Thermal Studies of the Silicon Multiplicity Vertex Detector Outer Enclosure

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1. Introduction

The PHENIX experiment is being constructed at the Brookhaven National Laboratory Relativistic Heavy Ion Collider (RHIC). RHIC will be capable of accelerating gold particles to 100 GeV/A per colliding beam and will be completed in approximately the year 1999. The main goals of the PHENIX experiment are to detect a new phase of matter, the quark gluon plasma (QGP), and measure the signatures which accompany the transition from normal hadronic matter to a QGP. Of the many detectors within PHENIX, a silicon multiplicity vertex detector (MVD) is used to determine the event multiplicity and vertex position. PHENIX is located in an experimental hall whose dew point had yet to be determined at the time of these tests. The goals of these tests were to determine whether the MVD dewpoint requirements would dictate the experimental hall dewpoint, and to study the thermal characteristics of the MVD outer enclosure. Dewpoint is defined as the temperature at which water vapor in a given sample of air becomes saturated.

2. Fabrication of the Environmental Chamber

A prototype MVD enclosure was constructed to study its thermal characteristics. A copper sheet of thickness 0.05cm was formed into a copper cylinder with a radius of 42.5cm and length of

75cm. Copper was selected because of its high conductivity, enabling it to act as an efficient cold sink. Copper tubing was soldered into a helix shape and mounted inside of the cylinder. The tubing ends protruded from one side of the cylinder and were attached to a water cooling system, thus allowing water circulation inside the copper cylinder. Then 0.3cm cardboard followed by 0.005cm aluminum foil were placed around the copper cylinder. The combination of cardboard and aluminum foil plus air gap mimics the thermal properties of the proposed outer enclosure materials, Rohacell and aluminum foil. Rohacell is a light weight rigid foam. A picture of the constructed MVD prototype enclosure is shown in figure 1.

Figure 1. Picture of the prototype MVD outer enclosure.

Two temperature sensors were mounted on the inside and outside of the enclosure. The calibration plot of the temperature sensors are shown in figure 2. The sensors record resistance which is converted to temperature. Finally, the chamber was elevated five feet to allow for unimpeded air circulation.

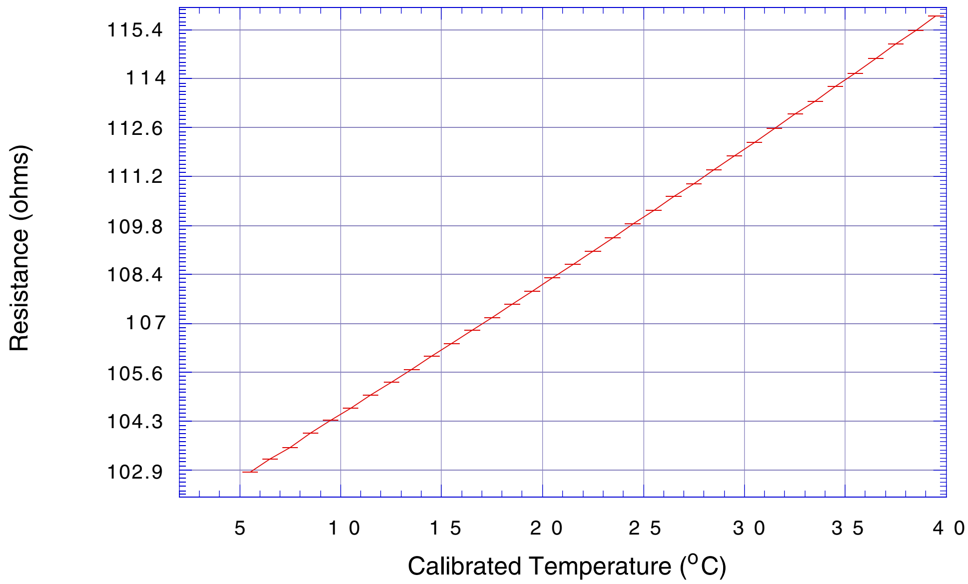

Figure 2. Calibration plot for temperature sensors in resistance versus temperature.

3. Collecting Data

The initial temperature of the enclosure environment was defined by the temperature of the water cooler. As the water circulated through the copper cylinder, the inner temperature of the enclosure became colder. When the inside temperature stabilized, the cooling bath was turned off. The enclosure was allowed to warm to room temperature while recording the following: resistance values inside (Ri=Ti) and outside (Ro=To) the enclosure, room temperature (Tr), and time.

4. Analysis of Data

Once the raw data was collected, power, thermal conductivity, and the heat transfer coefficient were evaluated. Power, Q, can be expressed as:

$$Q = (C * M * \Delta T_3)/t \qquad (1)$$

where $\Delta T_3 = Ti_{final} - Ti_{initial}$, (Ti are temperatures recorded on the inside of the prototype enclosure), t is time, M is the mass of the copper cylinder, and C is the specific heat of copper. Using equation 1, the total heat required to warm the copper sink to room temperature (Qt) was calculated to be 25kJoules. Values of Q as a function of short time intervals were also calculated from equation 1 and are plotted as a function of time in figure 3a. As time increases, the amount of heat entering the enclosure decreases until room temperature equilibrates with the temperature inside the enclosure. These numbers have significant uncertainty associated with them due to fluctuations in temperature over short time intervals. The shape of the distribution in figure 3a agrees well with an exponential shape; this behavior is confirmed by the fact that a plot of Ti as a function of time displays a reverse exponential shape. The data in figure 3a are fit with a free exponential and then the normalization of the data is varied until the expected value of Q*t from equation 1 (25kJ) is obtained. The result is shown in figure 3b.

From equation 2, the thermal conductivity, k, of the cardboard/air insulator was calculated:

$$Q = (k * A_{Cu} * \Delta T_2)/L \qquad (2)$$

where A_{Cu} is the surface area of the copper cylinder and ΔT_2 is the difference between the temperature on the outside (To) and the temperature the inside of the enclosure (Ti). The thermal conductivity was calculated to be 1.1×10^{-04} Watts/cm°C $\pm 2.3\times10^{-05}$, in comparison to the k of Rohacell, 3.3×10^{-04} Watts/cm°C. The cardboard/air insulator fairly well mimics the thermal properties of the Rohacell material.

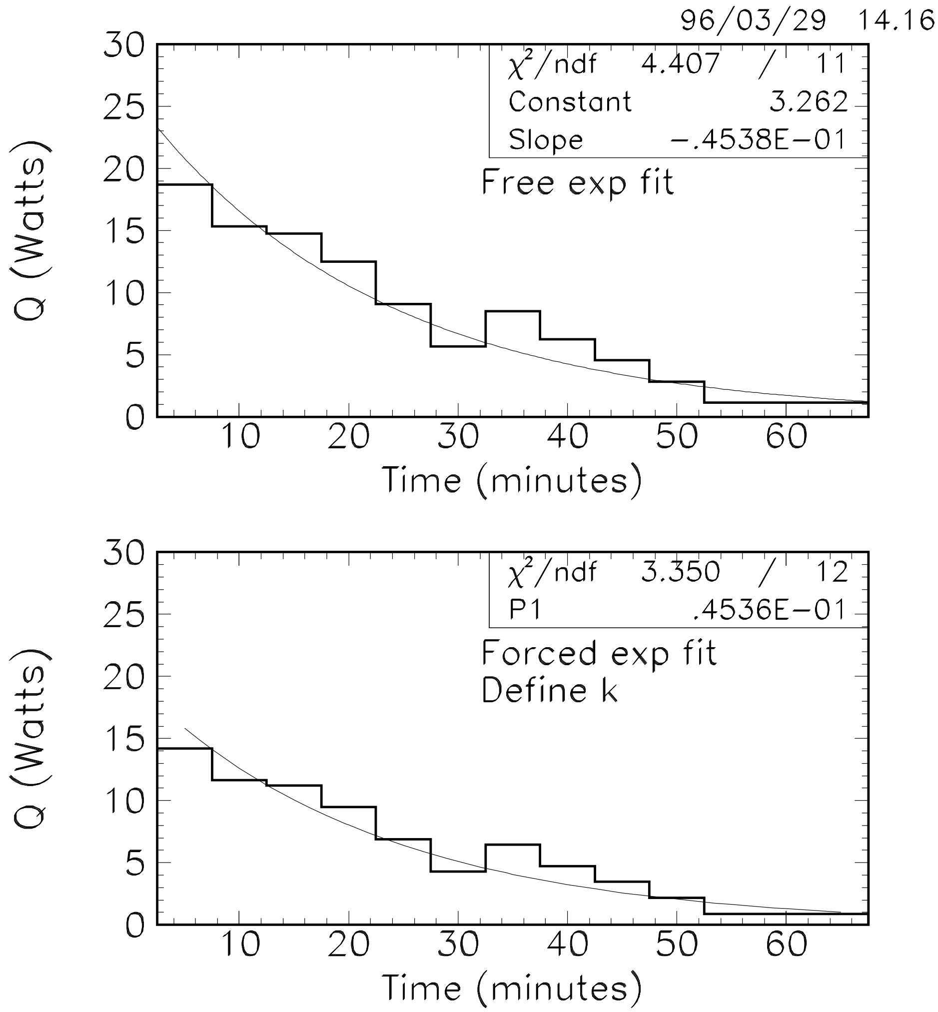

Figure 3a (top) Power as a function of time fit with a free exponential curve. 3b (bottom): Same data set fit with a forced exponential in order to yield the expected total power.

The heat transfer coefficient, h, was calculated by combining equations 2 and 3 into equation 4.

$$Q = h * A_{Cu} * \Delta T_1 \tag{3}$$

ΔT_1 is the difference between room temperature (Tr) and the temperature on the outside of the enclosure (To).

$$h = (k * \Delta T_2)/(L * \Delta T_1) \tag{4}$$

L is the thickness of the copper sheet. The heat transfer coefficient of the cardboard+air gap was calculated to be $1.6x10^{-04}$ Watts/cm^{2}°C $\underline{+}3.3x10^{-05}$ (see figure 4). It was unknown if convection patterns in and around the system would influence h over the range of test conditions; it was determined that the air convection patterns outside of the environmental chamber did not play a major role as h was found to be relatively constant.

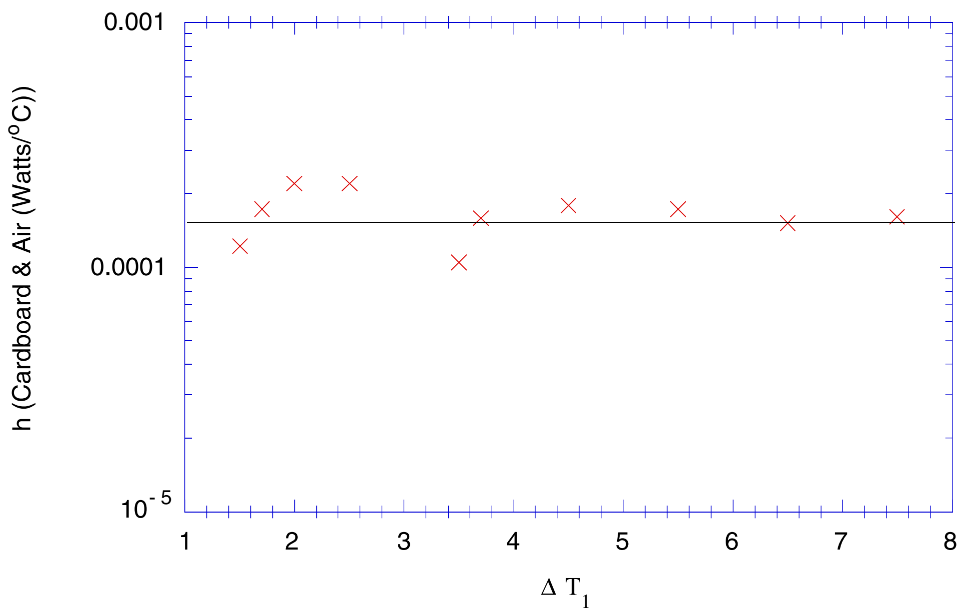

Figure 4. Calculated heat transfer coefficient as a function of ΔT_1.

With the calculated values of k and h, one can determine the MVD requirement for a dew point of the experimental hall, based on the desire that no condensation form on the outside of the enclosure. The diagram in figure 5 displays how these thermal tests can be applied to the actual MVD detector system in the experimental hall.

Heat being transferred from the hall to the inside of the enclosure will be influenced by four temperatures and three resistive barriers.

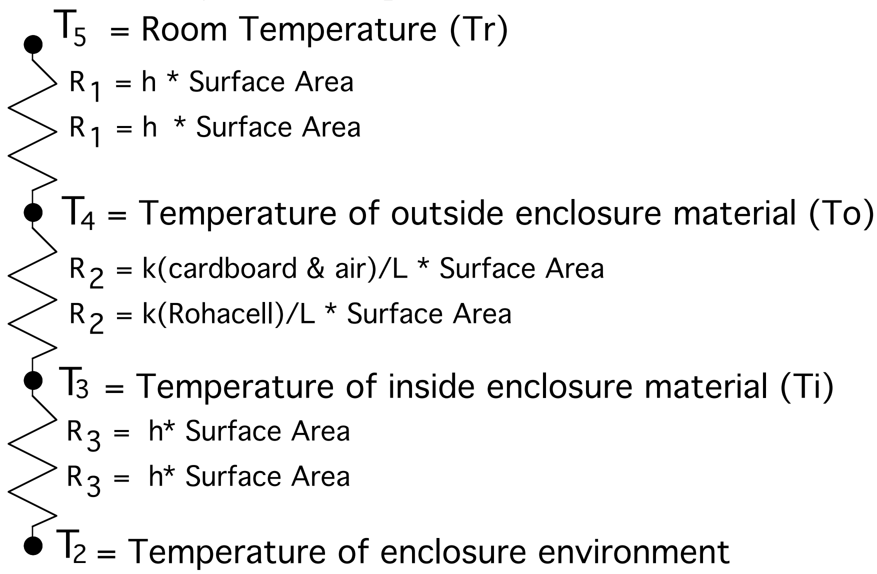

Figure 5. Resistive barriers of the MVD enclosure.

The upper limit of the experimental hall temperature (T_5) was assumed to be 32°C and the lower limit of the inside of the enclosure (T_2) was assumed to be 15°C. The variables R_1, R_2, and R_3 represent power/°C of the resistive barriers. One can calculate Q (Cardboard + air gap) of the total system from equation 5:

$$Q = \Delta T/(1/R_1 + 1/R_2 + 1/R_3) \qquad (5)$$

where $\Delta T=T_5-T_2=32°C-15°C$. From equation 5, Q is 13.46 Watts for the prototype setup. The dew point is the temperature of the outside of the enclosure (T_4) and can be solved from equation 6:

$$Q(\text{Cardboard + air gap}) = \Delta T/(1/R_1) \qquad (6)$$

where $\Delta T=T_5-T_4$. Solving for T_4 from equation 6, one obtains a value of 25 °C for the dewpoint. Similarly, one can calculate the theoretical value of the dewpoint for the actual outer enclosure using the k(Rohacell). Solving again for T_4, the theoretical dewpoint is equal to 25°C, the same as what was calculated for the prototype. Knowing the dewpoint enables one to define a range of acceptable operating temperatures and humidities for the MVD. Relative humidity is determined from the ratio of vapor pressure of water at saturation to vapor pressure of water at various temperatures which and are plotted in figure 6 [1]. The MVD outer enclosure will not experience condensation as long as the chosen operating conditions are values below the curve displayed in figure 6.

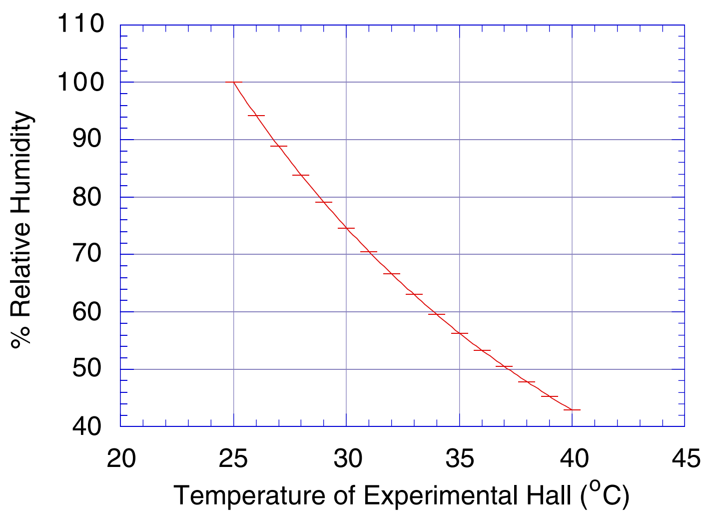

Figure 6. Range of possible operating temperatures and humidities in experimental hall, relative to the MVD.

5. Conclusion

A prototype of the MVD outer enclosure was constructed and tested for thermal properties. During the tests, no condensation was observed on the outside of the prototype. The calculated thermal properties of the prototype enclosure agreed with theoretical expectations of the actual outer enclosure. The heat transfer coefficient was found to be stable and not influenced by air convection patterns. An acceptable dew point of the experimental hall relative to the MVD was calculated to be 25°C . A range of possible operating temperatures and humidities of the experimental hall were defined. Since the time of these tests, the dewpoint of the experimental hall was defined to be 15°C. The results of our studies indicate that this is an acceptable dewpoint relative to the MVD and that the MVD does not dictate the definition of the experimental hall dewpoint.

The design of the outer enclosure appears to be thermally robust and requires no modifications to improve its thermal properties. However, the enclosure has not yet been tested for its ability to electrically isolate the MVD which will occur during a beam test in the Spring of 1996.

6. Bibliography

[1] Weast, Robert and Astle Melvin, eds., "CRC Handbook of Chemistry and Physics", 61rst edition, CRC Press Inc, Boca Raton, Florida, 1980, pg. D-196.