calculated from the lowest velocity used in this research and still is greater than $\bar{h}_r$ by a factor of 8.9, it may be concluded that the heat transferred by radiation will always be quite small compared to the heat transferred by forced convection. However, the most accurate results will be obtained by defining a total unit surface conductance " $\bar{h}_t$ " where:

$$\bar{h}_t = \bar{h}_r + \bar{h}_c .$$

CHAPTER 3

TESTS AND RESULTS

3.1 - TEST METHODS AND CALIBRATION

This research was conducted by isolating the parameters and investigating their effects separately. The first portion of the research was devoted to background investigation and construction of the equipment.

The anemometer was assembled after selection of the circuitry and thermistors as described in previous chapters. The Meriam Laminar Flow Meter was used as the primary velocity reference. The closed system was constructed so that the parameters could be controlled.

Prior to initial calibration of the anemometer, the test conditions for the entire research were fixed. The air stream temperature was held within ± 1 °F at 70 °F for all measurements. The thermistor temperature was held constant at 572 °F as determined from the resistance. For initial calibration, air at atmospheric pressure of 25 inches mercury (4900 feet altitude at laboratory site) and 0-10% relative humidity was used. The air was dried to 0-10% relative humidity by the dehydrating unit previously discussed.

Calibration curves for the following thermistors were obtained:

<u>Thermistor</u>	<u>Figure</u>	<u>Page</u>
GB42P2	6	27
GA45P27	7	28
GA51P68	8	29

These calibration runs were repeatable within $\pm 2\%$ current measurement and ± 1 °F temperature control. The only problem encountered with repeating calibration runs was when a thermistor was used which had been accidentally subjected to excessive temperature with consequent damage to the thermistor.

The effects of ambient pressure were investigated by pressurizing the

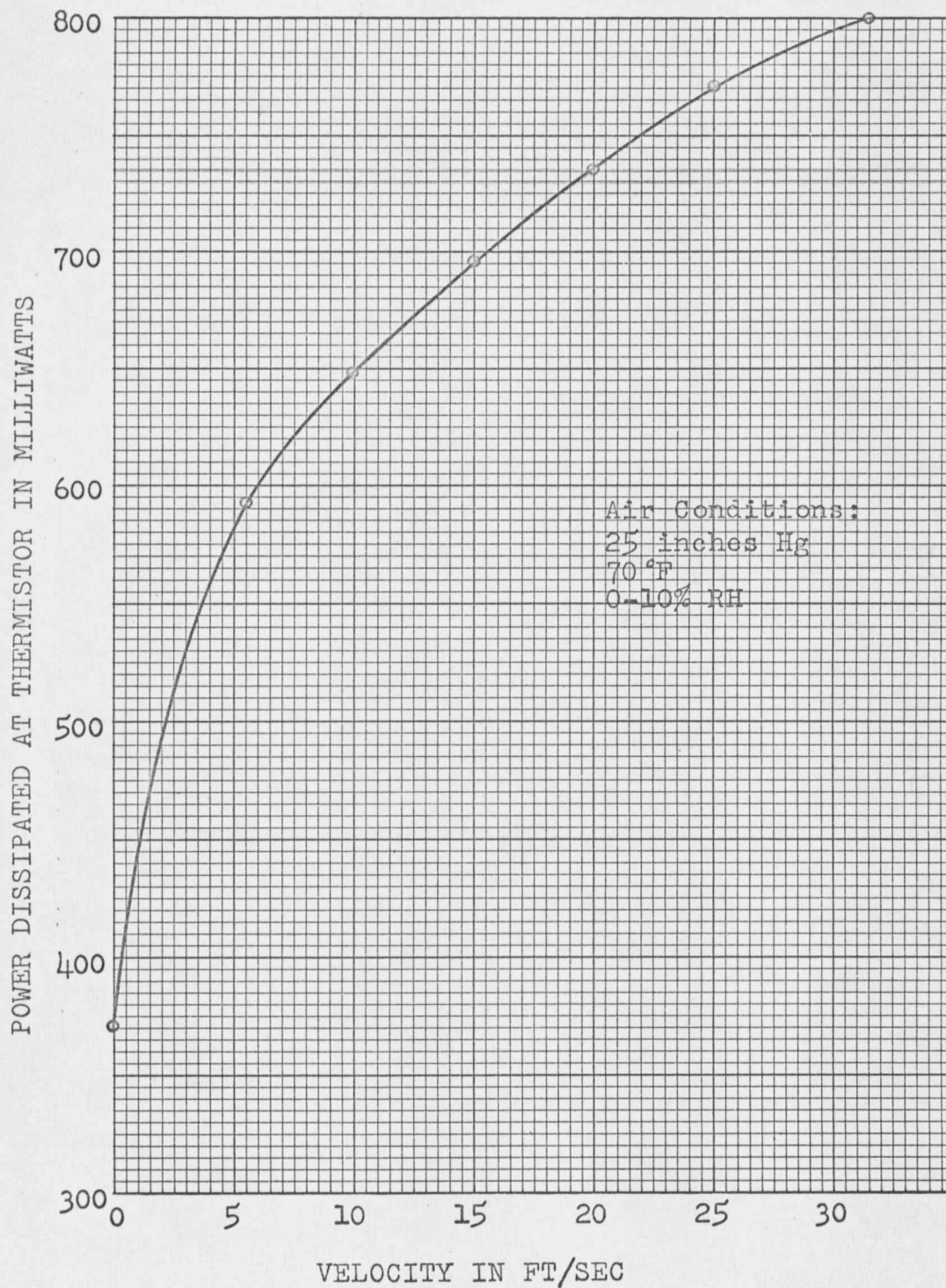


FIGURE 6. CALIBRATION CURVE FOR GB42P2 THERMISTOR

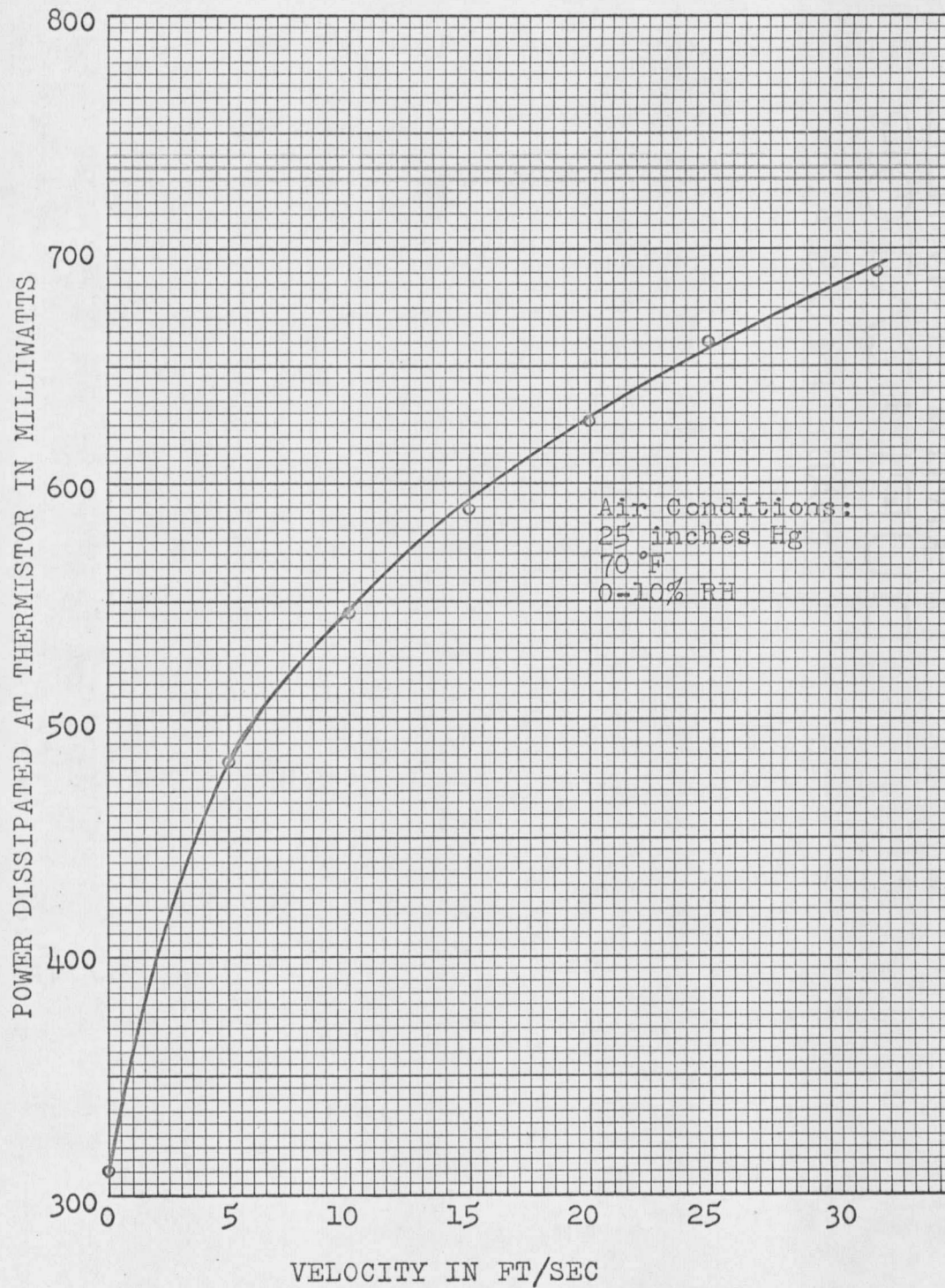


FIGURE 7. CALIBRATION CURVE FOR GA45P27 THERMISTOR

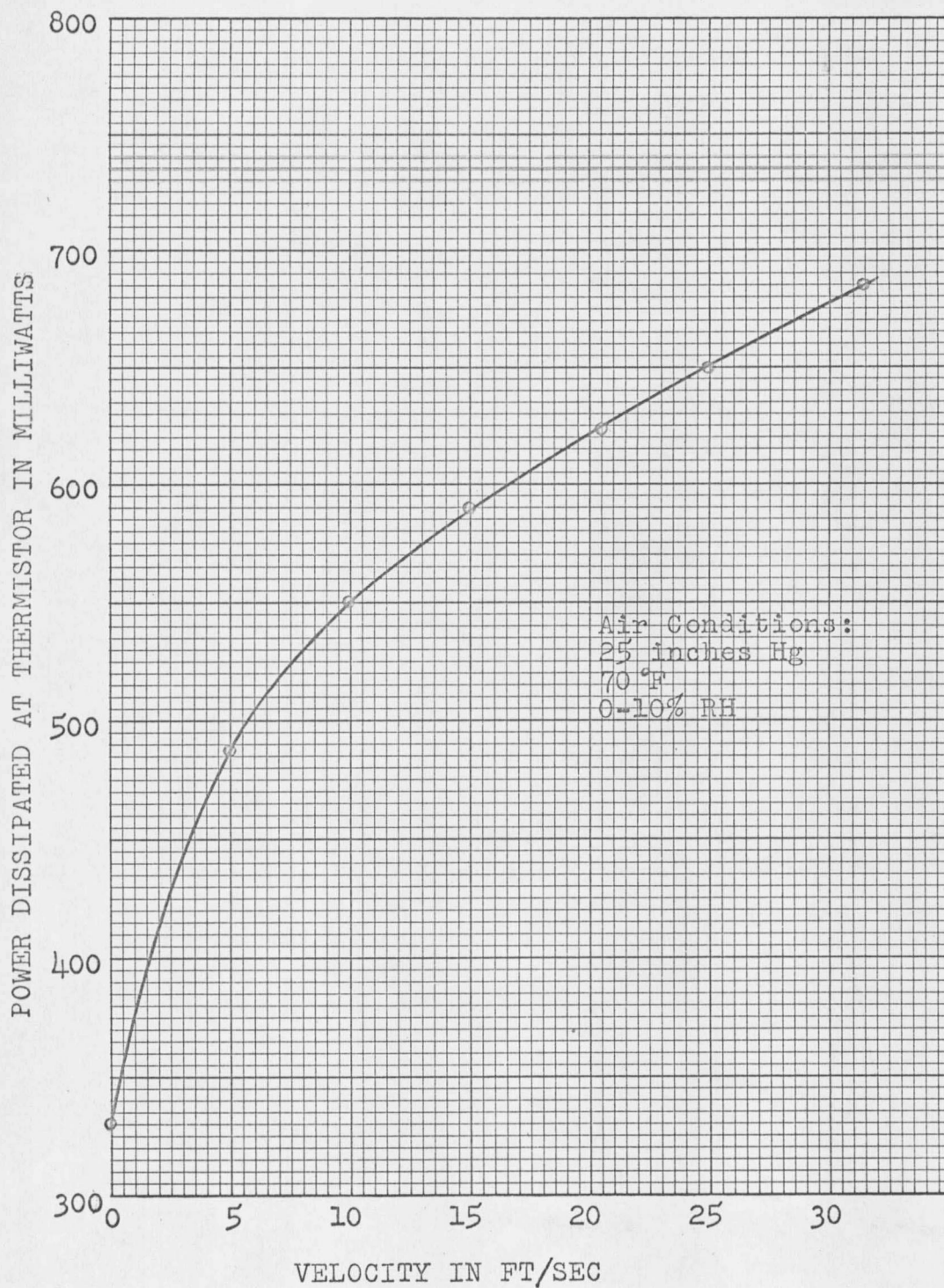


FIGURE 8. CALIBRATION CURVE FOR GA51P68 THERMISTOR

closed system to 1 psig. Temperature and humidity of the system were held constant. See Chapter 3.2.

The effects of ambient humidity were investigated by alternately drying and saturating the closed system while temperature and pressure were held constant. See Chapter 3.3.

The following has no bearing on the research conducted but is of technical interest. Considerable time was lost during the research on the humidity effect due to the change in resistance caused by overheating one of the thermistors. This change was first taken as the effect of humidity until the lack of consistency in the results was noted. At that time, measurements were made on the thermistor to determine its resistance. This was a GA51P68 thermistor which originally had a resistance at 78 °F of 100,000 ohm $\pm 5\%$. At the time it was removed from the humidity test, its resistance was about 300,000 ohm at 78 °F. This thermistor was operated once more at the same voltage to see if it would change resistance again. After this run, the resistance at 78 °F was found to be 345,000 ohm. This deterioration with time is shown in Figure 9. The thermistor was used on the dates given but was not used during the interval between dates.

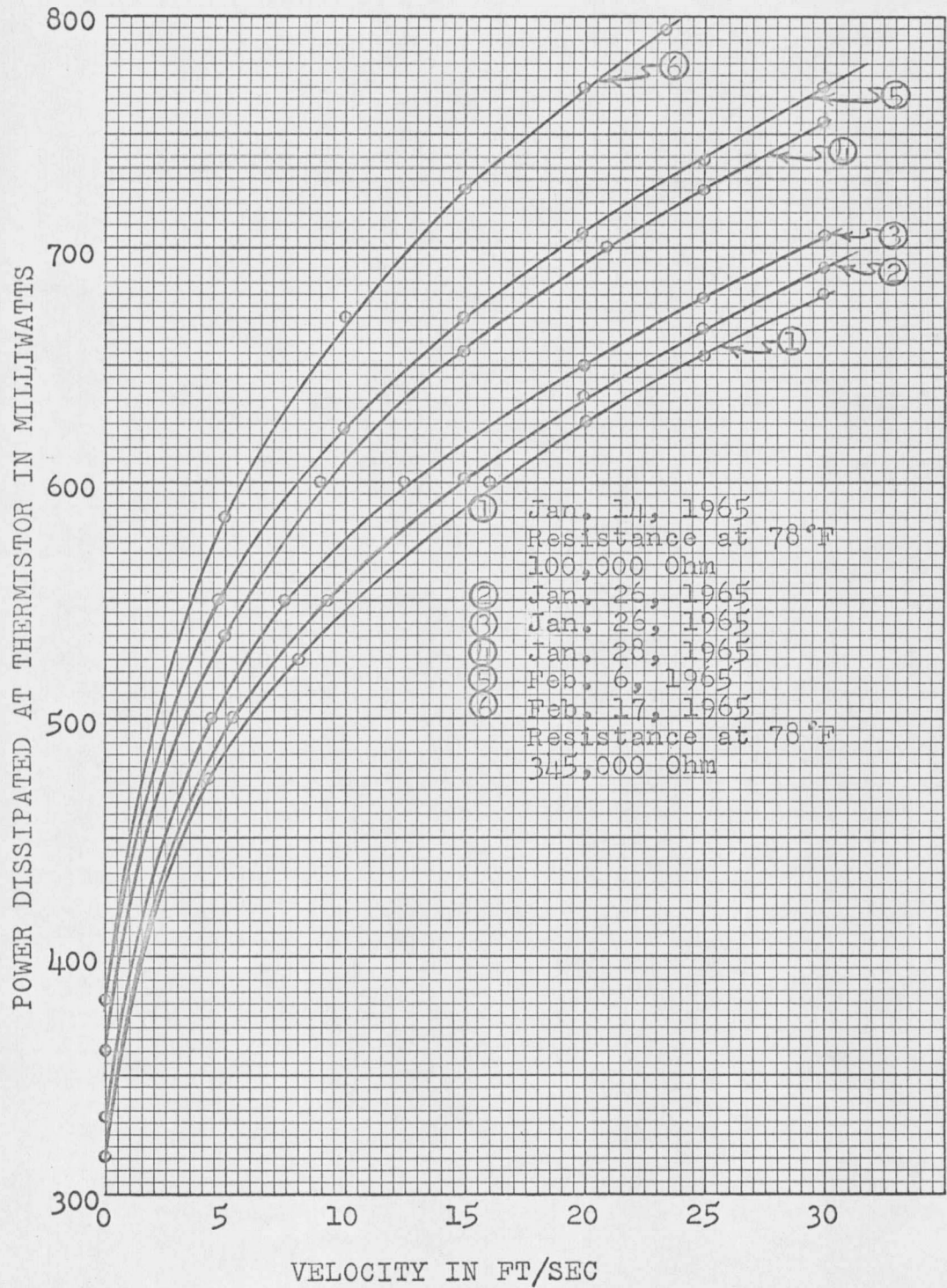


FIGURE 9. DETERIORATION BY OVERHEATING GA51P68
THERMISTOR

3.2 - PRESSURE EFFECTS

It was assumed for this portion of the research that small pressure variations have no effect on the physical properties of the thermistor.

The effect of changes in the air pressure was determined by comparing the calibration curve for a thermistor in a closed system at atmospheric pressure and the calibration curve for the same thermistor in the same closed system at an elevated pressure. For these measurements, all parameters were held constant except pressure.

The pressure effect was investigated using the three thermistors described in Chapter 2.1. Figures 10, 11, and 12 show the two pressure-calibration curves for each thermistor.

It is readily seen from the curves that the pressure effect for a 1 psig pressure change is fairly small but consistent for all three thermistors. The average increase of power dissipated at elevated pressure for all three thermistors lies in the range from 1.5 to 2.0%.

An examination of the heat transfer equations is of value to see if these results are consistent with the change in the total unit surface conductance " $\bar{h}_t$ " resulting from an increase in pressure.

The total unit surface conductance is a summation of the average unit surface conductance for forced convection " $\bar{h}_c$ " and the unit surface conductance for radiation " $\bar{h}_r$ ". As explained in Chapter 2.3, $\bar{h}_r$ is a function only of temperature differences and does not change with pressure. $\bar{h}_c$ is a function of the Reynolds number which does change with pressure:

$$\bar{h}_c = 0.37 \frac{K_f}{D_o} \left(\frac{V \rho D_o}{\mu_f} \right)^{0.6}$$

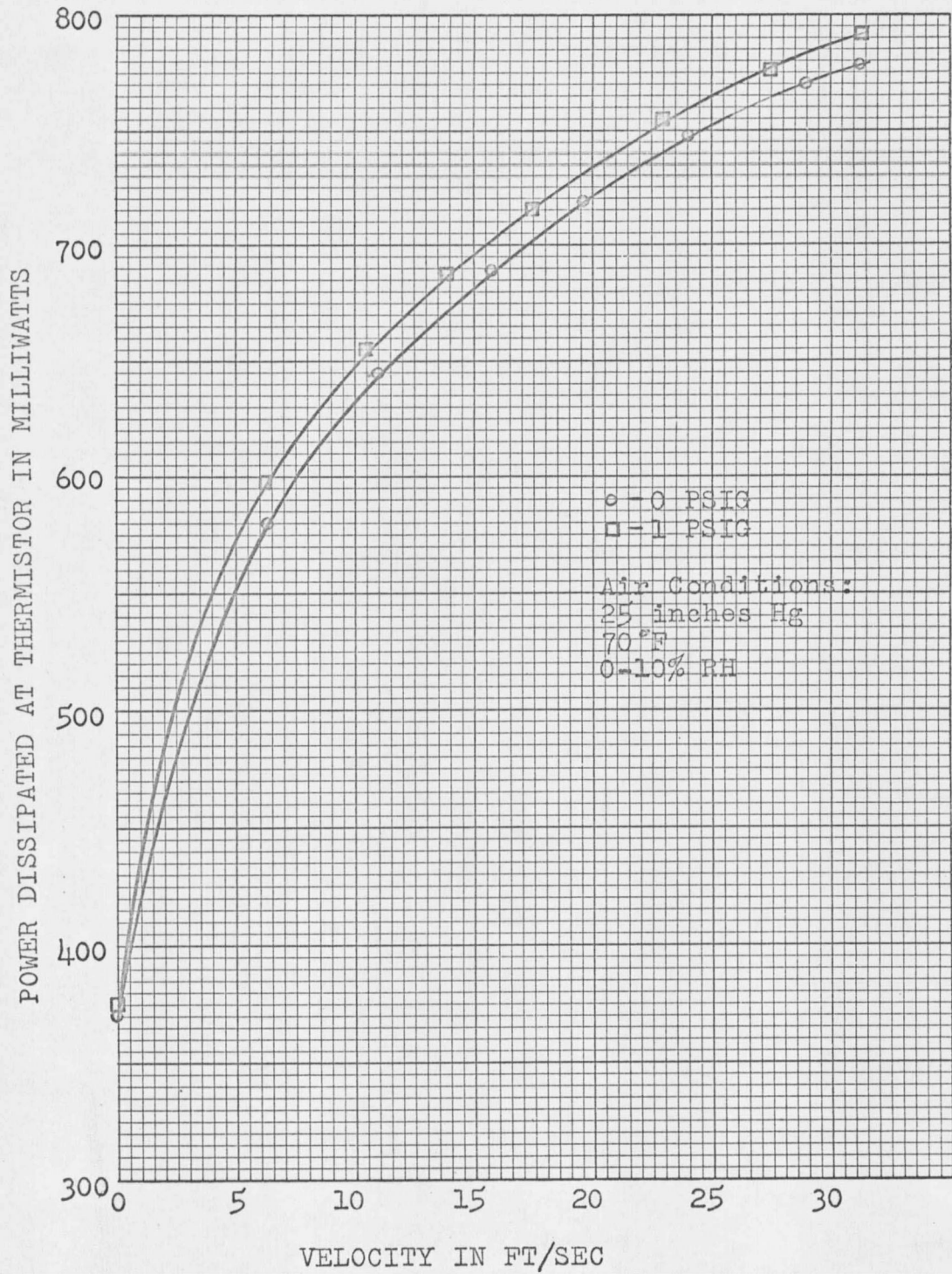


FIGURE 10. PRESSURE - CALIBRATION CURVE FOR GB42P2 THERMISTOR

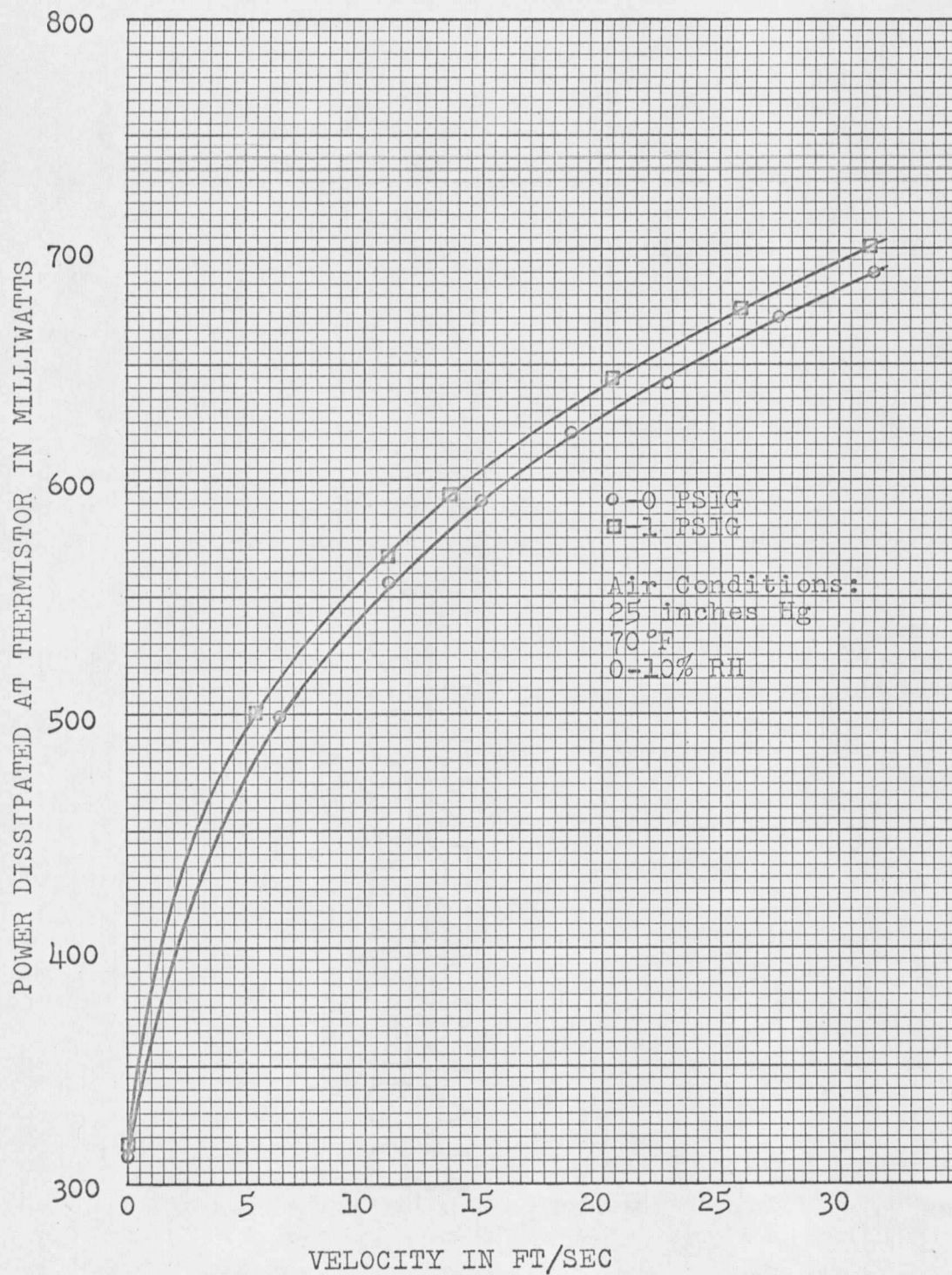


FIGURE 11. PRESSURE - CALIBRATION CURVE FOR GA45P27 THERMISTOR

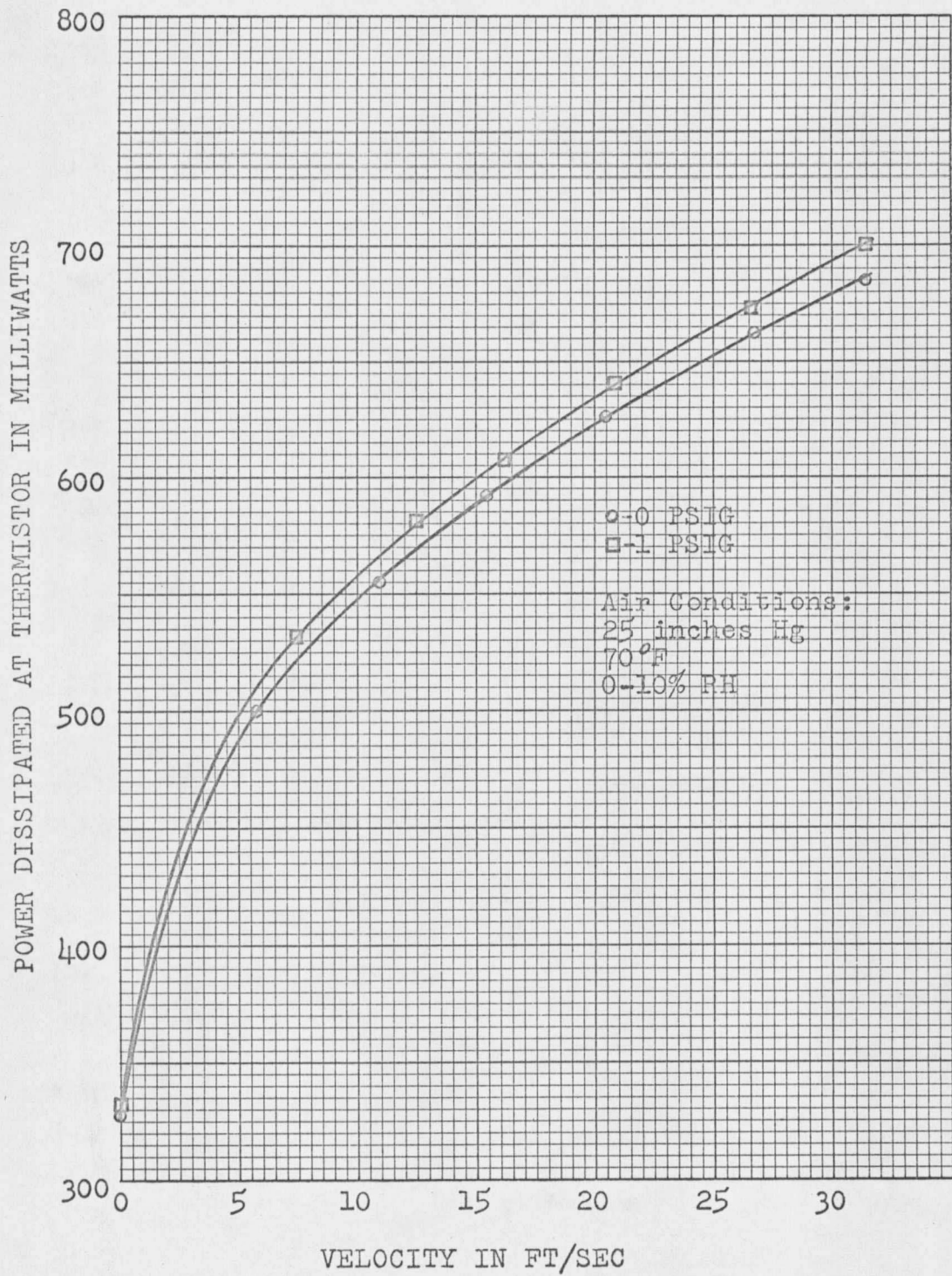


FIGURE 12. PRESSURE - CALIBRATION CURVE FOR GA51P68 THERMISTOR.

where D_o = sphere diameter in ft,
 K_f = thermal conductivity of air at a mean film temperature of 321 °F in Btu/hr ft °R,
 V = velocity of free stream in ft/sec,
 ρ = mass density of free stream in lb-sec²/ft⁴,
 μ_f = viscosity of air at a mean film temperature of 321 °F in lb-sec/ft².

For purposes of calculation, the assumed mean film temperature of 321°F shall be assumed constant over the range 0 to 1 psig. The film temperature will vary with heat flow but for the small changes in heat flow expected here, the variation in temperature will be assumed negligible. Another reason for this assumption is that the calculations are made for a constant velocity.

Consider the effect of a pressure change on each of the above terms. The free stream velocity and sphere diameter do not change with pressure. Thermal conductivity and viscosity are functions of the film temperature assumed constant over the range 0 to 1 psig and do not vary significantly with pressure for moderate ranges (i.e., for air this is true up to 100 atmospheres). Mass density is a function of pressure and the magnitude of its change is calculated below.

$$\rho = \frac{P}{g R T}$$

where P = absolute pressure in lb/ft²,
 g = 32.2 ft/sec² - gravity,
 R = 53.3 ft-lb/lb °R - ideal gas constant,

T = absolute temperature in $^{\circ}\text{R}$.

The mass density at 0 psig is calculated using:

$P = 23$ inches Hg absolute,

$T = 530$ $^{\circ}\text{R}$,

and $\rho = 0.00194$ lb-sec²/ft⁴.

The mass density at 1 psig is calculated using:

$P = 25$ inches Hg absolute + 1 psig,

$T = 530$ $^{\circ}\text{R}$,

and $\rho = 0.00210$ lb-sec²/ft⁴.

To calculate a typical value of $\bar{h}_c$ for comparing the effects of pressure change, a velocity value must be selected. A velocity value of 20 ft/sec will be used.

Then:

$$\bar{h}_c = 0.37 \frac{K_f}{D_o} \left(\frac{V \rho D_o}{\mu_f} \right)^{0.6}$$

where $K_f = 0.01970$ Btu/hr ft $^{\circ}\text{R}$,

$D_o = 0.00833$ ft,

$V = 20$ ft/sec,

$\mu_f = 5.1 \times 10^{-7}$ lb-sec/ft²

$\rho = 0.00194$ lb-sec²/ft⁴ for 0 psig,

and $\rho = 0.00210$ lb-sec²/ft⁴ for 1 psig.

Then:

$\bar{h}_c = 42.0$ Btu/hr ft² $^{\circ}\text{R}$ - for 0 psig and dry air,

and $\bar{h}_c = 44.2$ Btu/hr ft² $^{\circ}\text{R}$ - for 1 psig and dry air.

From Chapter 2.3;

$$\bar{h}_r = 1.795 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R.}$$

Then:

$$\bar{h}_t = \bar{h}_c + \bar{h}_r,$$

$$\bar{h}_t = 43.795 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{at 0 psig and dry air,}$$

and $\bar{h}_t = 45.995 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{at 1 psig and dry air.}$

The expected percentage increase of power dissipated at the thermistor for a pressure change of 1 psig at a velocity of 20 ft/sec would, by the above calculations, be 4.8%. The actual percentage increase at 20 ft/sec is about 1.5%. Although the results do not coincide in magnitude, the effects are both in the same direction; a pressure increase causes greater heat transfer. The fact that greater heat is transferred as a result of a pressure increase contradicts the assumption that the mean film temperature is constant over the range 0 to 1 psig.

The pressure - calibration curve for the GA51P68 thermistor (Figure 12) indicates that at a velocity of 20 ft/sec, the heat transferred from the thermistor or power dissipated is 623 milliwatts for 0 psig and 633 milliwatts for 1 psig. Now consider that the thermistor is enclosed in glass. This heat must pass through the glass and follows the equation for heat transfer by conduction:

$$Q = K (t_1 - t_2)$$

where Q = heat transferred,

K = a constant for a given material and configuration,

t_1 = temperature of inside surface of glass (i.e., thermistor

temperature),

t_2 = temperature of outside surface of glass (i.e., film temperature).

The inside surface of the glass is at the thermistor temperature of 572 °F which is constant. To evaluate the constant "K", the assumed film temperature of 321 °F and the heat transferred value of 623 milliwatts are used.

$$K = 2.48 \text{ milliwatts/ } ^\circ\text{F.}$$

Then to calculate a new film temperature, the inside surface temperature of 572 °F and the heat transferred value of 633 milliwatts are used. The new approximation for the film temperature is 317 °F.

This difference in film temperature, however, is not large enough to change the heat transfer coefficient " $\bar{h}_c$ " for engineering calculations.

It is therefore concluded that the difference in magnitude for the percentage change in heat transferred due to the pressure increase is attributed to a lack of correlation between theoretical equations and experimental results.

3.3 - HUMIDITY EFFECTS

It was assumed for this portion of the research that the moisture in the air has no effect on the physical properties of the thermistors since the thermistors are sealed in glass.

The effect of changes in the moisture content was determined by comparing the calibration curves for a thermistor in a closed system with dry air and for the same thermistor in the same closed system with saturated air. For these measurements, all parameters were held constant except relative humidity.

The air in the system was dried with the dehydrating unit previously described. Humidity measurements made with a wet and dry bulb thermometer indicated relative humidities below 10% for the dry system. Humidifying was accomplished by introducing steam into the system until saturation was obtained.

The humidity effect was investigated using the three thermistors described in Chapter 2.1. Figures 13, 14, and 15 show the two humidity-calibration curves for each thermistor.

It is readily seen from the curves in Figure 13, 14, and 15 that the humidity effect is not within the sensitivity of the test equipment. It may be concluded that while humidity does affect the heat transferred by forced convection, as shown in the following analysis, the effect is negligible for ordinary engineering calculations.

Consider the effect of a moisture content change on the value for heat transferred by forced convection. The average unit surface conductance for forced convection, " $\bar{h}_c$ " is a function of Reynolds number and the thermal con-

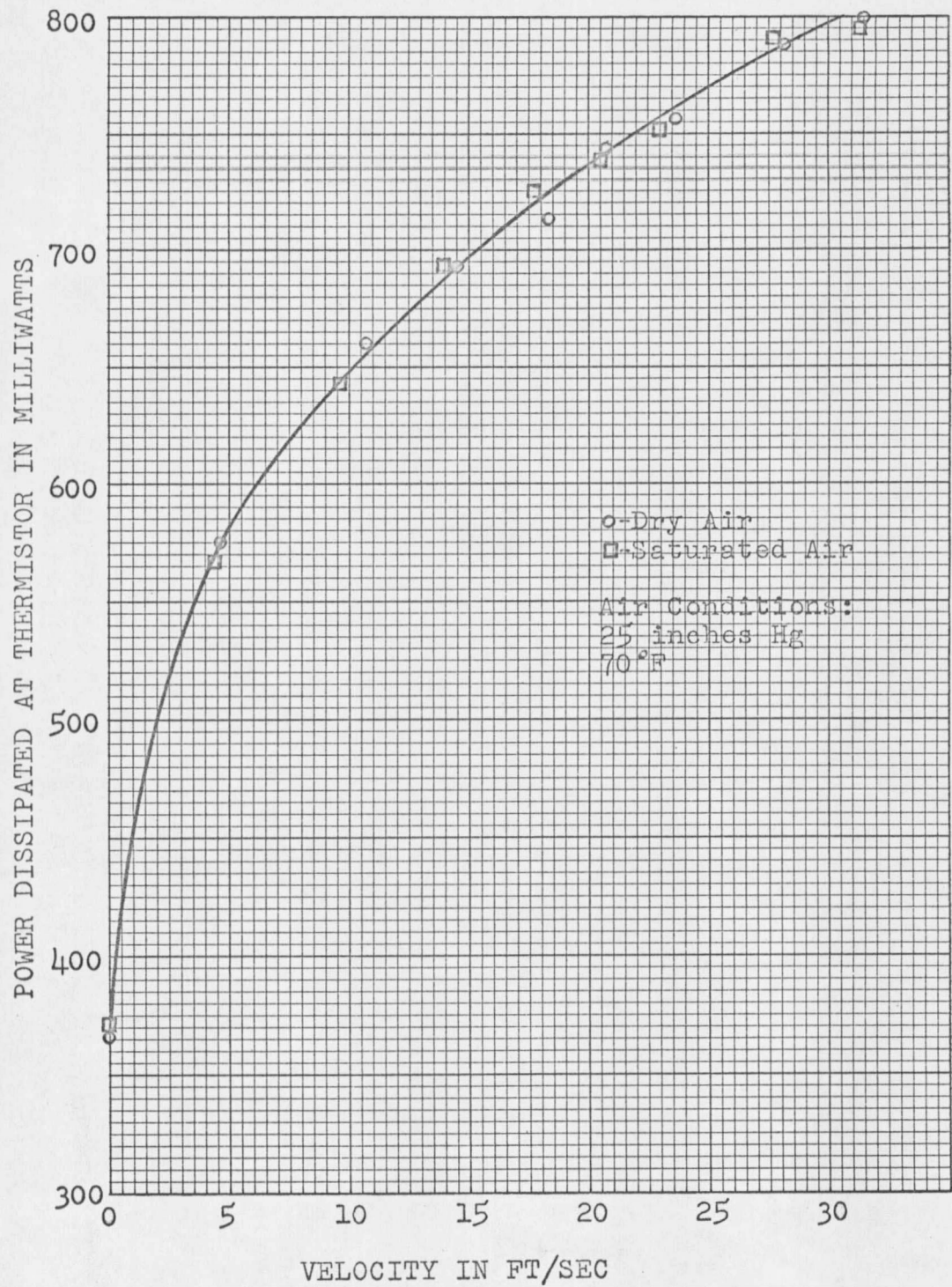


FIGURE 13. HUMIDITY - CALIBRATION CURVE FOR GB42P2
THERMISTOR

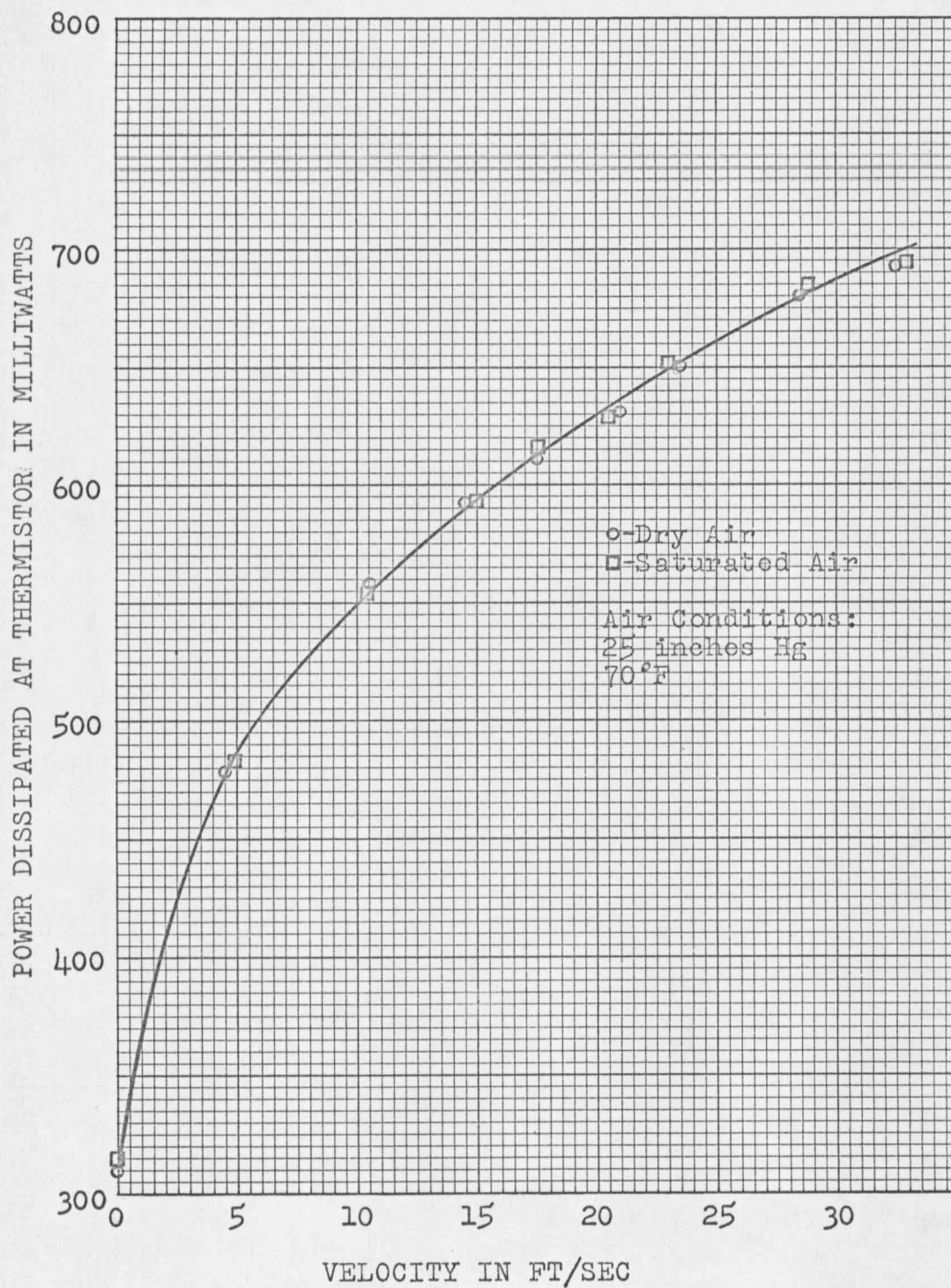


FIGURE 14. HUMIDITY - CALIBRATION CURVE FOR GA45P27
THERMISTOR

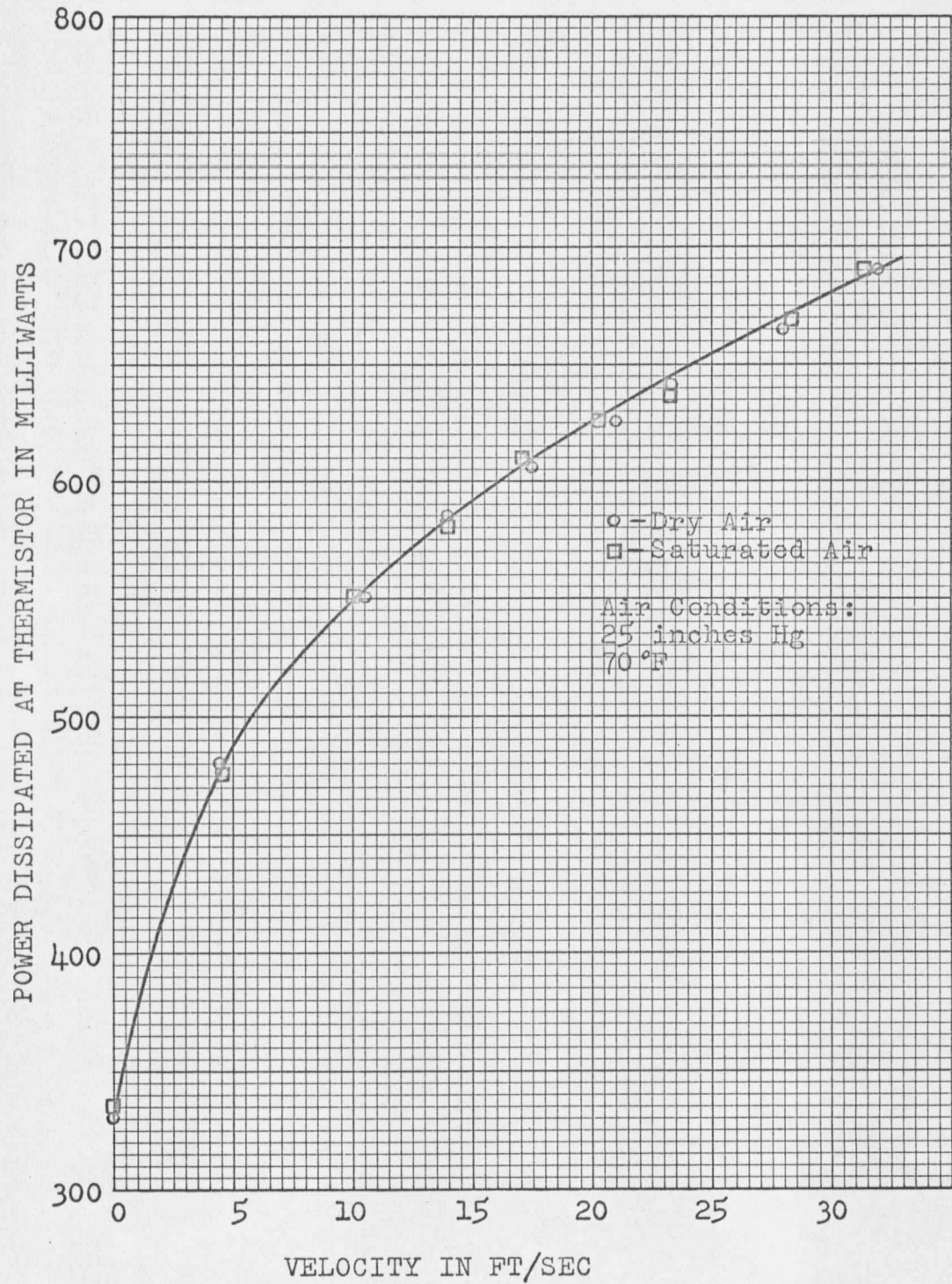


FIGURE 15. HUMIDITY - CALIBRATION CURVE FOR GA51P68
THERMISTOR

ductivity, both which change with humidity.

$$\bar{h}_c = 0.37 \frac{K_f}{D_o} \left(\frac{V \rho D_o}{\mu_f} \right)^{0.6}$$

where D_o = sphere diameter in ft,

K_f = thermal conductivity of air at a mean film temperature of 321 °F in Btu/hr ft °R,

V = velocity of free stream in ft/sec,

ρ = mass density of free stream in lb-sec²/ft⁴,

μ_f = viscosity of air at a mean film temperature of 321 °F in lb-sec/ft².

For purposes of calculation, the assumed mean film temperature of 321°F shall be assumed constant over the range 0 to 100% relative humidity.

Consider the effect of a humidity change on each of these terms. The free stream velocity and sphere diameter do not change with humidity. Mass density, viscosity and thermal conductivity are not direct functions of humidity but they do exhibit a small change with humidity.

The effect of humidity on mass density shall be considered first.

$$\rho = \frac{P_a}{g R_a T} + \frac{RH}{g v_{vs}}$$

where P_a = absolute pressure of dry air in lb/ft²,

g = 32.2 ft/sec² - gravity,

R_a = 53.3 ft-lb/lb °R - ideal gas constant,

T = absolute temperature in °R,

RH = relative humidity,

v_{vs} = specific volume of saturated water vapor at temperature T in

ft³/lb.

The mass density for dry air is calculated using:

$$P_a = 12.25 \text{ psia},$$

$$T = 530 \text{ } ^\circ\text{R},$$

and $\rho = 0.00194 \text{ lb-sec}^2/\text{ft}^4.$

The mass density for saturated air is calculated using:

$$P_a = 11.89 \text{ psia},$$

$$T = 530 \text{ } ^\circ\text{R}$$

and $\rho = 0.00192 \text{ lb-sec}^2/\text{ft}^4.$

Next consider the effect of a humidity change on the viscosity. Assuming air and water vapor to constitute a gas mixture:

$$\mu_{\text{mix}} = \sum_{i=1}^n \frac{X_i \mu_i}{\sum_{j=1}^n X_j \Phi_{ij}}$$

and
$$\Phi_{ij} = \frac{1}{\sqrt{8}} \left(1 + \frac{M_i}{M_j} \right)^{-\frac{1}{2}} \left[1 + \left(\frac{\mu_i}{\mu_j} \right)^{\frac{1}{2}} \left(\frac{M_j}{M_i} \right)^{\frac{1}{4}} \right]^2$$

where n = number of species in the mixture,

X_i & X_j = mole fractions of species i and j ,

μ_i & μ_j = viscosities of species i and j at the system conditions,

M_i & M_j = molecular weights of the species.

This is a very long and detailed calculation and only the results will be presented here. The equations and methods of calculation are fully ex-

plained in "Transport Phenomena" by Bird, Stewart, and Lightfoot, published by John Wiley and Sons, Inc.

For dry air at a mean film temperature of 321 °F,

$$\mu_f = 5.1 \times 10^{-7} \text{ lb-sec/ft}^2.$$

For saturated air at a mean film temperature of 321 F,

$$\mu_f = 5.05 \times 10^{-7} \text{ lb- sec/ft}^2.$$

Finally, consider the effect of a humidity change on the thermal conductivity.

$$K_f = b \mu_a C_v$$

where μ_a = absolute viscosity in lb/hr ft,

C_v = specific heat at constant volume in Btu/lb R,

b = a constant that depends upon the number of atoms in the gas molecule (1.90 for air).

For dry air at a mean film temperature of 321 °F,

$$K_f = 0.0197 \text{ Btu/hr ft } ^\circ\text{R}.$$

For saturated air at a mean film temperature of 321 °F,

$$K_f = 0.0199 \text{ Btu/hr ft } ^\circ\text{R}.$$

The small change in the thermal conductivity is due to the fact that the viscosity change and the specific heat at constant volume change tend to cancel each other.

Then:

$$\bar{h}_c = 0.37 \frac{K_f}{D_o} \left(\frac{V \rho D_o}{\mu_f} \right)^{0.6}$$

and for saturated air:

$$D_o = 0.00833 \text{ ft},$$

$$K_f = 0.0199 \text{ Btu/hr ft } ^\circ\text{R},$$

$$V = 20 \text{ ft/sec},$$

$$\mu_f = 5.07 \times 10^{-7} \text{ lb-sec/ft}^2,$$

$$\rho = 0.00192 \text{ lb-sec}^2/\text{ft}^4.$$

Then:

$$\bar{h}_c = 42.3 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{ for 0 psig and saturated air.}$$

From Chapter 9:

$$\bar{h}_c = 42.0 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{ for 0 psig and dry air.}$$

The unit surface conductance for radiation " $\bar{h}_r$ " is unchanged for either pressure of humidity.

$$\bar{h}_r = 1.795 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R}.$$

Consider the total unit surface conductance " $\bar{h}_t$ " where:

$$\bar{h}_t = \bar{h}_c + \bar{h}_r,$$

$$\bar{h}_t = 43.795 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{ for 0 psig and dry air,}$$

and $\bar{h}_t = 44.095 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{R} - \text{ for 0 psig and saturated air.}$

This result verifies that the change in heat transferred caused by moisture content variation is negligible. The calculated percentage change is only 0.68%.

CHAPTER 4

EXPLANATION OF PHOTOGRAPHS

The photographs of the whole test arrangement are collected after this chapter for quick reference. See Chapter 2.1 for the descriptions of the equipment shown in these photographs.

Figure 16 is a closeup of a typical thermistor probe. All of the GB42P2, GA45P27, and GA51P68 thermistors were mounted in identical probes. The tip of the glass enclosed thermistor may be seen protruding from the left end of the copper tube. The thermistor was sealed in the copper tube by injecting an epoxy into the copper tube. The electrical lead wires may be seen extending from the right end of the copper tube. The tape wrapped around the copper tube was for accurate placement of the probe in the test section. The entire probe is 3 3/4 inches from end to end.

Figure 17 is a closeup of the solid state temperature sensor probe. See Appendix. It was constructed in the same manner as the thermistor probe. No tape was wrapped around the tube for placement because this particular probe must be installed from the inside of the test section due to the size of the backing plate for the sensor.

Figure 18 shows the equipment used for the Wheatstone Bridge, temperature and flow indication. The decade resistors may be seen in the left foreground. The power supplies are shown in a stacked position immediately to the right of the resistors. Next to the power supplies are the multimeters used to balance the bridge and indicate thermistor power. The Brown Recording Potentiometer for temperature is right behind the multimeters. The inclined manometer for the reference flow meter is shown in the background. It should be stated that the equipment shown in this photograph

was not used in this crowded arrangement. The equipment was placed in this manner solely for the photograph.

Figure 19 shows the equipment used to dehumidify, pressurize and control pressure. The dehydrating unit is shown in the left foreground. This unit was used to dehumidify and pressurize the closed system. One of the plastic hose connections from the unit to the closed system may be seen coming from the side of the unit. The column of water used for pressure regulation is seen in the center foreground.

Figure 20 shows the closed system constructed for this research. The 4 inch diameter pipe extends about 18 feet from the air blower in a closed loop. The air blower is on the extreme right of the photograph. The rubber hose connections were used to make pressure tight joints.

Figure 21 shows a closeup of the test section, heat exchanger and reference flow meter. The reference flow meter is in the left foreground with an arrow indicating the direction of air flow. The two plastic hose connections to the inclined manometer extend from the top of the element.

The rectangular heat exchanger is shown in the center background. The two plastic hose connections for cold supply water and warm return water extend from the top of the heat exchanger.

The test section is shown in the center foreground. The first test port on the upstream side was used for the thermistor probe. The second port was used for the thermocouple probe. The last port was used for the wet and dry bulb thermometer to measure humidity. A plastic hose pressure tap is seen extending from the back portion of the test section.

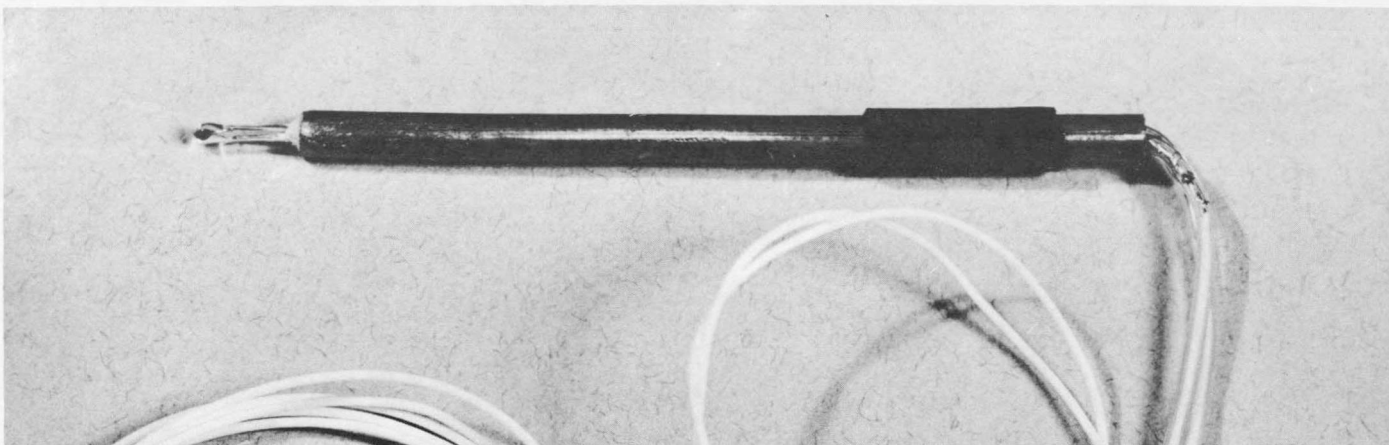


FIGURE 16. TYPICAL THERMISTOR PROBE

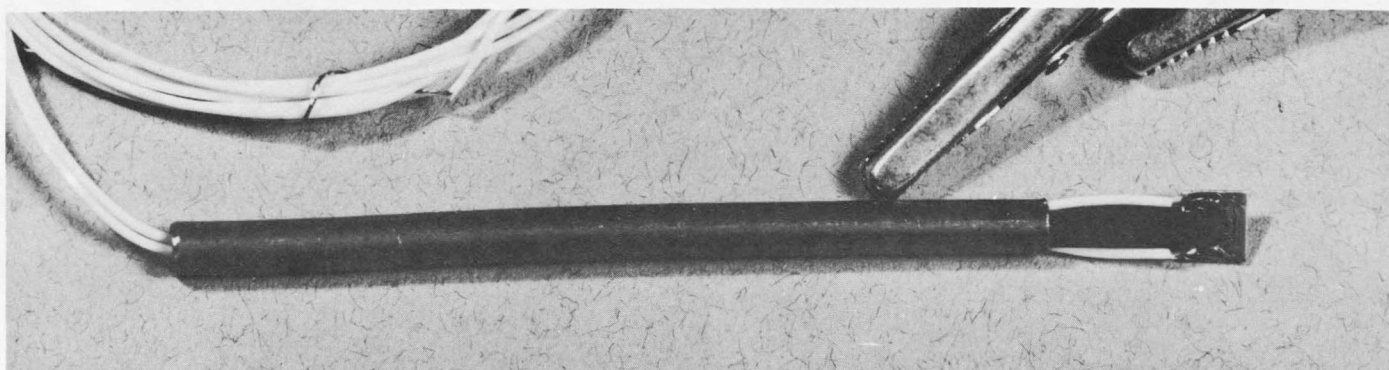


FIGURE 17. SOLID STATE TEMPERATURE SENSOR PROBE

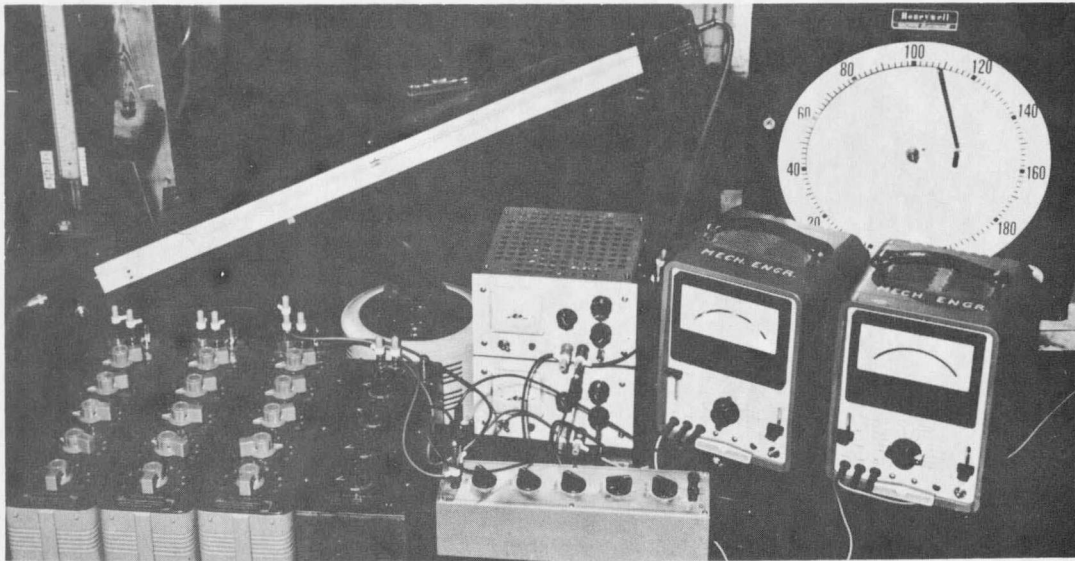


FIGURE 18. CIRCUITRY AND INSTRUMENTATION

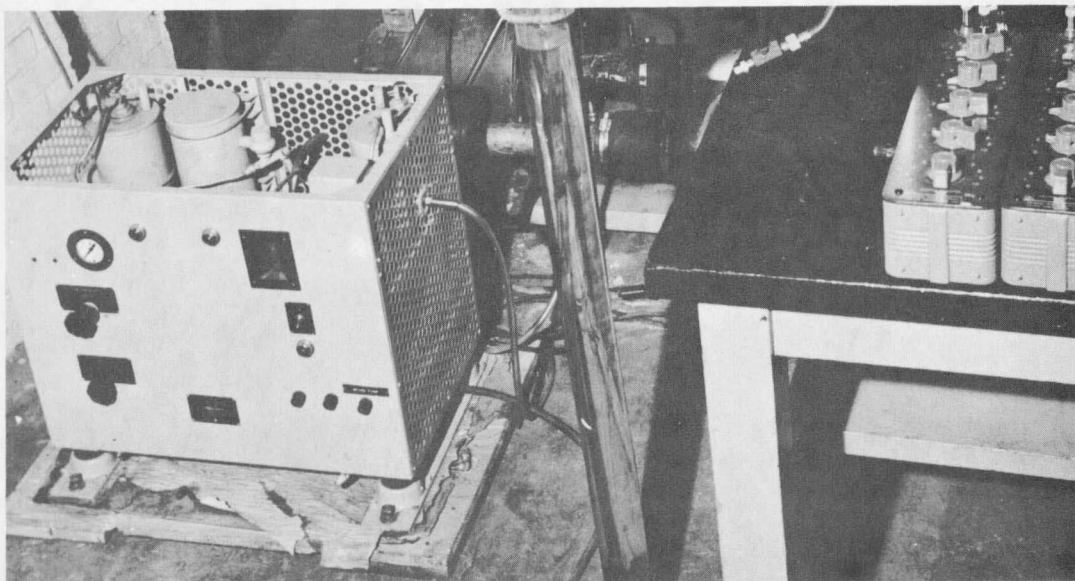


FIGURE 19. DEHUMIDIFYING AND PRESSURIZING EQUIPMENT

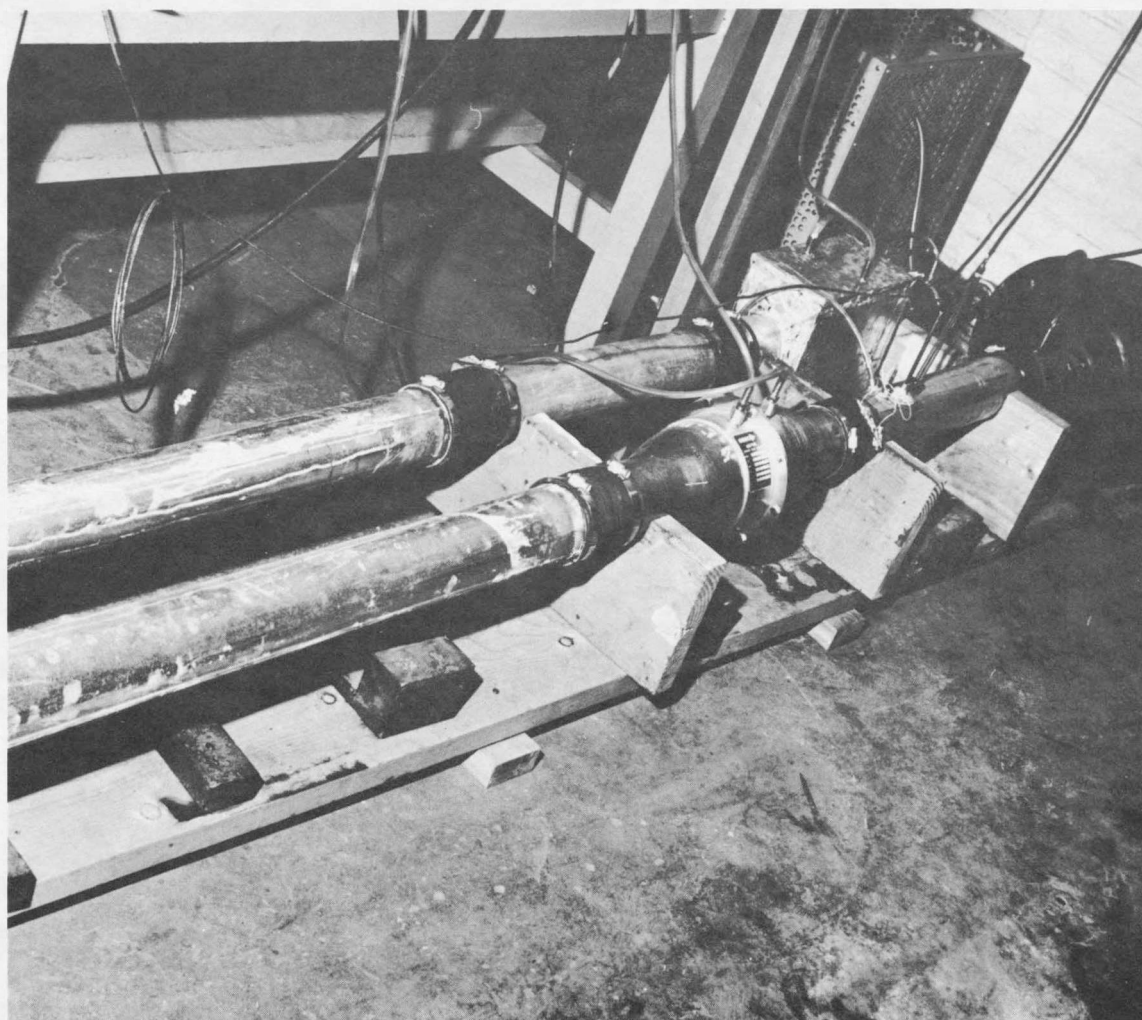


FIGURE 20. CLOSED SYSTEM AND AIR BLOWER

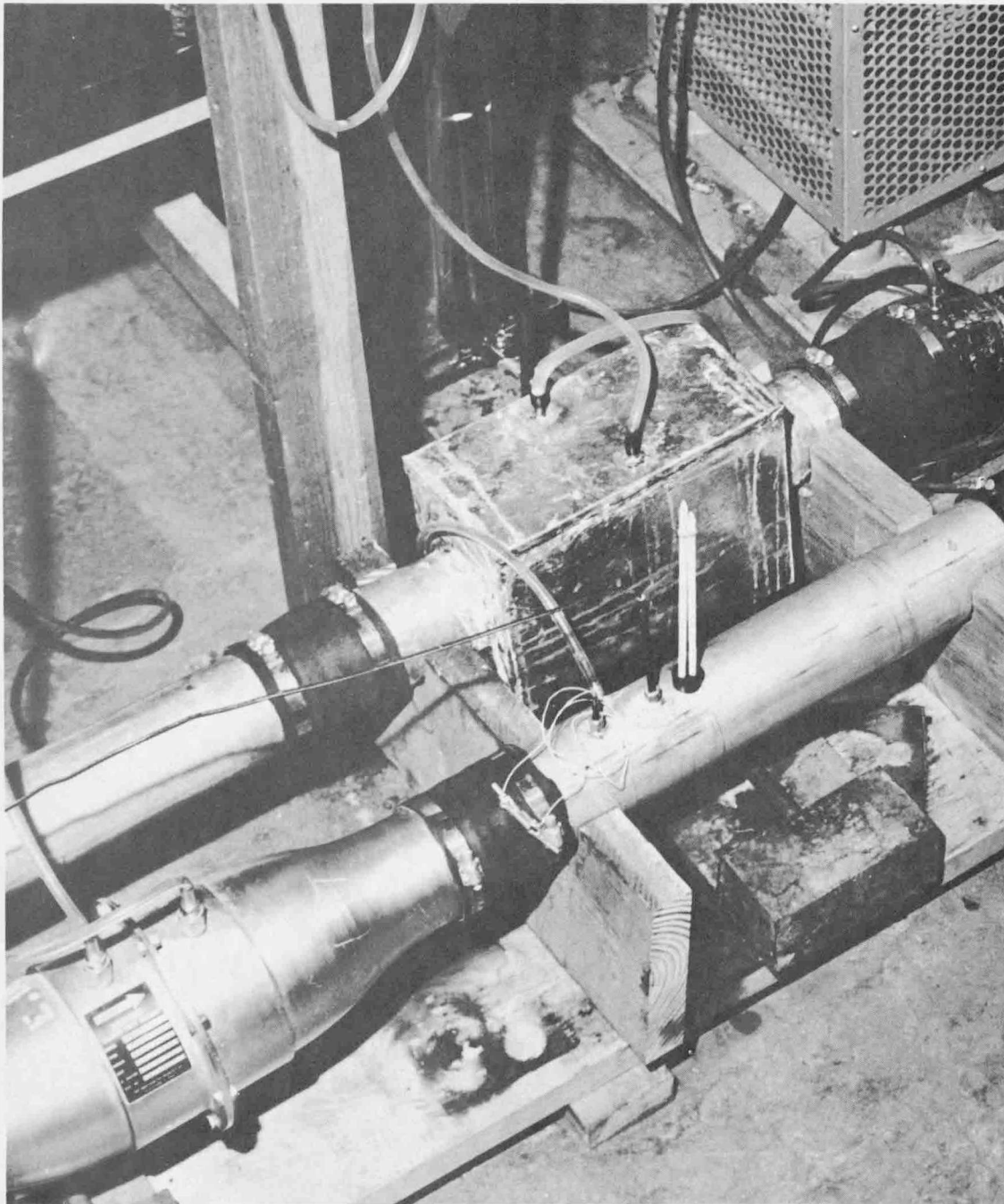


FIGURE 21. DETAIL OF TEST SECTION

CHAPTER 5

SUMMARY

This research revealed that a thermistor anemometer can be constructed which is reliable and accurate. There are two important items to consider when designing a thermistor anemometer; keep it as simple as possible and fully understand the function of the components. The thermistor anemometer used herein is certainly not the only type which could be used and may not even be the best but it was entirely satisfactory for this research. There are many variations of thermistors and circuitry which could be used.

The experiments presented here are complete in only one respect. The effect of humidity is shown to be negligible for ordinary engineering measurements over the entire range from dry air to saturated air. This encompasses all the humidity variation possible.

Pressure is shown to affect the accuracy of the velocity determination but this effect is determined from a pressure change of only 1 psig. This was the maximum range possible with the equipment used.

More research on thermistor anemometers is necessary as explained in the Appendix. Also included in the Appendix are the results of a calibration run of the anemometer with the thermistor replaced by a Micro System, Inc., Temp-Sensor. This sensor shows a lot of potential in the field of anemometry.

APPENDIX

SUGGESTED TESTS

Thermistor anemometry is a new field and requires research beyond this thesis. In addition, the recent development of solid state temperature sensors similar to thermistors opens a new area of research on flow measurement.

Additional work on thermistor anemometers is required to determine the effect of large pressure changes. It is the author's suggestion that work in this area be done with a variety of available thermistors. For this research, only glass coated bead probes were used.

Information is needed in the area of temperature compensation or calibration. For this research, temperature was simply held constant.

There is much work to be done on circuitry. This is primarily an Electrical Engineering problem. There are at least two major areas for this work. The calibration curves for a thermistor anemometer can be made linear by using the appropriate circuitry. A self-balancing bridge would be especially applicable to thermistor anemometers.

Another important area of work is to construct a thermistor flow measuring device and calibrate it in different media. With suitable calibration, this device could be used in different gases and liquids.

Micro Systems, Inc., Pasadena, California, supplied the author with free samples of their TE3-10-1000 and TE3-10-5000 Temp-Sensor* solid state temperature gages. Although not a part of the original scope of this research, one of the TE3-10-5000 sensors was installed in the test in place

*Temp-Sensor is a trade name of Micro Systems, Inc. for its solid state temperature gage.

of the thermistor.

"The Temp-Sensor solid state temperature gage is a stable, linear, rapid response, high sensitivity, and wide range device that owes its outstanding performance characteristics to careful selection of the silicon semiconductor material and to proper cutting of the crystal.

"The Temp-Sensor gage differs from the thermistor in several respects. Indeed, the only common characteristics are that both are semiconductors, and both exhibit a change of resistance with temperature. The Temp-Sensor gage has a positive temperature coefficient of resistance, and its resistance change with temperature can be so chosen as to be linear over a range of 400 °F....in contrast, the thermistor has a negative coefficient of resistance, and its resistance change with temperature is logarithmic. In addition, the Temp-Sensor gage provides considerably greater stability than the thermistor."¹

The Temp-Sensor which was installed in the test may be seen in Figure 17 on page 50. The sensor was mounted on a 0.25 inch square steel carrier when obtained from the manufacturer and could not be removed without damage. This is not an ideal shape to suspend in an air stream but other mountings can be obtained from the manufacturer upon request. The Temp-Sensor was operated exactly as the thermistor had been operated.

The Temp-Sensor was first suspended in the center of the air stream with the steel carrier perpendicular to the air stream. This calibration curve is shown in Figure 22. The Temp-Sensor was then suspended in the

1. "Temp-Sensor, Solid State Temperature Gages", Micro Systems, Inc., Pasadena, California.

center with the steel carrier parallel to the air stream. This calibration curve is shown in Figure 23.

Although the TE3-10-5000 Temp-Sensor has a linear resistance change with temperature over a range of 400 °F, the calibration curves are not linear due to the nature of the convective cooling of the air stream. The linearity could be improved by selecting a better configuration than the 0.25 inch square steel carrier for this type of work.

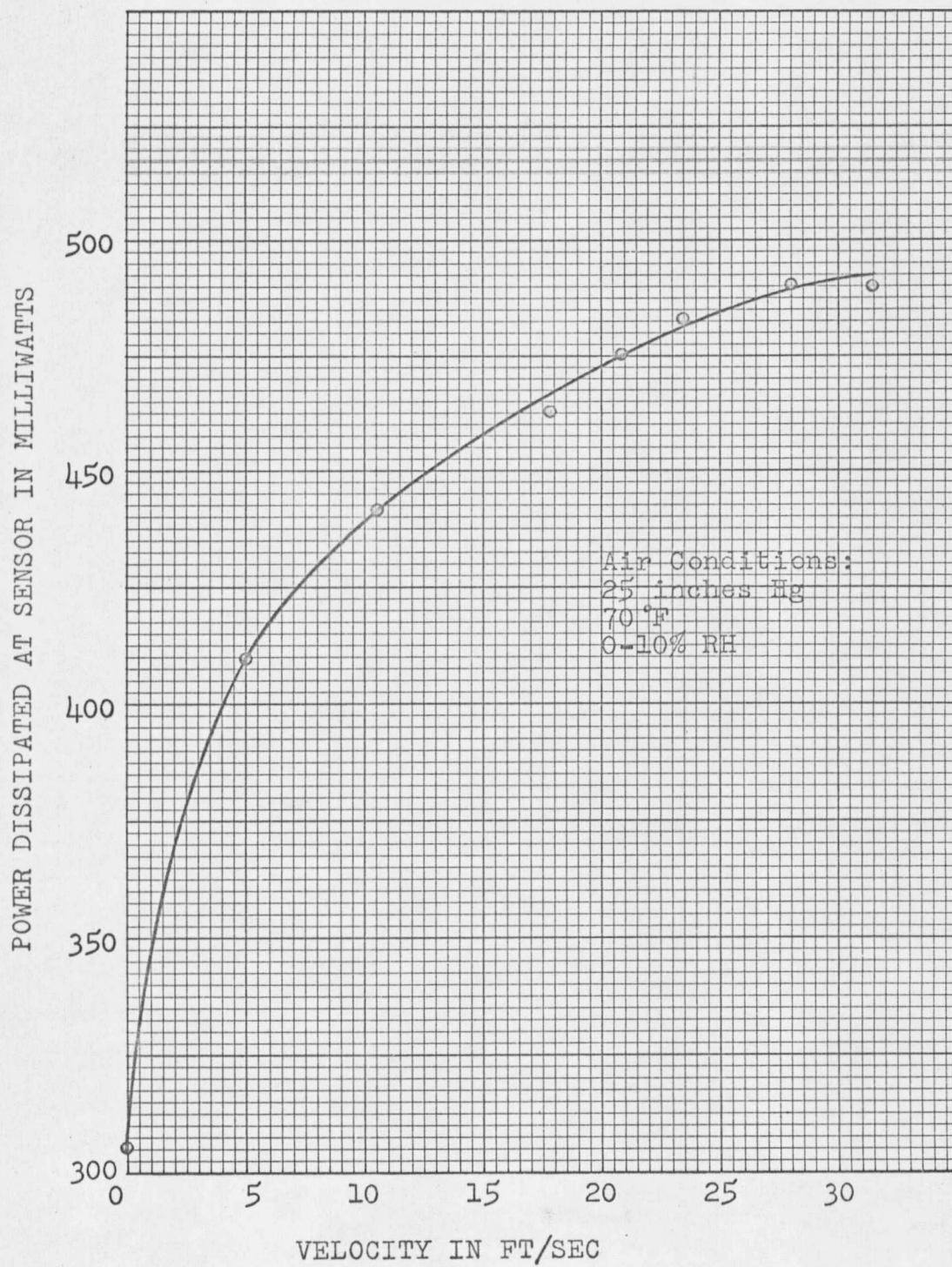


FIGURE 22. CALIBRATION CURVE FOR TE3-10-5000 SEMICONDUCTOR
TEMPERATURE SENSOR - PERPENDICULAR TO AIR FLOW

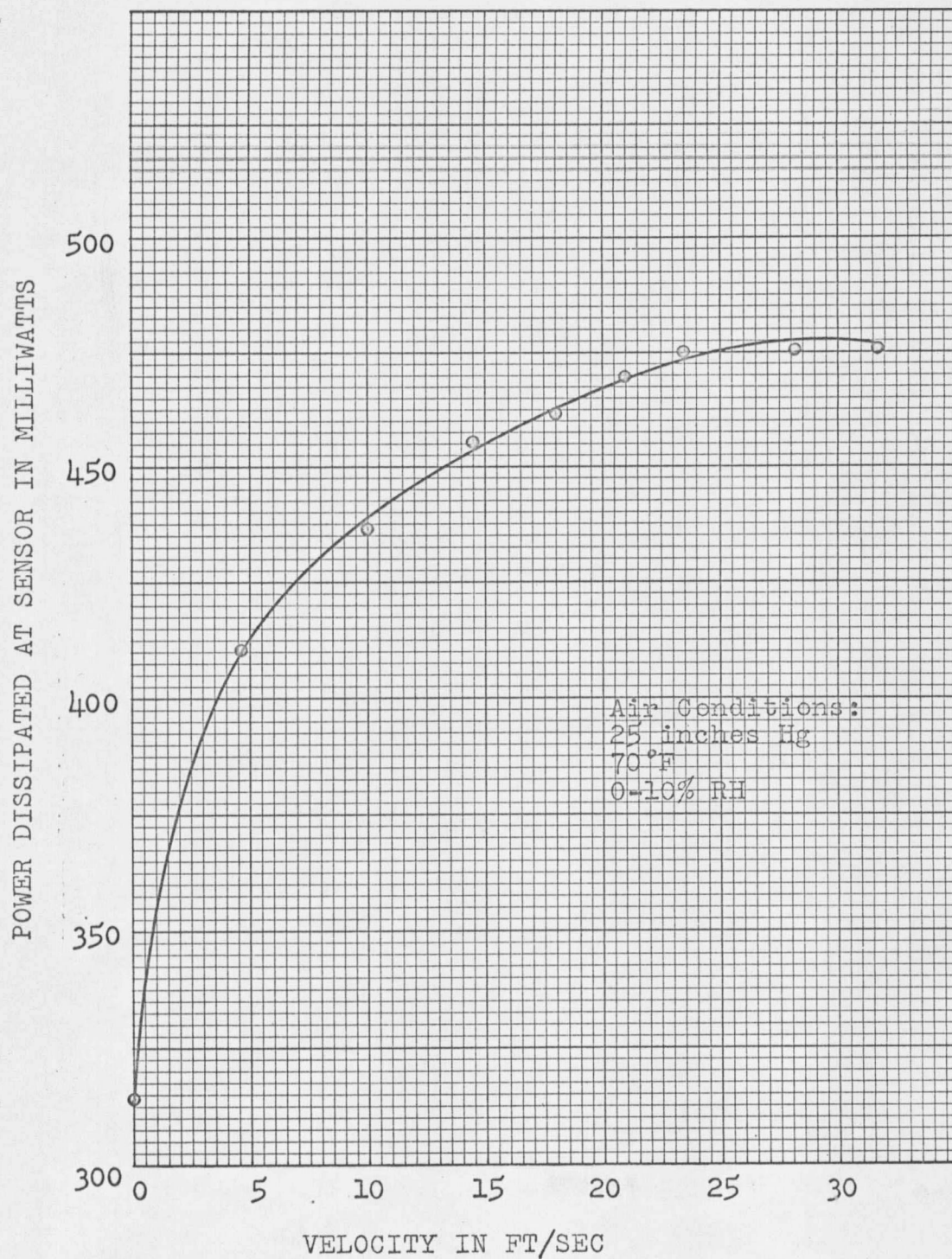


FIGURE 23. CALIBRATION CURVE FOR TE3-10-5000 SEMICONDUCTOR TEMPERATURE SENSOR - PARALLEL TO AIR FLOW

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